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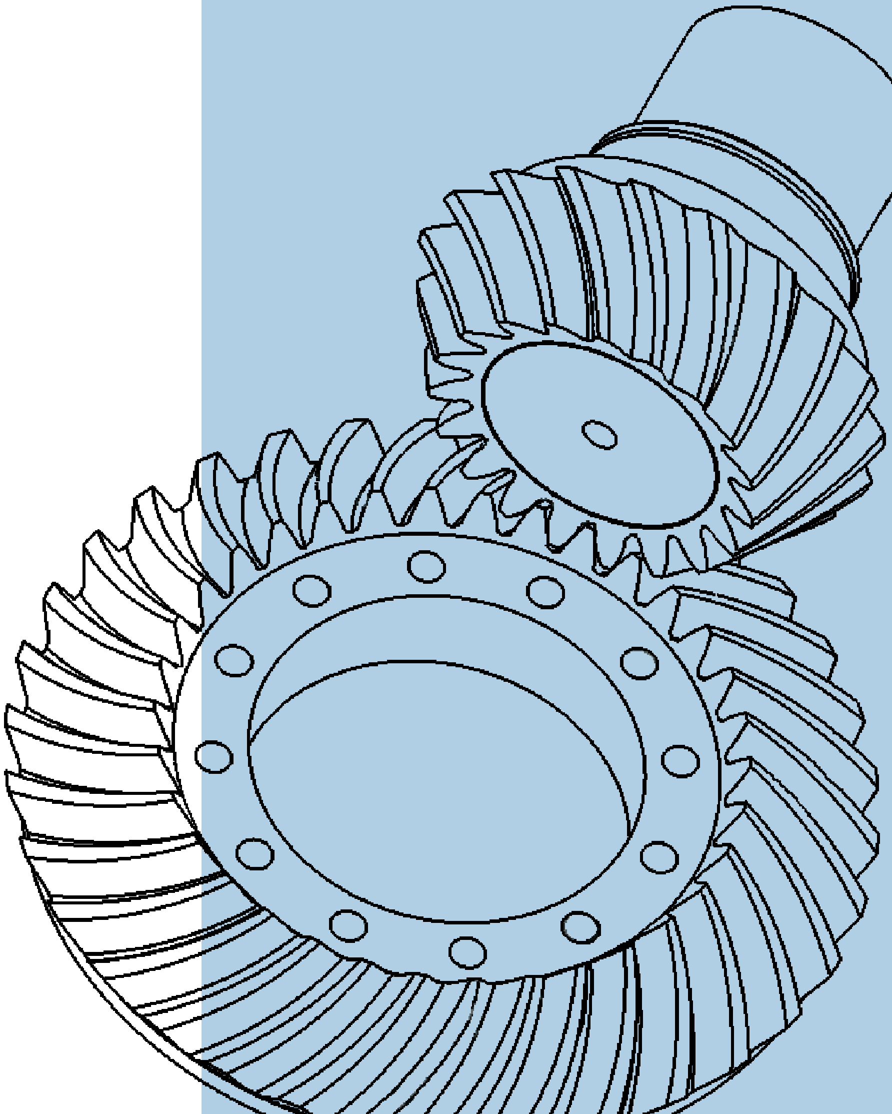

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Prilagodljivi sistem za hlajenje orodij za brizganje plastike s pomočjo termoelektričnih modulov

An Adaptive System for Cooling Injection-Moulding Moulds Via Thermoelectric Modules

Boštjan Žagar[1] - Blaž Nardin[1] - Andrej Glojek[1] - Dejan Križaj[2]
([1]Tecos, Razvojni center orodjarstva Slovenije, Celje; [2]Fakulteta za elektrotehniko, Ljubljana)

Obvladovanje temperaturnih razmer v orodjih za brizganje plastike je eden izmed pomembnejših dejavnikov pri razvoju in izdelavi orodij. Trenutno nadzor nad temperaturami temelji na sistemu hladilnih kanalov v orodjih, po katerih hladilno sredstvo dovaja ali odvaja potrebno toploto prek celotnega orodja. Zmogljivosti opisanega sistema so zaradi narave tehnologije v veliko primerih nezadostne potrebam v industriji, ki narekujejo hitre in locirane spremembe temperatur v orodjih.

Obsežna študija termodinamičnih pojavov je pokazala, da toplotni tok v orodjih lahko poteka termoelektrično. Predstavljeni sistem nadgrajuje običajne hladilne sisteme, lahko pa deluje tudi kot neodvisen sistem. V prvem primeru sestoji iz enote za merjenje temperatur v orodjih, krmilne enote in termoelektričnih elementov kot izvršilnih elementov. Samostojen sistem potrebuje dodaten prenosnik toplote. Vse enote sestavljajo sklenjeno krmilno zanko.

V predstavljenem delu pisci predstavljajo rezultate raziskovalnega dela, ki sestoji iz treh delov, rezultati pa so patentno zaščiteni v A686\2006. Poglavitni rezultati tako testnega, prototipnega kakor tudi industrijskega sklopa se izražajo v hitri in popolnoma obvladljivem krmiljenju temperature preko celotnega delovnega kroga in vplivih le-te na kakovost plastičnih izdelkov s poudarkom na obvladovanju deformacij.

Predstavljena tehnologija in uporaba sta pomemben mejnik na področju obvladovanja temperaturnih razmer in kakovosti končnih izdelkov pri postopku brizganja plastike.

(Ključne besede: brizganje polimerov, pojavi termodinamični, hlajenje orodij, moduli termoelektrični)

One of the basic problems in the development and production process of moulds for injection moulding is the control of the temperature conditions in the mould. Nowadays, the temperature is controlled through a mould cooling system, which is composed of cooling channels, where the cooling fluid transports the needed heat to and from the mould. Such systems control the heat-exchange process throughout the entire mould, which is insufficient in many cases. For this reason it is necessary to develop a new system that will allow fast and precisely located temperature changes in defined mould locations.

A precise study of the thermodynamic processes in moulds showed that heat exchange can be manipulated by thermoelectrical means. Such a system upgrades conventional cooling systems within the mould or can be a stand-alone application for the heat manipulation within it. The upgraded system comprises the temperature-detection unit, the thermo-control unit and the final controlling element (thermoelement module as a heat pump), while the stand-alone unit needs an additional heat exchanger. These units form an adaptive closed loop.

In this paper we present the results of a research project that was carried out in three phases, and its results are included in the A686\2006 patent. The testing stage, the prototype stage and the industrialisation phase will be presented. The main results of the project were the total and rapid online thermo-control of the mould over the cycle time and the overall influence on the quality of the plastic product, with an emphasis on deformation control.

This system represents a milestone in the field of mould-temperature and product-quality control during the injection-moulding process.

(Keywords: injection moulding, thermal processes, injection mould cooling, thermoelectric modules)

0 UVOD

Potreba po razvoju hladilnih sistemov s termoelektričnimi moduli (TEM) izhaja iz vsakdanjih problemov pri konstruiranju, izdelavi in uporabi orodij. Hladilni sistemi, ki so trenutno v uporabi, temeljijo na sistemu hladilnih kanalov, po katerih hladilno sredstvo prenaša toploto iz želene lokacije ali nanjo. Navadno se kot hladilno sredstvo uporablja voda, v temperaturno zahtevnejših primerih (gretje) se uporablja olje, za hlajenja pa podhlajeni plini. Vse naštete tehnologije imajo tehnološke omejitve, ki jih s sodobnimi računalniškimi paketi (metode končnih elementov, v nadaljevanju MKE) lahko napovemo, ne moremo pa se jim popolnoma izogniti (poglavje 2).

Rezultati najnovešje analize so pokazali, da trenutne tehnologije hlajenja s svojimi pomanjkljivostmi ne zadoščajo več vsakodnevno zahtevnejšim okvirom tehnoloških oken predelave polimerov.

Glavna omejitev predelave polimerov je trenutna nezmožnost hladilnih sistemov glede skrajševanja delovnih krogov in posledično zmanjševanja stroškov. Trenutne tehnološke (stroji) in polimerne zmožnosti nadaljnjega naglega optimiranja postopka so izčrpane [3].

Predelave polimerov temeljijo na prehodu toplote med plastičnim materialom in orodno votlino. Zaradi vseh navedenih dejstev in omejitev trenutno uporabljene tehnologije je očitna potreba po razvoju novejših tehnologij in njihovo uvajanje v industrijsko prakso.

Prenos predstavljene tehnologije v vsakdanjo prakso je bil izveden na postopku brizganja plastike, čeprav je možnost uporabe širša; v veliki večini proizvodnih postopkov, ki temeljijo na prehodu toplote.

1 TERMODINAMIČNI POJAVI V POSTOPKIH BRIZGANJA PLASTIKE

Pri postopku brizganja plastike se morata preučiti in upoštevati dve poglavitni dejstvi; celotna porabljena energija, ki temelji na prvem zakonu termodinamike (zakon o ohranitvi energije), ter hitrost prenosa toplote [1].

Osnovni nalogi termodinamične analize orodij za brizganje plastike sta časovni potek temperatur prek delovnega kroga ter fizična porazdelitev temperatur po orodni votlini. Slednja porazdelitev je

0 INTRODUCTION

The development of the technology of cooling moulds via thermoelectrical (TEM) means derives from industrial practise and problems, i.e., during the design, the tool making and the exploitation of tools. Current cooling technologies are based on a cooling-channels system through which the cooling fluid transports heat to or from the desired location. Water is commonly used as the transport medium; however, oil is used in cases of high temperature (heating function), and under-cooled gases are used in cases of high cooling demands. All of these principles have technological limitations, and these limitations can be located and predicted in advance with finite element analyses (FEA) simulation packages; however, they cannot be completely avoided (Section 2).

The results of a diverse state-of-the-art analysis revealed that all existing cooling systems do not provide controllable heat-transfer capabilities appropriate for the demanding technological windows of current polymer-processing technologies.

Polymer processing is nowadays limited (in terms of shortening the production cycle time and within that reducing costs) only by the heat-capacity manipulation capabilities. Other production optimization capabilities have already been driven to the mechanical and polymer processing limitations [3].

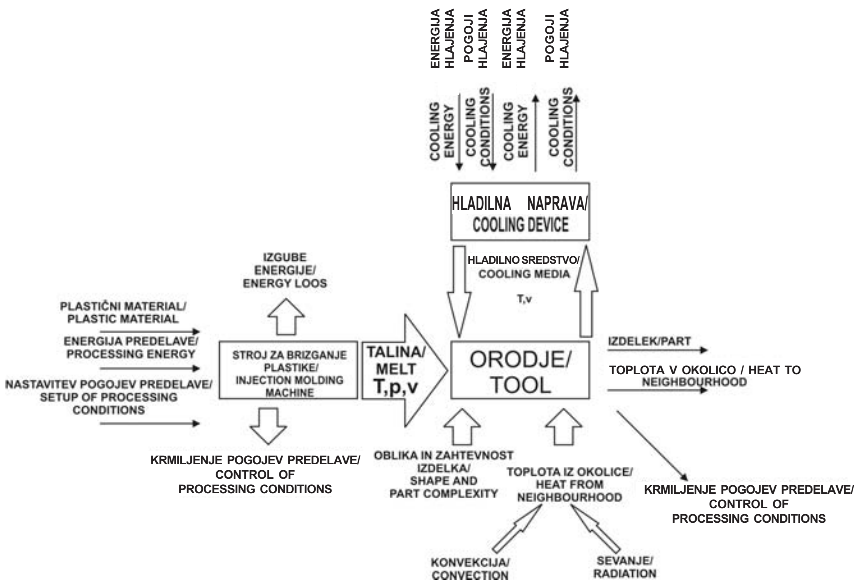
Polymer processing is based on the heat transfer between the plastic material and the mould cavity. Based on this and the other mentioned technological and economic facts it is justified to invest in and invent new technologies for heat manipulation in polymer processing and transfer them to everyday industrial practise.

The transfer of TEM cooling technology to the industrial phase has been applied through the injection-moulding process, although it can be applied during most processes where heat manipulation is required.

1 THERMAL PROCESSES IN INJECTION-MOULDING PLASTIC PROCESSING

As part of the calculation of heat transfer two major facts must be considered. First, all the used energy that is based on the first law of thermodynamics, i.e., the law of energy conservation [1], and the velocity of heat transfer.

The basic task during heat-transfer analyses is the calculation of temperature over time and its distribution inside the studied system. This last task depends on the velocity of the heat transfer between



Sl. 1. *Termodinamična shema postopka brizganja plastike [1]*
Fig. 1. *Thermodynamic block diagram for the injection-moulding process [1]*

odvisna od hitrosti prenosa toplote med opazovanim sistemom in okolico ter hitrostjo prenosa toplote v samem opazovanem sistemu [1]. Prenos toplote v orodjih temelji na prevodu, prestopu in sevanju [1], ki so nazorno predstavljeni na sliki 1.

the system and its surroundings, and the velocity of the heat transfer inside the system. Heat transfer is based on heat conduction, convection or radiation [1], and is described in the block diagram shown in Figure 1.

1.1 Čas hlajenja

Delovni krog postopka brizganja plastike je sestavljen iz faze zapiranja orodja, brizga taline v orodno votlino, faze dodatnega tlaka za izravnavo skrčkov, faze hlajenja, faze odpiranja orodja in izmetavanja izdelka. V večini primerov je čas hlajenja najdaljši od vseh navedenih časov, kar pomeni tudi stroškovno največjo postavko v ceni izdelave in upravičuje nujnost raziskav na tem področju.

Čas hlajenja v postopku brizganja plastike je določen kot čas, potreben za ohladitev plastičnega kosa do temperature izmetavanja in je podan v enačbi:

1.1 Cooling time

The complete injection-moulding process cycle comprises the mould-closing phase, the injection of the melt into the cavity, the packing-pressure phase for compensating the shrinkage effect, the cooling phase, the mould-opening phase and the part-ejection phase. In most cases the longest of all the phases described above is the cooling phase. Because it is closely related to costs, it is clear why it is reasonable to explore this field.

The cooling time in the injection-moulding process is defined as the time needed to cool the plastic part to the ejection temperature, and it is described by the following equation:

$$t_h = t_u + t_{np} + t_{ps} + t_{mr} + t_d \qquad (1),$$

kjer so:
t_h čas hlajenja, t_u čas brizga, t_{np} čas dodatnega tlaka, t_{ps} časa plastifikacije, t_{mr} časa morebitnega pomika plastifikatorja ter t_d kot dodatnega časa hlajenja [1].

with:
t_h as the cooling time, t_u as the injection time, t_{np} as the packing-pressure time, t_{ps} as the plastification time, t_{mv} as the movement of the injection time to the zero position and t_d as the additional cooling time [1].

Poglavitni cilj optimiranja postopka in razvoja predstavljene tehnologije je skrajšati čas t_d [1], ki je teoretično nepotreben, v praksi pa zaseda od 45% do 67% celotnega delovnega kroga.

Razmerje med pomembnimi temperaturami v postopku brizganja plastike so izkustveno podane z razmerjem:

The main aim of a cooling process is to lower t_d [1], which in theory is not necessary, but in practise it extends from 45% up to 67% of the whole cycle time.

The relationship between the important temperatures in the process of injection moulding is experimentally given by the following ratio:

$$\Delta T_U : \Delta T_K : \Delta T_T = 6 : 5 : 1 \qquad (2),$$

kjer so:
ΔT_U sprememba temperature izmetavanja, ΔT_K sprememba temperature orodja in ΔT_T sprememba temperature taline.

Iz slednjega razmerja (2) je razvidno, kako pomembno vlogo ima temperatura orodja.

where:
ΔT_U is the deviation of the ejection temperature, ΔT_K is the deviation of the mould temperature and ΔT_T is the deviation of the melt temperature.

From this relationship one can conclude that the mould temperature has an enormous influence on the ejection time and therefore on the cooling time.

1.2 Temperaturne razmere v postopka brizganja plastike

Postopek brizganja plastike je krožni postopek, kakršen je tudi potek temperature površine orodne votline (sl. 2), ki se spreminja okoli srednje vrednosti.

Najvišja temperatura orodne votline je določena kot:

1.2 Temperature conditions in the injection-moulding processes

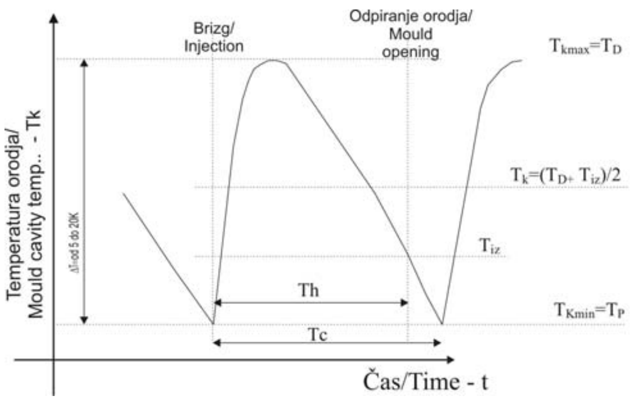
The injection-moulding process is a cyclic process where the mould temperature varies. As shown in Figure 2, the temperature varies around the average value through the whole cycle time.

The maximum mould temperature is defined as:

$$T_{KMAX} = T_D = \frac{T_P b_k + T_T b_P}{b_k + b_P} \qquad (3),$$

kjer so:
$T_P = T_{KMIN}$ začetna temperatura orodne votline (nastavljiva vrednost), T_{KMAX} največja dosežena temperatura v orodni votlini, b_k toplotna prevodnost orodnih vložkov, b_P toplotna prevodnost polimera,

where:
$T_P = T_{KMIN}$ is the initial mould temperature of the cavity (adjustable temperature), T_{KMAX} is the maximum achieved temperature in the mould cavity, b_k is the mould-cavity heat conductivity, b_P is the polymer



Sl. 2. *Spreminjanje temperature orodja preko delovnega kroga [2]*
Fig. 2. *Mould-temperature during one cycle [2]*

T_P začetna temperatura orodnega vložka in T_T temperatura polimerne taline.

Enačba časa hlajenja plastičnega kosa se glasi:

$$t_h = \frac{b_0^{\,2}}{K_0 \pi^2 a_{ef}} \ln\left(K_U \frac{T_T - T_K}{T_U - T_K} \right) \quad (4),$$

kjer so:
b_0 značilna izmera izdelka, a_{ef} toplotno prevodnostni koeficient snovi, K_U faktor oblike, T_T temperatura taline, T_K temperatura orodne vlo□ka ter T_U temperatura izmetavanja.

Termodinamični pojavi v orodjih za brizganje plastike (in drugih izdelovalnih postopkih) so dandanes dobro popisani s sodobnimi paketi MKE. Pred tem so bili v pomoč konstrukterjem orodij v pomoč t.i. nomogrami, diagrami z informacijami o porazdelitvi, prerezi itn. hladilnih kanalov glede na lastnosti plastičnega kosa. Trenutno se pri večini konstrukcij orodij uporabljata analiza in optimizacija termodinamičnih lastnosti orodja s sodobnimi paketi MKE zaradi tehnološke in ekonomske optimizacije proizvodnje. Zaradi navedenih analiz so omejitve hladilnih sistemov povezane in izvirajo samo iz omejitev hladilnega sredstva in konstrukcije hladilnega sistema (poglavje 2).

Opisane metode analiz v poglavju 2 so osnova nadaljnjemu razvoju sedanjih in v poglavju 4 opisani predstavljeni rešitvi hlajenja.

2 SEDANJI SISTEMI HLAJENJA V ORODJIH ZA BRIZGANJE PLASTIKE

Večina novih razvojev orodij gre preko postopka optimizacije hlajenja. Sodobni paketi termodinamičnih analiz temeljijo na metodah končnih elementov. Vstopni podatek slednjim analizam je več mogočih rešitev postavitve hladilnih kanalov, rezultati pa so natančne ocene termodinamične slike in potrebnega časa hlajenja.

2.1 Optimizacija hlajenja

Rezultati analiz poleg časa delovnega kroga (cena izdelave) obsegajo tudi posledice vpliva hlajenja na plastičnem kosu (kakovost izdelka).
a) Z optimizacijo temperiranja pridobimo vpliv na lastnosti kakovosti plastičnega kosa:

- videz površine in njena kakovost,
- zaostale mehanske napetosti,
- kristalizacijo,

heat conductivity, T_P is the mould-cavity starting temperature, and T_T is the polymer melt temperature.

The cooling time of the injection-moulded part time equation is:

where:
b_0 is the characteristic dimension, a_{ef} is the heat-conduction coefficient of the material, K_U is the shape coefficient, T_T is the melt temperature, T_K is the mould cavity temperature, and T_U is the ejection temperature.

The thermodynamic processes in the moulds for injection moulding (and other processing tools) can be well described using modern FEA tools. Common diagrams (nomograms) on design and thermodynamic relationships were used in the mould-design practise. Cooling-system design is nowadays carried out and analyzed by FEA tools in order to optimize the technological and economic aspects of production. That is why current cooling-system capabilities are limited only by the cooling media's properties and the construction limitation (Section 2).

The described methods for temperature control are the basis for extended use in further research activities. A special challenge is foreseen in the use of the described methods in new systems of cooling the moulds, which is presented in Section 4.

2 CURRENT COOLING TECHNOLOGY FOR PLASTIC INJECTION MOULDS

Most newly designed plastic parts go through an optimization of the cooling system before the realization of the mould. The modern concept of thermodynamic analyses is based on numerical FEA methods, and the results are a precise prediction of the production cycle based on the required cooling time.

2.1 Optimization of cooling

With the results of the cooling-system FEAs the cooling's consequences on the plastic part can be precisely explored in terms of product quality and cost.
a) With the optimization of the cooling, several plastic-part quality properties can be controlled:

- the surface appearance and quality,
- the residual stress rate,
- the crystallisation,

- toplotne deformacije,
- velikost in smer toplotnega zvijanja,
- razsežno stabilnost,
- sestavo materiala in usmeritev vlaken.

b) Prav tako pa so vsebina optimizacije tudi časi hlajenja in s tem povezana čas in cena izdelave:

- temperatura izmetavanja,
- delovni čas proizvodnje.

Oba navedena rezultata optimizacije (kakovost in cena izdelave plastičnega kosa) sta s trenutnimi orodji MKE natančno določljiva. Potek optimizacije temelji na iteraciji različnih možnosti postavitve hladilnih kanalov v orodju (spremembe računališkega modela orodja) in primerjanju medsebojnih rezultatov. Zaradi slednjih iteracij so opisane optimizacije inženirsko zapletene in časovno potratne (poleg časa delovnega kroga se optimirajo tudi lastnosti navedene v 2.1.a. Nenazadnje so zmogljivosti optimizacije hlajenja omejene z zmožnostmi hladilnega sredstva in same tehnologije hlajenja, opisane v 2.2 [3].

2.2 Konstrukcija hladilnih kanalov

Postavitev hladilnih kanalov v praksi poteka na podlagi izkušenj in razpoložljivega prostora v orodjih. Glede na omenjeno neenakomernost postavitve prihaja do oblikovanja t.i. "mrtvih con", predstavljenih na slikah 3a in 3b. Mogoči izboljšavi tega problema sta povečanje števila hladilnih kanalov in zmanjšanje njihovega premera; slednje je navzdol omejeno z zamašitvijo kanalov zaradi odlaganja vodnega kamna (1 mm vodnega kamna ima enako toplotno upornost kakor 50 mm orodnega jekla [3]).

Na sliki 4 so vidne posledice neenakomerne postavitve hladilnih kanalov; natančneje, problemi učinka kota in dolgih tankih sten. Velike temperaturne spremembe po površinah orodja (sl. 4 levo) se kažejo v velikih zaostalih mehanskih napetostih, toplotnih deformacijah in razsežnostne nestabilnosti.

- the thermal deformation,
- the magnitude and direction of the thermal deflection,
- the dimensional stability of the product,
- the structure of the material and the fibers' orientation.

b) Cost control as a consequence of manipulating the cooling time

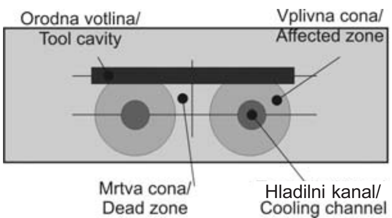
- the ejection temperature,
- the cycle time of the production.

Both stated issues (the quality and cost control) can be analyzed using state-of-the-art FEA tools. The optimization of the cooling is realized via an analyzed iteration, where the computer-aided design (CAD) of a design (cooling-system positioning) needs to be changed, and each such configuration later checked in the FEA environment for thermodynamic results. The described optimizations are, therefore, complex and time consuming (besides the cooling optimization, other properties stated in 2.1.a are also being optimized) but still cost efficient in terms of production cost saving. Finally, most optimization cases are limited by the current cooling system's capabilities, and its properties are described in 2.2. [3].

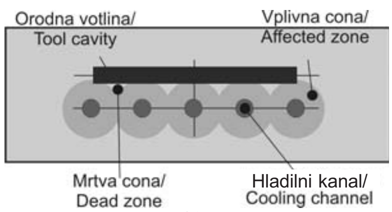
2.2 Design of the cooling channels

The positioning of the cooling channels is, in practise, based on experience and the available space inside the mould. Due to the fact that cooling is unevenly distributed the formation of "dead zones", seen in Figures 3a and 3b is uneven. A possible solution to this problem is increasing the number of channels and decreasing the diameter of them where one must consider the problem of stuffing the channels with water limestone (1 mm of water limestone has a similar heat resistance to 50 mm of tool steel [3]).

The direct consequences of unevenly distributed cooling channels (corner effects, long thin ribs) can be seen in Figure 4. A high temperature variation results in a residual stress rate, thermal deformation and dimensional instability.



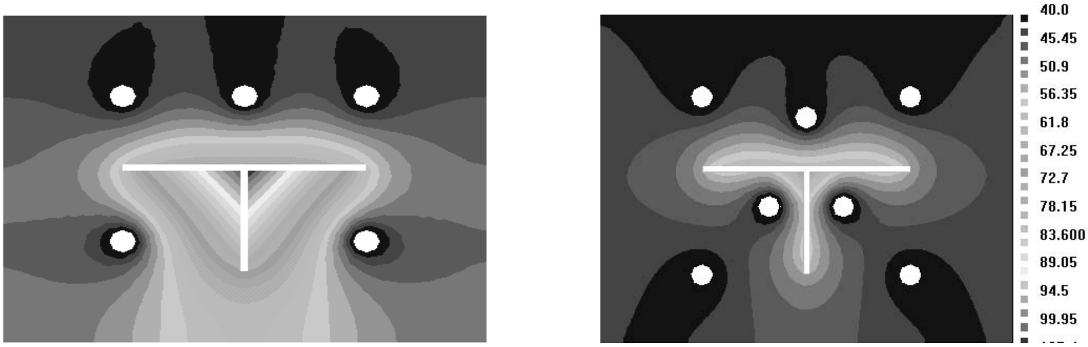
a)



b)

Sl. 3. *Vplivne cone glede na postavitev hladilnih kanalov*

Fig. 3. *Affected area due to positioning of the cooling channels*



Sl. 4. *Analiza učinka hlajenja kota z MKE*
Fig. 4. *FEA of a corner cooling effect*

Z uporabo termodinamičnih paketov MKE pri razvoju orodja se lahko temperaturno problematična mesta postavijo in kasneje z iteracijo variacij postavitve hladilnih kanalov (v okolju računalniško podprtega načrtovanja - RPN) se poišče temperaturno najboljšo rešitev. S tem se tehnološke omejitve trenutno uporabljenih tehnologij oz. posledice le-teh lahko omilijo ne moremo pa se jim izogniti.

Zaradi navedenih omejitev je treba razviti novo tehnologijo hlajenja, ki bo omogočala njihovo odpravo. Ena izmed takšnih je osnovana na termoelektričnih elementih.

3 HLAJENJE S TERMOELEKTRIČNIMI MODULI

Pobuda za razvoj nove hladilne tehnologije izhaja iz industrije in opisanih problemov, s katerimi se industrija vsakodnevno srečuje. Uporaba, opisana v nadaljevanju pa se lahko uporablja tudi v drugih pojavih in postopkih s potrebo po hitrih in nadzorovanih spremembah temperature.

3.1 Termoelektrični moduli (TEM)

Zveza med električnimi in toplotnimi lastnostmi temelji na Peltier-ovem učinku. TEM (sl. 5) je sestavljen iz mreže urejenih parov polprevodnikov tipa P in N. Slednji so vmeščeni med dve keramični osnovi, ki sta hkrati tudi prevodnika toplote TEM. Moč in smer črpanja toplote je enoznačno definirana s polarito in višino napajalne električne napetosti [4].

3.2 Uporaba za hlajenje orodij za brizganje plastike

Temelj uporabe je vgradnja TEM-ov v stene orodne votline, ki prevzemajo vloge ustvarjanja

With use of FEM packages during the design phase of the part and the mould for the injection moulding problematic spots can be located in advance. With an iterative approach in computer-aided engineering (CAE) the phase physical reparation of moulds can be avoided, although water-cooling technology limitations cannot be avoided.

To avoid all the mentioned problems and to enhance the cooling process in the mould some new approaches need to be considered. One of such approaches is cooling-process control by electrical means.

3 COOLING APPLICATIONS WITH THERMOELECTRIC MODULES

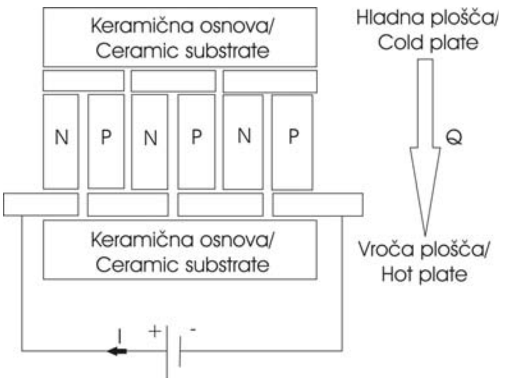
The initiative for exploring new technology has come from industry on the basis of the described facts and the current cooling-technology limitations. The solutions described in this paper can be used in applications where technology deals with heat transfer.

3.1 Thermoelectric modules (TEM)

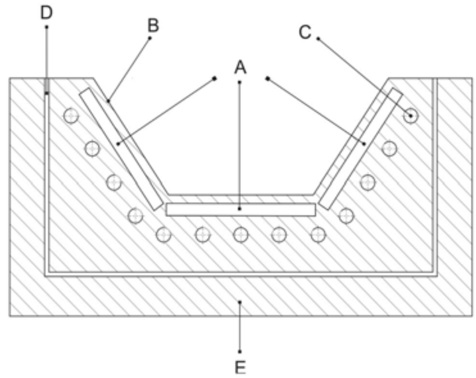
The relationship between the heat and the electrical variables is based on the Peltier effect. A TEM module (see Figure 5) is a device composed of properly arranged pairs of P- and N-type semiconductors that are positioned between two ceramic plates forming the hot and cold thermoelectric cooler sites. The power of heat transfer can be easily controlled through the magnitude and the polarity of the supplied voltage [4].

3.2 Application for mould cooling

The main idea of the application is inserting TEM modules into walls of the mould cavity to serve



Sl. 5. *Shema termoelektričnega modula*
Fig. 5. *Thermoelectric module block diagram*



Sl. 6. *Vgradnja TEM-a v orodje*
Fig. 6. *Structure of TEM cooling assembly*

primarnega toka toplote. Takšna osnovna zgradba sklopa je predstavljena na sliki 6. Sekundarni tok toplote je uresničen z običajnim hladilnim sistemom, ki odvaja/dovaja toploto iz okolice ali v okolico do termodinamičnega sistema orodja.

Sklop, predstavljen na sliki 6, sestoji iz termoelektričnih modulov (A), ki zagotavljajo primarni prenos toplote s temperaturno nadzorovane površine orodja ali nanjo (B). Sekundarni prenos toplote je uresničen z običajno tehnologijo hladilnih kanalov (C); omenjeno sekundarno toploto pa TEM-i prečrpavajo po nastavljivih časovnih in temperaturnih potekih. Za zmanjšanje toplotne vztrajnosti oblikovnega vložka orodja (F) skrbi toplotna izolacija (D), ki oblikovni vložek ločuje od toplotne vztrajnosti preostalih delov orodja (E).

Celotna uporaba je sestavljena iz TEM-ov (sl. 7), temperaturnih zaznaval in elektronske enote, ki skrbi za pravilno delovanje sistema. Poleg tega sistem vsebuje tudi vhodno enoto (uporabniški vmesnik), napajalno enoto ter močnostno elektronsko enoto (most H).

Vsi omenjeni elementi sestavljajo zaključeno krmilno zanko, prek katere elektronska enota poskuša slediti želenim temperaturno časovnim potekom.

Opisani sistem je zmožen delovanja v režimu hlajenja ali gretja, kar omogoča popoln nadzor temperatur v orodju prek celotnega delovnega kroga. Poleg tega omogoča sledenje različnim temperaturno časovnim potekom; tako za zagonske kakor ustavitvene režime delovanja.

Izvedba je s slednjo lastnostjo uporabna na vseh postopkovno proizvodnih področjih z

as a primary heat-transfer unit. Such a basic assembly can be seen in Figure 6. Secondary heat transfer is realized via a conventional fluid-cooling system that allows heat flows in and out of the mould cavity's thermodynamic system.

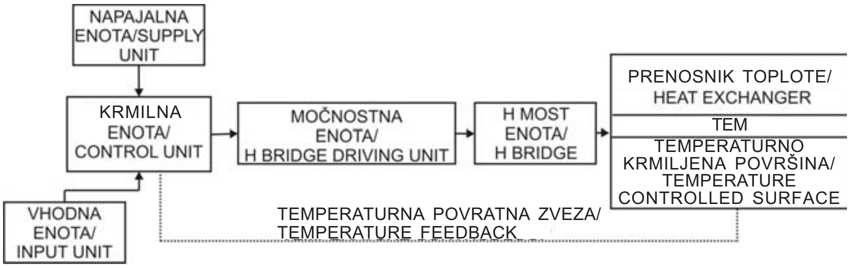
The device presented in Figure 6 is composed of thermoelectric modules (A) that enable primarily heat transfer from or to the temperature-controllable surface of a mould cavity (B). The secondary heat transfer is enabled via cooling channels (C) that deliver constant temperature conditions inside the mould. The thermo-electric modules (A) operate as a heat pump, and as such manipulate with the heat derived to or from the mould by the fluid-cooling system (C). The system for secondary heat manipulation with cooling channels works as a heat exchanger. To reduce the heat capacity of the controllable area thermal insulation (D) is installed between the mould cavity (F) and the mould structure plates (E).

The whole application consists of TEM modules, a temperature sensor and an electronic unit that controls the complete system. The system seen in Figure 7 also comprises an input unit (input interface) and a supply unit (unit for electronic and power electronic supply – H bridge unit).

The input and supply units with the temperature-sensor loop information are attached to a control unit that acts as an execution unit trying to impose predefined temperate/time relations.

Such a system can achieve heating as well as cooling operations. This enables the complete control of processes in terms of temperature and times through the whole cycle. Furthermore, it also allows various temperature/time profiles within the cycle for the starting and ending procedures.

The described technology can be used for various industrial, research and medical purposes where a



Sl. 7. *Shema sistema za hlajenje površin v okviru 0,1 K*
Fig. 7. *Structure for temperature detection and the control of surfaces to within 0.1 K*

zahtevami po natančnem temperaturno-časovnem nadzoru.

Predstavljana izvedba s slike 7 je bila raziskana in preizkušena na dveh ravneh, in sicer v okolju MKE ter dejanskem, prototipnem okolju. Obe okolji sta namenjeni za medsebojni in obojesmerni nadzor rezultatov in boljše razumevanje problemov termodinamike.

4 ANALIZA Z MKE

Sodoben razvoj orodij za brizganje plastike je sestavljen iz več faz. Ena izmed njih je tudi konstrukcija in optimizacija hladilnega sistema. Slednje opravilo pa se izvaja v okviru namenskih paketov MKE (Moldflow, Moldex…[3]), ki uspešno napovedo zmogljivosti hladilnega sistema in njihov vpliv na mehanske lastnosti plastičnih kosov. Z iteracijo optimizacije v okolju MKE se optimira delovni krog in parametri brizganja s popolnim nadzorom nad kakovostjo in reologijo plastičnih izdelkov.

Reološki rezultati simulacij so navadno točni in zanesljivi v primerih natančnih popisov snovnih parametrov. Glede na visoko stopnjo prirejenosti tovrstnih paketov MKE je analiza tehnologij, ki so drugačne kakor sedanji hladilni sistemi, zelo otežena. Zaradi navedenega, se je za potrebe analiz predstavljene izvedbe, uporabilo splošnejše pakete MKE.

4.1 Fizikalni model, analiza z MKE

Uvajanje analiz MKE v razvoj predstavljene tehnologije je bilo izvedeno zaradi izkušenj na področju [3] in možnosti nadzora različnih parametrov znotraj računalniškega okolja. Celoten resnični prototip je bil preslikan v okolje MKE (sl. 8), kjer je

precise temperature/time control is required; in fact in some applications it is already being implemented.

The presented system in Figure 7 needs to be evaluated from the theoretical as well as from the practical point of view. The theoretical aspect was evaluated by the FEM simulations, while the practical one was evaluated by the development of the prototype and real application testing. FEM simulations also served as support control during prototype testing and vice-versa.

4 FEM ANALYSIS

The current development of designing moulds for injection moulding consists of several phases. Among them is the design and optimization of a cooling system. This involves days of simulations using customized FEM packages (Moldflow, Moldex, etc. [3]) that can predict cooling-system capabilities and especially its influence on plastic. With such an iteration in the FEM environment the mould designers gather information on rheological data, deformation and esthetical information due to shrinkage and production time-cycle information.

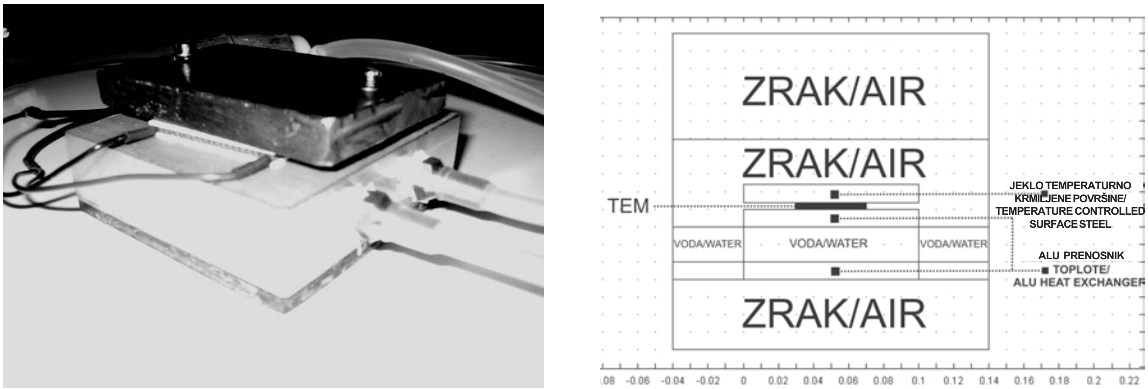
This thermal information is usually accurate but can still be unreliable in cases of insufficient information about the rheological material. Although such FEM packages are frequently used, mould designers cannot depend entirely on the results of simulations. Furthermore, due to the customization of such packages, they are of limited use in our development where more common FEM packages had to be used.

4.1 Physical model, FEM analysis

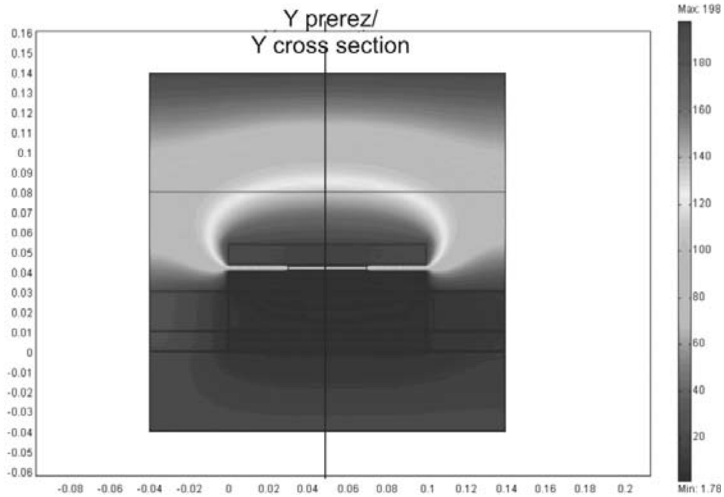
The implementation of FEM analyses into the development project was done as a result of long experiences with such packages [3] and the possibility to perform different tests in a computer environment. The whole prototype cooling system was bila izvedena termodinamična analiza posameznih sklopov prototipa in prestopnih pogojev med njimi. Glede na ozko prirejenost sedanjih paketov MKE in nezmožnost analiziranja različnih prestopnih pogojev med elementi prototipa in popisa TEM, delujočega v režimu toplotne črpalke, je bilo izbrano splošno fizikalno okolje MKE COMSOL Multiphysics. Slednje omogoča hkratno analizo več fizikalnih veličin (toplotni tok, pretok tekočin, elektromagnetne veličine itn.). Termodinamični rezultati prenosa dejanskih lastnosti prototipa v okolje MKE so bili primerjani z dejanskimi rezultati prototipnih preizkusov. S tem je bilo omogočeno optimiranje parametrov simuliranja samega elementa TEM z namenom natančnega popisa celotnega sklopa in pridobitve zanesljivih rezultatov termodinamičnih analiz. Analiza z MKE je potekala z upoštevanjem dveh poglavitnih fizikalnih veličin. Analizirana sta bila dva toplotna vira, in sicer, prevajanje in prestopanje (sevanje je bilo zaradi razmeroma nizke temperature in majhnega vpliva na temperaturo zanemarjeno) za analizo TEM ter prevajanje in prestopanje z vplivom dinamike tekočin za analizo vodnega prenosnika.

Slika 8 predstavlja oba prototipa, v resničnem in prerez prototipa v simulacijskem okolju. Na levi sliki je prikazan resničen prototip, pri katerem je spodnji segment aluminijast vodni prenosnik, sredinski člen je TEM, zgornji jekleni blok pa nadzorovano površino. Enaka sestava je bila simulirana v okolju MKE s poudarkom na pravilno opisanem modelu člena TEM, kar je bilo dose□eno s sinhrono primerjavo preizkusov v dejanskih razmerah. Skratka, model na osnovi MKE je bil popisan in optimiziran z uporabo enakih preizkusov designed in a FEM environment (see Figure 8) through which the temperature distribution in each part of the prototype cooling assembly and the contacts between them were explored. Customized packages for heat-transfer simulation inside moulds were not appropriate due to the limited input designs; in those packages it is not possible to implement a device like a TEM working as a heat pump, and also the contacts seen in Figure 6 cannot be properly described. For simulating physical properties inside a developed prototype a simulation model was constructed using the COMSOL Multiphysics software. This package enables multiphysics modelling, taking into account several physical phenomena (heat transfer, fluid flow, electromagnetic fields, etc). The result was a FEM model identical to the real prototype through which we were able to compare and evaluate the results. The FEM model was explored in terms of heat-transfer physics, taking into account two heat sources: a water exchanger with fluid physics and a thermoelectric module with heat-transfer physics (only conduction and convection were analyzed, radiation was ignored due to the low relative temperature and therefore the low impact on temperature).

Figure 8 shows both prototypes: the real one and a cross-section of the prototype in the FEM environment. In the left-hand figure the lower sections represent the water heat exchanger in aluminium; the middle layer represents the TEM in cooling operation; while the upper layer represents a steel block that requires its temperature to be controlled. The structure in Figure 8 has been explored in detail: the FEM model of the TEM was designed and synchronized with tests of a real TEM, and with those data the heat parameters were gained and the model was tuned up in order to achieve the correct properties.



Sl. 8. *Prototip v resničnem (levo) in okolju MKE (desno)*
Fig. 8. *Prototype in real (left) and FEM environment (right)*



Sl. 9. *Porazdelitev temperature znotraj obravnavanega sistema prototipa glede na analizo z MKE*
Fig. 9. *Temperature distribution according to FEM analysis through the prototype*

v dejanskih razmerah. Rezultat pa je bil zanesljiv simulacijski model, ki se uporablja za optimiranje resničnih uvajanj predstavljene izvedbe v orodjih.

Začetni (okolica modela) in robni pogoji (stalno prisilno stanje vodnega prenosnika) analiz z MKE so bili nastavljeni na 20 °C.

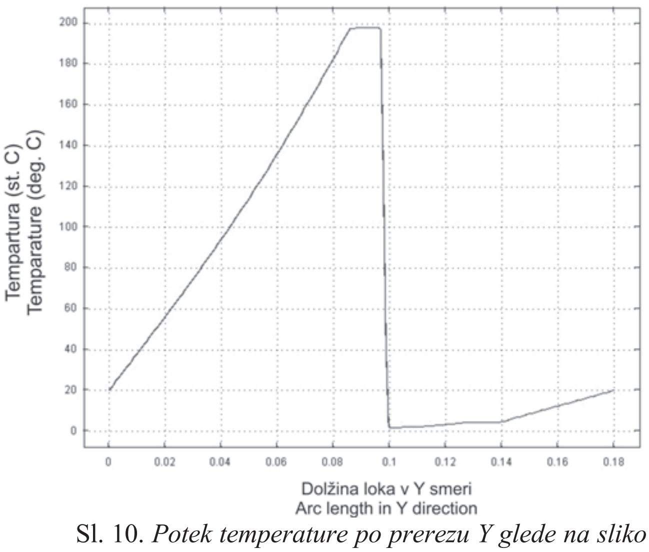
Slika 9 prikazuje temperaturno polje okrog modela po MKE v ustaljenem stanju (končan prehodni pojav) slike 8. Rezultati te simulacije prikazujejo ujem resničnega prototipnega in okolja MKE.

Prav tako pa so bili izvedeni prehodni pojavi, tako v prototipnem (sl. 11) kakor okolju MKE.

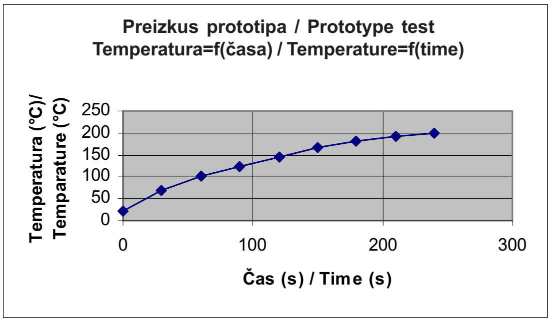
The boundary conditions for the FEM analyses were set so as to achieve identical working conditions as in the real testing. The surrounding air and the water exchanger were set and forced at a stable temperature of 20°C.

The results of the FEM analysis can be seen in Figure 9, i.e., the temperature distribution field through the simulation area shown in Figure 8. Figure 9 represents a steady-state analysis, which was very accurate in comparison to the prototype tests.

Transient FEM analyses were also implemented. The transient results of a real prototype are shown in Figure 11.



Sl. 10. *Potek temperature po prerezu Y glede na sliko 9*
Fig. 10. *Temperature distribution across the Y cross-section according to Fig. 9*



Sl. 11. *Prehodni temperaturni odziv resničnega prototipa s slike 7*
Fig. 11. *Transient temperature response of a real prototype test from Figure 7*

Nazornejši prikaz poteka temperature vzdolžnega prereza Y s slike 9 je ponazorjen s sliko 10.

Vroča stran TEM (najtoplejša točka sklopa) se je segrela na približno 200 °C, medtem ko je bila temperatura hladne strani 5 °C. Tako velike temperaturne razlike na samem TEM se kažejo v velikih mehanskih napetostih, kar privede do mehanskega uničenja TEM. Zaradi tega je treba veliko pozornosti posvetiti mehanskim spojem in pritrditvam TEM kakor tudi inteligentni elektronski zaščiti krmilne enote (glej poglavje 3.2).

Kakor je bilo že omenjeno, so bili opravljeni tudi prototipni preizkusi. Na sliki 11 je prikazan prehodni pojav celotnega sklopa. Ob času 0 je na TEM skočno obremenjen s polno vrednostjo električne moči v delovnem režimu gretja (začetni in robni pogoji so bili postavljeni enako kakor na preizkusu s slike 8).

Bilo je več razlogov za simulacijo z MKE. Prvi leži v preizkusu toplotnih lastnosti posameznih snovi v sklopu in stikov med njimi, medtem ko je drugi temeljen z boljšim razumevanjem pridrode pojava.

Trenutno delo poteka tudi v smeri nadaljnjega optimiranja modela po MKE na podlagi različnih prototipnih preizkusov v dejanskem okolju. Z rezultati vseh preizkusov in optimiranim modelom TEM in spojev bodo rezultati simulacij celotnih sklopov in uporabe pri dejanskih orodjih prepričljivejši.

4.2 Elektronska krmilna enota

Za potrebe krmiljenja hladilnega postopka celotne naprave je bilo treba razviti namensko krmilno enoto, ki bo zmo na vzporednega nadzora

The temperature distribution over the vertical cross section shown in Figure 9 can be seen in Figure 10.

Figure 10 represents the highest temperature, just below 200°C at the TEM surface on the hot side, while the cold side cools to approximately 5°C; both ends of the curve converge to the surrounding temperature of 20°C. Such a high temperature difference in the TEM results in a high mechanical stress, which can lead to a TEM physical break. Those problems are solved by several solutions, such as correct mounting, choosing an appropriate TEM and applying intelligent electronic control (see Section 3.2).

Transient analyses and a prototype test were also performed, as already described. Figure 11 shows the transient response of the temperature at the controllable surface when driving the TEM with a unit pulse (at time 0 s full heating was applied and the starting conditions were set to 20°C).

There were several reasons for using a FEM model simulation. The first reason was to test the thermal properties of different materials and the bonds between them, while the second was to gain a better understanding of the underlying physics.

After getting a satisfactory FEM model with a behaviour that was similar to the real prototype, some fluid tests were made to test different fluids and their heat-transfer abilities. Fine tuning of the FEM model was carried out according to real testing, which is why the FEM analyses showed precise results.

4.2 Electronic control unit

In order to control the cooling process with TEM, the control unit also needs to be developed and adapted to the process needs. The control unit must be able to več različnih TEM po različnih temperaturno-časovnih potekih znotraj orodja za brizganje plastike. Omenjena krmilna nadzorna enota deluje na načelu mehke logike zaradi narave termodinamičnega modela, kratkih odzivnih časov in enostavnosti samega krmiljenja. V primerjavi s PID krmiljenjem se mehko krmiljenje izkaže z večjo odzivnostjo ob enakih zahtevah natančnosti ([7] in [8]). Omenjeno mehko krmiljenje je zagotavljalo veliko natančnost 0,1 K ob izredno hitrih odzivih.

Gonilna enota za močnostno enoto (most H, slika 7) je uporabljala impulzno širinsko modulacijo (IŠM - PWM) z nosilno frekvenco 1kHz. Slednja frekvenca je bila izbrana na osnovi kompromisa med največjo učinkovitostjo TEM (valovitost električnega toka naj ne bi bila večja od 5%), zmanjšanjem izhodnega filtra za glajenje omenjenega toka in zmanjšanjem preklopnih izgub na mostu H.

5 PREGLED RAZVOJNIH FAZ PROJEKTA

Celoten projekt razvoja izvedbe je bil zaradi zapletenosti razdeljen na več faz:

a) Prototipna faza; osnovni preizkusi uvajanja TEM v krožni režim delovanja kot črpalke toplote, izdelava prototipa sklopa;
b) Preizkusi v okolju MKE; izdelava modela TEM in optimiranje le-tega glede na rezultate dejanskega prototipa; preizkusi celotnega sklopa v okolju MKE ter preizkušanje različnih snovi, konstrukcij itn.;
c) Izdelava preizkusnega orodja za brizganje plastike; izdelano je bilo orodje za preizkušanje skrčkov različnih snovi z veliko zahtevnostjo obvladovanja temperatur; hlajenje s TEM je potrdilo velika pričakovanja (sl. 12);

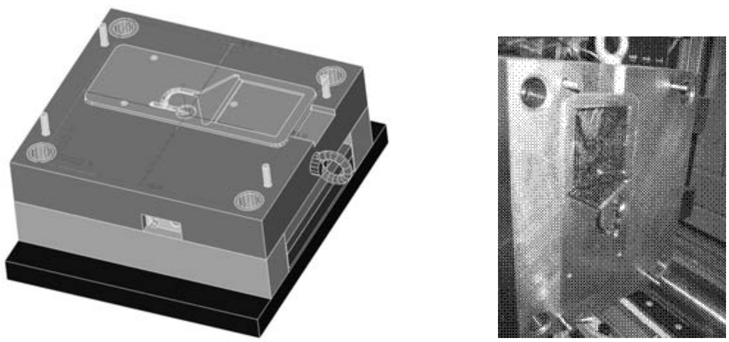
control the temperature on all the needed areas in the mould during the whole injection-moulding cycle. Therefore, the already described electronic unit is based on fuzzy logic due to the nature of the regulation system, short response times and simplicity. Classical PID control was not chosen because the required accuracy with fuzzy logic was gained using shorter control cycle times ([7] and [8]). There are more parameters to be calculated during PID control and, therefore, the required control cycle time is longer and yet there was no need for higher accuracy (below 0.1 K).

The driving unit for the H Bridge seen in Figure 7 was applied with pulse width modulation (PWM) and the modulating frequencies set to 1 kHz. Such frequencies were chosen to ensure the maximum efficiency of the TEM (the ripple in the electric current should not exceed 5%), physically minimize the output filter unit and minimize the switching losses on the H bridge.

5 DEVELOPMENT PHASE REVIEW

The whole development project was, due to its complexity, divided into several phases:

a) Prototyping phase: tests for the implementation of the TEM into cyclic operation in injection moulds were made.
b) Tests of the FEM environment were implemented: data from the FEM analysis were fine tuned according to the tests on the prototype. Results were the control parameters, on the one hand, and materials, construction and data on fluid properties, on the other.
c) With all these gathered data the project went into the final testing phase: a mould for injection moulding for the measurement of default shrinkage for the plastic material was constructed, realized and can be seen in Figure 12.



Sl. 12. *RPN model orodja (levo) in preizkus predstavljene izvedbe (desno)*
Fig. 12. *CAD model of a mould (left) and a test of the presented application (right)*

d) Z vsemi preizkusnimi prototipnimi in industrijskimi podatki je bila narejena primerjalna analiza običajne in predstavljene tehnologije hlajenja orodij za brizganje plastike; ugotovljene so bile izboljšave in omejitve predstavljene izvedbe.

6 SKLEPI

Vnos tehnologije termoelektričnih modulov z neposredno zvezo med električnimi in toplotnimi lastnostmi pomeni pomemben mejnik v razvoju problematike hlajenja orodij. Prenos slednje v zapleteno problematiko hlajenja orodij za brizganje plastike pa prinaša velika pričakovanja pri izdelavi plastičnih izdelkov velikih tehničnih zahtev.

Predstavljena izvedba omogoča popolno obvladovanje zapletenih toplotnih razmer v orodjih za brizganje plastike v smislu nastajanja različnih temperaturnih potekov prek delovnega kroga. Prav tako odpravlja pomanjkljivosti neenakomernega hlajenja z običajno tehnologijo pri izdelavi zelo estetskih izdelkov (površine A). Z možnostjo trenutnega predgretja problematičnih lokacij ob času polnjenja odpravlja problematiko zapolnitve tankih reber. Ena najpomembnejših lastnosti izvedbe (z omenjenim časovno temperaturnim potekom, ki so lahko različni za različne lokacije v orodju oz. TEM) je nadzor in možnost vpliva na velikost in smer toplotnega zvijanja plastičnega izdelka. Poleg vseh naštetih izboljšav pa predstavlja tudi vzpostavitev nadzora nad reološkimi lastnostmi plastičnih izdelkov, predstavljenih v poglavju 2.1.

Trenutno in nadaljnje delo je usmerjeno na tri področja. Prvo je industrializacija in nadaljnja optimizacija izvedbe, predvsem v smeri optimiranja krmilne enote z definiranimi strategijami, omejitvami in lastnostmi posameznih modelov hladilnih sistemov z namenom popolnega termodinamičnega nadzora nad toplotnimi pojavi v orodjih.

Drugi poudarek je na izvedbi popolnoma delujočega in resničnega računalniškega modela celotne izvedbe. Razvoj poteka v dveh smereh, in sicer v smeri razvoja računalniškega modela in simulacij termodinamičnih pojavov znotraj orodja (poglavje 4) ter v smeri razvoja računalniškega modela električnega krmiljenja sistema s slike 7. Oba prek električnih veličin povezana simulacijska modela bosta omogočala popoln vpogled v dejansko stanje, tako glede

d) This application needed precise and stable temperature conditions, and the tested application also proved to be appropriate in an industrial environment. With these tests a proper industrial comparison between the current cooling technology and the solution with TEM were performed, revealing the advantages as well as the limitations of the system.

6 CONCLUSIONS

The use of a thermoelectric module, with its straightforward connection between the input and output relations, represents a milestone in the solution to the cooling problem. Its introduction into moulds for injection moulding, with its problematic cooling construction, and the problematic processing of precise and high-quality plastic parts, represents a difficult challenge.

With the mentioned functionality of a profile temperature driving across the cycle time the injection-moulding process can be fully controlled. Industrial problems (presented in Section 2.1) such as the uniform cooling of problematic A-class surfaces and its consequence on the appearance of the plastic part can be solved. The problems of filling long thin walls can be solved with overheating some surfaces during the injection time. One of the most important application properties is the ability to control and influence the magnitude and direction of the thermal deflection of the plastic part. Furthermore, with such an application control over the rheological properties of plastic materials can be gained.

Current and future work is divided into three areas. The fisrt is industrialization and final tuning of the real application with an emphasis on the control unit where different strategies, limitations and model properties are being optimized in order to get the desired result, i.e., full control over the physical model's thermodynamic behaviour.

The second is focusing on creating a fully independent process in the computer environment. A FEM model of the thermodynamic behaviour of the physical model, i.e., assembly of the mould and TEM together, which we have already accomplished (Section 4) and the process model of the control unit that would represent a complete control unit, shown in Figure 7. Both computer environments will be linked together through real electric parameters – time-dependent electric current and voltage. With such a fully described and customized FEM model termodinamičnih kakor električnih veličin. Tako popisan model bo omogočal računalniško optimiranje termodinamičnih pojavov s predstavljeno izvedbo.

Tretji del razvoja pa zajema razširitev predstavljene izvedbe v ostale preoblikovalne ali nepreoblikovalne industrijske postopke s potrebo po zahtevnih temperaturnih parametrih.

we will be able to promptly adapt and optimize the parameters for diverse cooling systems in industrial practise while performing an optimization in the computer environment.

The third goal of our future work will be directed in the dissemination of the presented application in the processing or non-processing industry environment where heat manipulation is required.

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Naslova avtorjev: Boštjan Žagar
Dr. Blaž Nardin
Andrej Glojek
Tecos, Razvojni center
orodjarstva Slovenije
Kidričeva cesta 25
3000 Celje
bostjan.zagar@tecos.si
blaz.nardin@tecos.si
andrej.glojek@tecos.si

Dr. Dejan Križaj
Univerza v Ljubljani
Fakulteta za elektrotehniko
Tržaška cesta 25
1000 Ljubljana
dejan.krizaj@fe.uni-lj.si

Authors' address: Boštjan □agar
Dr. Bla□ Nardin
Andrej Glojek
Tecos, Slovenian Tool and Die
Development Centre
Kidričeva cesta 25
3000 Celje, Slovenia
bostjan.zagar@tecos.si
blaz.nardin@tecos.si
andrej.glojek@tecos.si

Dr. Dejan Kri□aj
University of Ljubljana
Faculty of electrotehnics
Tr□aška cesta 25
1000 Ljubljana, Slovenia
dejan.krizaj@fe.uni-lj.si

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Nekatere možnosti določitve rezalnih podatkov pri laserskem odrezovanju

Some Possibilities for Determining Cutting Data when using Laser Cutting

Miroslav Radovanović
(University of Niš, Serbia)

Tehnološki problemi, s katerimi se srečujemo na področju uporabe strojev za lasersko odrezovanje, so posledica nezadostnega poznavanja laserske tehnike in pomanjkanja dovolj zanesljivih podatkov iz prakse in poznavanja parametrov, ki vplivajo na sam postopek odrezovanja. Pomemben parameter, ki ga moramo določiti in vključiti v programsko opremo številskega krmiljenja, je rezalna hitrost. Različni avtorji analizirajo postopek lasreskega odrezovanja in podajajo matematične modele, pri katerih je lasersko odrezovanje modelirano z rezalno hitrostjo, kot ključnim parametrom odrezovanja. Številčno vrednost rezalne hitrosti lahko določimo iz priporočil proizvajalca, z določenimi izkušnjami ali z uporabo enačb iz literature. V tem prispevku so predstavljene nekatere možnosti za določitev parametrov laserskega odrezovanja in rezalne hitrosti, na podlagi stopnje odrezovanja, ločilne energije in rabe specifične moči.

(Ključne besede: rezanje lasersko, hitrosti rezanja, parametri rezanja, podatki o rezanju)

The technological problems faced in the field of the application of laser-cutting machines lie in insufficient knowledge of the laser technique and the absence of both sufficiently reliable practical data and knowledge about the parameters affecting the work process itself. A significant parameter that is necessary to determine and to enter in an NC-program is the cutting speed. Various authors analyze the laser-cutting process and give mathematical models where laser cutting is modeled by using the cutting speed as the key cutting parameter. The numerical value of the cutting speed can be determined from the recommendations of manufacturers, a particular experience or with the aid of equations from the literature. This paper presents some possibilities for determining laser-cutting data and cutting speed based on cutting rate, severance energy, and the specific power consumption.

(Keywords: laser cutting, cutting speed, cutting parameters, data analysis)

0 INTRODUCTION

Laser technology is a very young technology, which has not even reached the age of 40 years. For a long time it has been a development searching for industrial applications. For most engineering applications, the laser can be regarded as a device for producing a finely controllable energy beam, which, in contact with a material, generates considerable heat. The heat energy is supplied by the laser beam. The laser beam permits tool-free machining with active heat energy. The energy of the light contained in the laser radiation is absorbed by the workpiece and transformed into thermal energy. The laser beam is becoming a very important engineering tool for cutting, welding and heat treatment.

Laser cutting (especially of mild steel) is a well-known and attractive process for cutting thin sheets. It is one of the most important applications for industrial lasers.

Laser cutting is a thermal, non-contact, mechanized process capable of cutting most materials with a high degree of precision and accuracy. Laser cutting is the process of melting or vaporizing material in a very small, well-defined area. The processes of heating, melting, and evaporation are produced by the laser beam, affecting a workpiece's surface. The desired cut is obtained by moving the laser beam along a given contour. The laser beam is a cutting tool that is able to cut all materials. A laser beam, focused into a very small spot of 0.1 to 0.2 mm in diameter, concentrates thousands of watts. The power density for cutting steels is typically 10^5 to 10^6 MW/m^2. There is no other way to concentrate so much energy into such a small spot. The high power density of the focused laser beam in the spot melts or evaporates material in a fraction of a second. Since our desire is to remove the evaporated and molten material from the affected zone as soon as possible, the laser cutting is performed with a coaxial current of an assisting gas – gas-assisted laser cutting, Fig. 1. When cutting metals, gas-assisted laser cutting is normally applied. In gas-assisted laser cutting, the gas is usually introduced coaxially with the focused laser beam into the cutting area. The gas cools the cut area, thus reducing the heat-affected zone, and also removing the molten dross from the cut.

There are three different methods of cutting with a laser:

- sublimation cutting,
- fusion cutting,
- flame cutting.

In laser-sublimation cutting, the material is removed by evaporating the metal. This requires light densities obtainable by appropriate adjustment of the laser radiation and focusing, so that material removal occurs exclusively by means of evaporation. The metal vapour emitted is blown out of the cut by the cutting-gas jet. Generally, either argon or nitrogen is used in order to avoid oxidation of the cut edge.

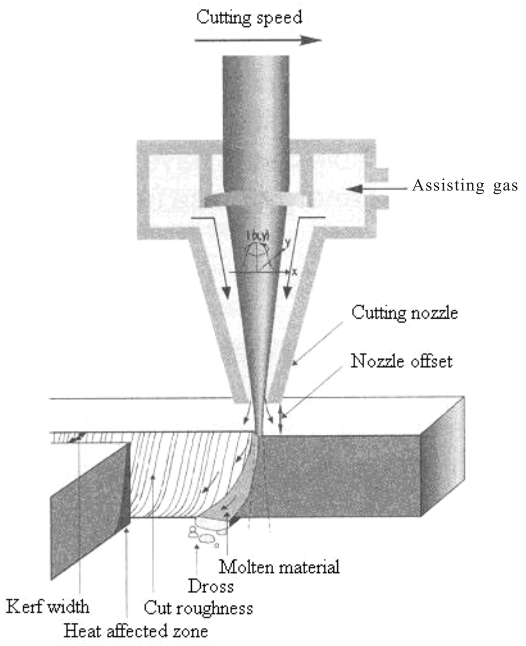

Fig. 1. *Gas assisting laser cutting*

Since almost no melting occurs, smooth cuts can be obtained.

Laser fusion cutting requires less power than laser sublimation cutting as the material is only melted and then blown out of the cut with an inert-gas jet. With laser flame cutting, instead of an inert gas, pure oxygen is used. The material is heated beyond its ignition temperature until a highly exothermic reaction occurs, which supplies the energy for the cutting process. The liquid material is burned using the oxygen and blown downwards out of the cut by part of the assisting-gas jet. Laser-flame cutting and laser-fusion cutting are the predominant methods in the precision cutting of metals.

1 MELTING SPEED, EVAPORATING SPEED AND CUTTING SPEED

If the intensity of the laser-beam radiation is such that the temperature of the workpiece material for a small time attains the temperature of the melt or the evaporation then a small volume of material can be excluded from the cutting zone. After the temperature of melting or evaporation is achieved in the material a frontier layer is formed between the liquid and the hard phase or the gas phase and the liquid/hard phase. With a melt or evaporation of the layer of material thickness Δs the frontier layer moves in depth, while the laser beam makes a hole, i.e., perforates the material of the workpiece with a thickness s. Fig. 2 shows a model for determining the melting speed, the evaporating speed and the cutting speed.

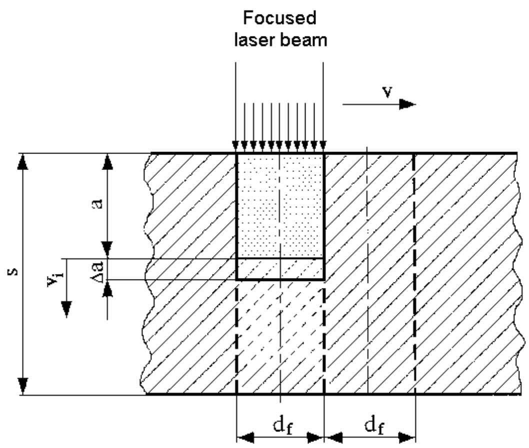

Fig. 2. *Model for the determination of the cutting speed*

The melting speed can be estimated by employing a simple energy-balance arrangement. The principle is based on the fact that the material locally reaches the melt temperature because of the focused radiation.

The energy required to convert a mass m of a material in a cubic element (cylinder with base A_f and height s) at an initial temperature T_0 into a melt can be written with the equation:

$$E_m = m\left[C_S\left(T_m - T_0\right) + L_m\right] \quad (1),$$

where E_m (J) is the energy required to melt the material, m (kg) is the mass of material, C_S (J/kgK) is the solid specific heat capacity, T_m (K) is the melting temperature, T_0 (K) is the ambient temperature, and L_m (J/kg) is the latent heat of melting. We then use the following equation:

$$m = \rho V = \rho A_f s = \rho A_f v_m t = \rho \frac{\pi d_f^2}{4} v_m t \quad (2),$$

where ρ (kg/m³) is the mass density, V (m³) is the cubic element (cylinder with base A_f and height s), A_f (m²) is the area of the laser spot, s (m) is the material thickness, v_m (m/s¹) is the melting speed, t (s) is the time, and d_f (m) is the focal spot diameter. We can put C_S=C, where C (J/kgK) is the specific heat capacity.

Thus, the equation of the energy balance is given by:

$$E_m = \frac{\pi d_f^2}{4} v_m t \rho \left[C\left(T_m - T_0\right) + L_m\right] \quad (3).$$

The power required to melt the material can be determined by the equation $P_m = \alpha P$, where P_m (W) is the power required to melt a material, α is the absorptive index, and P (W) is the laser power.

The absorption varies with the temperature for the metal surface and the CO_2 wavelength. The absorptive index at ambient temperature for steel is 0.1, at the melting temperature it is 0.55, and at the vaporization temperature it is 0.9 ([1] and [2]).

Since $E_m / t = P_m = \alpha P$ we can determine the melting speed in the form:

$$v_m = \frac{4\alpha P}{\pi d_f^2 \rho \left[C\left(T_m - T_0\right) + L_m\right]} \quad (4).$$

The focused laser beam bores into the surface of the material with a velocity v_m, directed into the material. Using Equation (4) we can determine the cutting speed, providing that it does not include the effect of the assisting gas. If the focused laser beam has a diameter d_f and is being scanned at a velocity v, Figure 2, then the beam will traverse a distance equal to its diameter in a time $t = s/v_m$. During this time a cubic element of material V will be removed, and we have $t = s/v_m = d_f/v$.

The equation for determining the cutting speed is:

$$v = v_m \cdot \frac{d_f}{s} \quad (5).$$

The theoretical equation for the cutting speed, in the event that the material is removed by melting, based on a balance of energy, is made with:

$$v = \frac{4\alpha P}{\pi d_f s \rho \left[C\left(T_m - T_0\right) + L_m\right]} \quad (6),$$

where v (m/s) is the cutting speed.

From Equation (6) we can see that the cutting speed is linearly proportional to the laser power and inversely proportional to the sheet thickness. The cutting speed is small for materials with a high melting point and for materials with a high latent heat of melting. The cutting speed is also small for a large focal-spot diameter.

Based on Equation (6) we can determine the necessary energy for melting the material per unit area of the cut as:

$$E_{nm} = \frac{P}{vs} = \frac{\pi d_f \rho \left[C\left(T_m - T_0\right) + L_m\right]}{4\alpha} = const. \quad (7),$$

where E_{nm} (MJ/m²) is the necessary energy for melting the material per unit area of the cut.

The necessary energy for melting the material per unit area of the cut is a constant value for the concrete material of the workpiece and the system for focusing the laser beam.

For mild steel ρ=7860 kg/m³, C=620 J/kgK, T_m=1515°C=1788K, T_0=20°C=293K, and L_m=247·10³ J/kg. For a focal-spot diameter d_f=0.2 mm and the absorptivity index at the melting temperature α=0.55 we have Equation (6) in the form:

$$v = 0.379 \frac{P}{s} \quad (8)$$

and Equation (7) in the form

$$E_{nm} = \frac{P}{vs} = 2.63 \quad (9).$$

Equations (8) and (9) include v(mm/s), P(W), s(mm) and E_{nm} (MJ/m²).

The evaporating speed can be estimated using the equal-energy method. The energy required to convert a mass m of a material cubic element at an initial temperature T_0 into a vapor can be written with the equation:

$$E_v = m\left[C_S\left(T_m - T_0\right) + C_L\left(T_v - T_m\right) + L_m + L_v\right] \quad (10),$$

where E_v (J) is the energy required to evaporate the material, C_L (J/kgK) is the liquid specific heat capacity, T_v (K) is the vaporization temperature, and L_v (J/kg) is the latent heat of vaporization.

Usually, $L_v >> L_m$ and $T_v >> T_0$ and we assume we can put $C_S \approx C_L = C$. Thus, the equation of energy balance (10) is given by:

$$E_v = m\left(CT_v + L_v\right) \quad (11).$$

Since $m = \rho \frac{\pi d_f^2}{4} v_v t$, where v_v (m/s) is the evaporating speed, we have:

$$E_v = \frac{\pi d_f^2}{4} v_v t \rho\left(CT_v + L_v\right) \quad (12).$$

The power required to evaporate the material can be determined by the equation $P_v = \alpha P$, where P_v (W) is the power required to evaporate the material.

Since $E_v / t = P_v = \alpha P$ we can determine the evaporation speed in the form:

$$v_v = \frac{4\alpha P}{\pi d_f^2 \rho\left(CT_v + L_v\right)} \quad (13).$$

A focused laser beam boring into the surface of a material with a velocity v_v directed into the material.

Using Equation (13) we can determine the cutting speed, providing that the effect of the assisting gas is not included.

If the focused laser beam has a diameter d_f and is being scanned at a velocity v, Figure 2, then the beam will traverse a distance equal to its diameter in a time $t = s/v_v$. During this time a cubic element of material V will be removed, and we have $t = s/v_v = d_f/v$.

The equation for determining the cutting speed is:

$$v = v_v \cdot \frac{d_f}{s} \quad (14).$$

The theoretical equation for the cutting speed, in the event that the material is removed by vaporization, based on the balance of energy is carried out with:

$$v = \frac{4\alpha P}{\pi d_f s \rho\left(CT_v + L_v\right)} \quad (15),$$

where v (m/s) is the cutting speed.

Equations (6) and (15) are theoretical, and do not include the influence of the assist gas, especially if it is oxygen. With laser oxygen cutting the oxidation reaction generates heat, which accelerates the cutting process. It contributes approximately 40% of the energy required for the cutting.

The necessary energy for the vaporization of material per unit area of cut is:

$$E_{nv} = \frac{P}{vs} = \frac{\pi d_f \rho\left(CT_v + L_v\right)}{4\alpha} = const. \quad (16),$$

where E_{nv} (MJ/m^2) is the necessary energy for the vaporization of material per unit area of cut.

For mild steel T_v=3160 K and L_v=6.8·10^6 J/kg. For the focal spot diameter d_f=0.2 mm and the absorptivity index at the vaporization temperature α=0.9 we have Equation (15) in the form:

$$v = 0.083 \frac{P}{s} \quad (17).$$

And Equation (16) in the form:

$$E_{nv} = \frac{P}{vs} = 12.08 \quad (18).$$

In Equations (17) and (18) we have v(m/s), P(W), s(mm) and E_{nv} (MJ/m^2).

We can determine the severance energy (the energy necessary for cutting the material per unit area of cut) as:

$$E_s = \frac{P}{vs} \quad (19),$$

where E_s (MJ/m^2) is the severance energy.

The value of the severance energy is essentially a quantitative measure of the difficulty of cutting: the larger the value, the more difficult a material is to cut. It is evident that the severance energy E_s is between the energy necessary for melting the material E_{nm} and the energy necessary for the vaporization of the material E_{nv}:

$$E_{nm} < E_s < E_{nv} \quad (20).$$

The theoretical range of values for the severance energy, E_{st}, for mild steel is:

$$2.63 < E_{st} < 12.08 \quad (21),$$

where E_{st} (MJ/m^2) is the theoretical value of the severance energy.

2 DETERMINATION OF THE CUTTING DATA

The process of laser cutting can be described by the following characteristics: cutting rate, severance energy, and specific power consumption.

The cutting rate is defined by the equation:

$$R = v \cdot s \quad (22),$$

where R is the cutting rate, v is the cutting speed, and s is the material thickness.

The severance energy is defined by the equation:

$$E = \frac{P}{v \cdot s} \quad (23),$$

where E is the severance energy, P is the laser power, v is the cutting speed, and s is the material thickness.

The specific power consumption is defined by the equation:

$$P_s = \frac{P}{Q} \quad (24),$$

where P_s is the specific power consumption, P is the laser power, and Q is the material removal rate.

Tables 1 and 2 contain the numerical values of the cutting rate, the severance energy, and the specific power consumption for the laser oxygen cutting and laser nitrogen cutting of mild steel. The values of the parameters are variable and change with the lens, the focal point stand-off, the nozzle diameter, the nozzle stand-off, and the assisting-gas pressure. All the parameters are the middle values, calculated on the basis of cutting data from manufacturers ([10] and [11]).

Between the cutting rate, R, and the severance energy, E, we have a dependence in the form:

$$R \cdot E = P \quad (25),$$

where P is the laser power.

Between the cutting rate, R, and the specific power consumption, P_s, we have a dependence in the form:

$$R \cdot P_s = \frac{P}{s_r} \quad (26),$$

where P is the laser power, s_r is the cut width.

Between the severance energy, E, and the specific power consumption, P_s, we have a dependence in the form:

Table 1. *Cutting data for laser oxygen cutting*

Characteristics	Laser power		
	1000 W	2000 W	3000 W
Cutting rate R(m^2/s)	$(12÷16)10^{-5}$	$(20÷25)10^{-5}$	$(23÷33)10^{-5}$
Severance energy E(MJ/m^2)	6÷8	8÷10	9÷13
Specific power consumption P_s(GJ/m^3)	21÷33	23÷42	25÷54
Material: mild steel; assisting gas: oxygen			

Table 2. *Cutting data for laser nitrogen cutting*

Characteristics	Laser power		
	1000 W	2000 W	3000 W
Cutting rate R(m^2/s)	$(8.5÷12)10^{-5}$	$(17÷20)10^{-5}$	$(17÷25)10^{-5}$
Severance energy E(MJ/m^2)	8.5÷12	10÷12	12÷18
Specific power consumption P_s(GJ/m^3)	54÷72	60÷72	60÷90
Material: mild steel; assisting gas: nitrogen			

$$\frac{E}{P_s} = s_r \quad (27),$$

where s_r is the cut width.

With these characteristics we can determine the principal parameters of the laser-cutting process.

2.1 Determining the cutting data using the cutting rate

On the basis of the cutting rate, R, we can estimate:

- cutting speed, v:

$$v = \frac{R}{s} \quad (28),$$

- material removal rate, Q:

$$Q = v \cdot s \cdot s_r = R \cdot s_r \quad (29),$$

- time to make the cut, t_g:

$$t_g = \frac{\ell}{v} = \frac{V}{Q} \quad (30),$$

Example 1: Laser, oxygen cutting; laser, 1000W; material of workpiece, mild steel; material thickness, 4mm; and length of cut, 1m.

For the laser of 1000W, from Table 1 we have a middle value of the cutting rate R=14·10^{-5}m^2/s. For the cut width we take s_r=0.25mm,
for the volume of removed material, V=1.0cm^3,
for the cutting speed, v=0.035m/s,
for the material removal rate, Q=3.5·10^{-8}m^3/s,
and for the cutting time, t_g=28.57s.

2.2 Determining the cutting speed with the severance energy

On the basis of the severance energy, E, according to Equation (23), we can estimate the cutting speed v as:

$$v = \frac{P}{E \cdot s} \quad (31).$$

Example 2: Laser, oxygen cutting; laser, 1000W; material of workpiece, mild steel; and material thickness, 4mm.

For a laser of 1000W, from Table 1 we have a middle value of the severance energy E=7MJ/m^2.
Cutting speed, v=0.035m/s

2.3 Determination of the cutting speed by specific power consumption

On the basis of the specific power consumption, P_s, according Equations (24) and (29), we can estimate the cutting speed v as:

$$v = \frac{P}{P_s \cdot s \cdot s_r} \quad (32).$$

Example 3: Laser, oxygen cutting; laser, 1000W; material of workpiece, mild steel, and material thickness, 4mm.

For a laser of 1000W, from Table 1 we have a middle value of the specific power consumption of P_s=27MJ/m^3. For the cut width we take s_r=0.25mm.
Cutting speed, v=0.037m/s

2.4 Determination of the cut width by cutting rate and specific power consumption

On the basis of the cutting rate, R, and the specific power consumption, P_s, we can estimate the cut width s_r as:

$$s_r = \frac{P}{R \cdot P_s} \quad (33).$$

Example 4: Laser, oxygen cutting; laser, 1000W; and material of workpiece, mild steel.

For a laser of 1000W, from Table 1 we have middle values of the cutting rate R=14·10^{-5}m^2/s, and a specific power consumption P_s=27GJ/m^3.
Cut width, s_r=0.26mm

2.5 Determination of the cut width by the severance energy and specific power consumption

On the basis of the severance energy, E, and the specific power consumption, P_s, according to Equations (25) and (26) we can estimate the cut width s_r as:

$$s_r = \frac{E}{P_s} \quad (34).$$

Example 5: Laser, oxygen cutting; laser, 1000W; and material of workpiece, mild steel.

For a laser of 1000W, from Table 1, we have middle values of the severance energy E=7MJ/m^2, and specific power consumption P_s=27GJ/m^3.
Cut width, s_r=0.26mm

3 DETERMINATION OF THE CUTTING SPEED BASED ON EXPERIMENTS

The experiments were performed on a CO_2 laser-cutting machine with computer numerical control. The technical characteristics of the CO_2 laser are: radiation wavelength, 10.6μm; zone of the continual laser power control, 0.2 to 1.3kW; continual work regime; beam divergence less than 4 mrad; mode TEM_{00}; and circular polarization. The optimal laser power is 0.8kW. The focusing system lens is 28mm in diameter and a focal distance of 125mm. The nozzle con opening is 1.6 mm. The material used for the examination is steel Ust 13 (DIN). The work process is carried out using the oxygen process with 99.8% purity.

The experimental equation for the cutting speed is obtained with the aid of a factorial plan of experiments and with ([5] and [8]):

$$v = 0.379 \frac{P^{0.78} h_s^{0.59}}{s^{0.91}} \qquad (35),$$

where v (m/s) is the cutting speed, P (kW) is the laser power, h_s (mm) is the dross height, and s (mm) is the material thickness.

Equation (38) can be rearranged to give:

$$\frac{P}{v} = 2.64 \frac{P^{0.22} s^{0.91}}{h_s^{0.59}} \qquad (36).$$

The experimental range of values of the severance energy is:

$$E_{se} = \frac{P}{vs} = 2.64 \frac{P^{0.22}}{s^{0.09} h_s^{0.59}} \qquad (37),$$

where E_{se} (MJ/m^2) is the experimental value of the severance energy.

In Equations (36) and (37) we have P/v (kJ/m) and P/vs (MJ/m^2).

The tentatively derived Equations (36) and (37) model the interrelation between the cutting speed, the laser power, the material thickness, and the dross height for the laser oxygen-cutting of mild steel.

The experimental range of values of the severance energy for mild steel is:

$$2\ \mathrm{MJ/m^2} < E_{se} < 13\ \mathrm{MJ/m^2} \qquad (38).$$

The value E_{se} includes the influence of the reactive gas on the laser-cutting process.

Fig. 3 and Fig. 4 show *P/v* and *P/(vs)* against *s* for various dross heights, molten and evaporated material. From Figure 3 we can see that the relationship between the laser power/cutting speed *P/v* and the material thickness *s* is linear. The field of cutting is between the line of melting and the line of vaporization. In this case dependence is a function of the dross height. From Figure 4 we can see that the relationship between the laser power/(cutting speed and material thickness) *P/(vs)* and the material thickness *s* is approximately constant. This means that the severance energy E_{se}, or *P/(vs)*, has a constant value for the concrete material of the workpiece. The required severance energy, E_{se}, is between the energy for melting and the energy for the vaporization of the material.

4 CONCLUSION

The technological problems related to the application of the laser-cutting process are an insufficient knowledge of the application of the laser technique as well as an absence of sufficiently reliable practical data and knowledge about the parameters influencing the work process itself. High levels of contour precision at a precisely defined cutting speed have an important effect on the quality of the cut. Therefore, in order to contribute to practical data, this paper gives the results of both the theoretical and the experimental research,

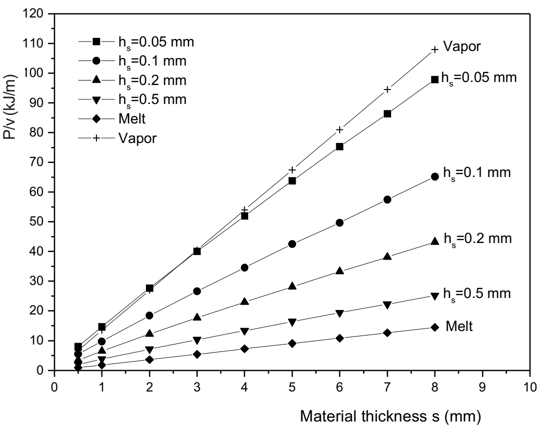

Fig. 3. *P/v vs. s*

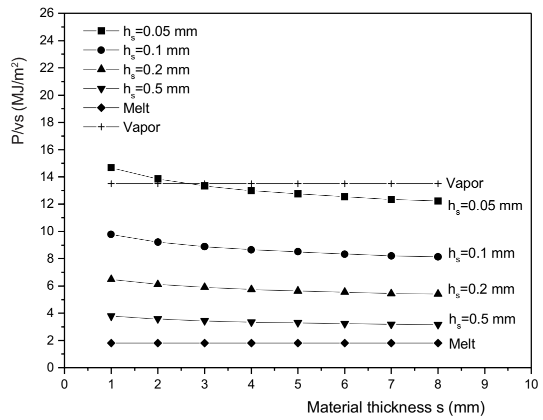

Fig. 4. *P/(vs) vs. s*

referring to the determination of the principal parameters by laser cutting. The cutting data of the laser cutting can be described by the cutting rate, the severance energy, and the specific power consumption. The theoretical range of values for the severance energy is 2.63 MJ/m^2 < E_{st} < 12.08 MJ/m^2, and the experimental range of values is 2 MJ/m^2 < E_{se} < 13 MJ/m^2. The value of E_{se} is essentially a quantitative measure of the cutting difficulty: the larger the value, the more difficult a material is to cut. The value E_{se} includes the influence of the reactive gas on the laser-cutting process. For a determination of the cutting data, as useful initial data it is useful to take the mean values from Table 1 and 2 for the laser cutting of mild steel. Using these values an operator of a laser cutting machine can rapidly estimate the kind of cutting speed for a given thickness of material and the optimal laser power.

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Author's Address: Prof. Dr Miroslav Radovanović
University of Niš
Faculty of Mechanical Engineering
Aleksandra Medvedeva 14
18000 Niš, Serbia
mirado@masfak.ni.ac.yu

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Dinamično vedenje vrtilnega sistema turbine

The Dynamic Behavior of a Turbine Rotating System

Vytautas Barzdaitis[1] - Marijonas Bogdevicius[2]
([1]Kaunas University of Technology, Lithuania; [2]Vilnius Gediminas Technical University, Lithuania)

Prispevek s pomočjo teoretičnega modeliranja in simuliranja obravnava dinamiko vrtilnega sistema turbine, skupaj s spremljanjem eksperimentalnih pogojev in diagnosticiranjem rezultatov. Dva rotoja sta povezana z zobniško sklopko s svežnji prožnih plošč. Med ploščami in polvprijemajočimi zobmi so tanke plasti olja. Med ploščami, ploščnimi svežnji in polvprijemajočimi zobmi lahko nastane prožni stik. Dinamični model vrtilnega sistema z zobniškim sklopom smo oblikovali in simulirali. Simulacijske in preizkusne merilne rezultate vrtilnega sistema smo uporabili za razpoznavo vira nihanj ter napak na prožnih ploščah, ki nastanajo zaradi korozijske obrabe in neporavnanosti rotorjev.

(Ključne besede: turbine parne, sistemi rotorski, dinamika, numerične metode, korozija stična)

This paper looks at the dynamics of a steam-turbine rotating system using theoretical modeling and a simulation combined with experimental condition monitoring and diagnostics results. Two rotors are connected by the toothed wheel coupling with elastic plate packets. There are thin layers of oil between the plates and the semi-couplings' teeth. Elastic contacts can occur between the plates, the plate packets and the semi-couplings' teeth. A dynamic model of the rotating system with the toothed-wheel coupling was designed and simulated. The simulation and experimental measurement results of the rotating system were used to identify the vibration sources and the failures of the elastic plates due to fretting corrosion and the misalignment of the rotors.

(Keywords: steam turbines, rotating systems, dynamics, numerical methods, fretting corrosion)

0 INTRODUCTION

The power generating machines in Lithuania are usually steam turbines that have been in service for about 25 to 40 years. The renovation of these machines results in an increase in both reliability and efficiency. The steam-turbine rotor system (Figure 1) consists of two rotors: a high-pressure cylinder rotor (HPR), a medium- and low-pressure cylinder rotor (MLPR), and a toothed-wheel coupling (TWC). The design of the coupling used in high-power turbo generators (60 MW) is modern but not sufficiently applied in practice. Figure 1 b shows a TWC with 80 teeth connected by plate packets. Each plate packet contains three carbon-steel plates: two of them are 7×45×300 mm and one is 2×45×300 mm. The TWC transmits about 70% of the turbine's torque to the MLPR and to the electric generator.

The rotating system with the hydrodynamic journal bearings and the TWC vibrates during the 3000-rpm rotation speed, and this vibration movement is activated by the toothed-wheel coupling with the plate packets. The dynamic movement of each plate in the packet as well as of all the plate packets is complex, but important for the safe continuous operation of the machine, and it is complicated or even impossible to measure the vibration displacement of the plates and the coupling teeth in situ. The TWC plates inside the packet have to operate under unfavorable conditions: high inertia forces, high temperature, variable lubrication and loading, and relative displacements of the semi-couplings with teeth. Furthermore, during dynamic loading a metallic contact between a plate packet and a tooth may occur. The friction between the plates inside a packet and the teeth cause the fretting corrosion phenomenon that damages the plates. Fretting wear will occur in any material under the conditions of cyclic slip under load [1]. Therefore, the operating conditions of plate packets determine their reliability and, in general, the reliability of the whole rotating system. In this paper the machine-condition monitoring and the diagnostic method is evaluated with modeling and simulation of the dynamics of the rotating system with journal bearings and with the TWC plate packets, both theoretically and experimentally.

The dynamic analysis of a non-linear torsion motion flexible coupling with elastic links is presented in [2]. The results of the analysis of the steady running and transient vibration performance are applied to the determination of the optimum proportions of the couplings. Some influence of the flexible coupling's stiffness on the torsion-motion torque amplitude is considered in [3]. The vibration of the high-power asynchronous electric motor of an air blower is estimated by the periodic monitoring of the absolute vibration of the housing bearings and the relative vibrations of the rotor shaft, and is presented [4]. The insufficient *1X* dynamic stiffness of the rotor system is the main reason for the high vibration amplitude of the *1X* frequency in the electric motor. The dynamics of the rotor is investigated using the finite-element method. The complex finite element of the rotor has twenty-six degrees of freedom. The general dynamic model for a large-scale rotor-bearing system with a cracked shaft is presented in reference [5]. The model accommodates shafts with tapered portions, multiple disks and anisotropic bearings. The dynamic processes in the driver together with the asynchronous engine, coupling with gas, and mechanical drive are considered in [6] to [8]. A coupling of this type consists of separate segments, where additional masses are input. The dynamic model of the driver, the pressure-wave propagation in gas, the interaction of separate coupling bodies and gas are studied. The direct mathematical simulation method of the rotor system with an elastic link is presented [9]. The finite-element method approximates a rotor-bearing system with a finite-degree-of-freedom system, the motions of which are described by ordinary differential equations ([10] to [13]).

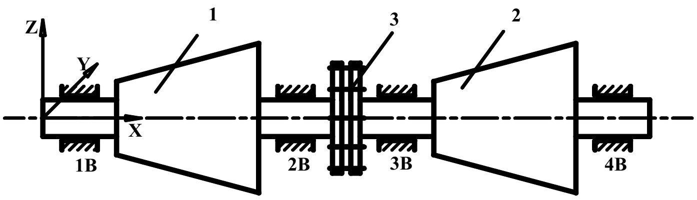
1 THE DYNAMIC MODEL OF STEAM TURBINE ROTORS

The steam-turbine rotating system is shown in Figure 1 and consists of two rotors supported by oil-film bearings and TWC. The following general assumptions were made: the material of the rotors and the coupling are elastic; shear forces are taken into account; the deflection of the rotor is produced by the displacement of points of the centre line; the axial motion of the rotors is neglected; and the semi-couplings are treated as rigid.

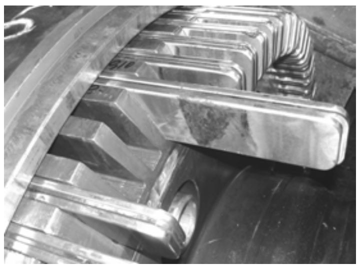
The rotor dynamics is simulated by the finite-element method, where the finite element consists of two nodes and five degrees of freedom (DOF) at each node. The first and the second DOF are displacements along the y and z axes and the last three DOF are angles around the X, Y and Z axes. The vector of translation displacement and the rotation angles of the rotor's finite element can be described as follows:

$$\begin{Bmatrix} v \\ w \end{Bmatrix} = \begin{bmatrix} N_v(\xi) \\ N_w(\xi) \end{bmatrix} \{q(t)\} = [N]\{q\}$$

$$\{\theta\} = \begin{Bmatrix} \alpha \\ \beta \\ \gamma \end{Bmatrix} = \begin{bmatrix} N_{1\theta}(\xi) \\ N_{2\theta}(\xi) \\ N_{3\theta}(\xi) \end{bmatrix} \{q(t)\} = [N_\theta]\{q\} \tag{1}$$



a)



b)

Fig. 1. *Rotors and bearings layout: a – rotating system, 1 – HPR, 2 – MLPR, 3 – TWC, 1B and 3B – radial-axial bearings, 2B and 4B radial bearing; b - view of TWC with plate packets*

where $\{q\}$ is the nodal element displacement vector; $[N]$ and $[N_\theta]$ are the matrices of the shape functions (see appendix A).

Cardin's angles are used to determine the relationship between the angular velocity $\{\dot\theta\}$ and the angular velocity $\{\omega\}$ in the XYZ coordinate system.

The equations of motion of the rotor's finite element are derived by applying a Lagrange equation of the second order, which can be written as follows:

$$\left[M(q)\right]\{\ddot{q}\}+\left(\left[C\right]+\left[G\right]\right)\{\dot{q}\}+\left[K\right]\{q\}=\left\{F(q,\dot{q})\right\} \quad (2),$$

where $[M(q)]$, $[C]$, $[G]$ and $[K]$ are the mass, damping, gyroscopic and stiffness matrices of the finite element, respectively (see appendix B); and $\left\{F(q,\dot{q})\right\}$ is the load vector of the finite element.

2 BEARING MODEL

Under the assumption of small displacements of the journal centre, the fluid-film force components in the horizontal and vertical directions, F_y and F_z, turn out to be as follows:

$$\{F_b\}=\begin{bmatrix}c_{yy} & c_{yz}\\ c_{zy} & czz\end{bmatrix}\begin{Bmatrix}\dot{v}\\ \dot{w}\end{Bmatrix}+\begin{bmatrix}k_{yy} & k_{yz}\\ k_{zy} & k_{zz}\end{bmatrix}\begin{Bmatrix}v\\ w\end{Bmatrix}=[C_b]\begin{Bmatrix}\dot{v}\\ \dot{w}\end{Bmatrix}+[K_b]\begin{Bmatrix}v\\ w\end{Bmatrix} \quad (3),$$

where k_{ij} and c_{ij} $(i,j) = (Y,Z)$ are the stiffness and damping coefficients, respectively (Figure 2).

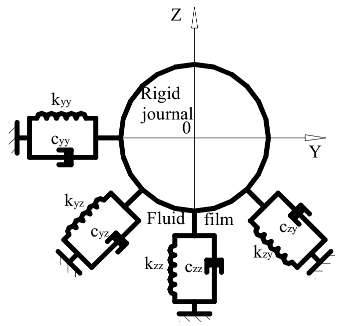


Fig. 2. *Fluid-film bearing model*

3 COUPLING MODEL

The rotors and the bearing system are considered in the global coordinate system XYZ (Figure 1). The semi-couplings are considered as a rigid body attached to the rotors. The mass centers of the first and second semi-couplings are input into the moving coordinate systems $X_1Y_1Z_1$, $X_2Y_2Z_2$, respectively. Each semi-coupling contains notches where a package of three plates is inserted. In each notch of the first and second semi-coupling coordinate systems $X_{1k}Y_{1k}Z_{1k}$ and $X_{2k}Y_{2k}Z_{2k}$ are input, respectively, where $k = 1,2,...,NZ$ and NZ is the number of notches (Figure 3).

The coordinates vector of point P_i in the global coordinate system XYZ in the i^{th} semi-coupling and in the k^{th} notch are given by:

$$\{R_{pi}\}=\{R_{ci0}\}+\{U_{ci}\}+[A_i]\left(\{r_{ik,0}\}+\left[A_3(\gamma_k)\right]\{r_{ik,1}\}\right) \quad (4),$$

where:
- $\{R_{ci0}\}$ is the initial coordinate vector of the mass centre point C_i of the i^{th} semi-coupling;
- $\{U_{ci}\}$ is the vector translation displacements of the point C_i;
- $[A_i]$ is the transformation matrix between the coordinate systems XYZ and $X_iY_iZ_i$;
– γ_k is the angle $\gamma_k = 3\pi/2 + \alpha_k$, $\alpha_k = (k\text{-}1)2\pi/NZ$, $(k=1,2,...,NZ)$;

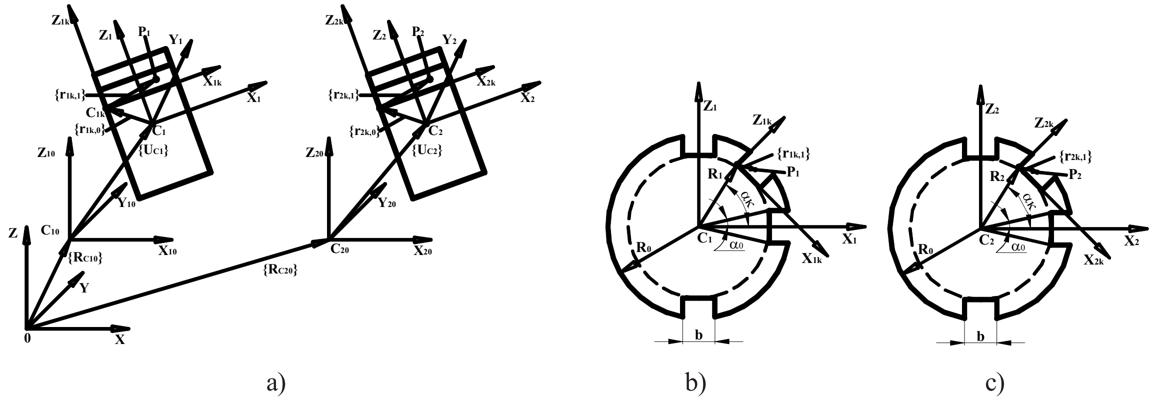


Fig. 3. *The TWC model: a – coupling; b – HPR semi-coupling; c – MLPR semi-coupling*

- $[A_3(\gamma_k)]$ is the transformation matrix between the coordinate system $X_1Y_1Z_1$ and $X_{1k}Y_{1k}Z_{1k}$,
- $\{r_{ik,0}\}$ is the vector that is equal to $\{r_{ik,0}\}$ = $[-\alpha_1 \quad R_1\cos(\alpha_k + \alpha_0) \quad R_1\sin(\alpha_k + \alpha_0)]$; $\sin(\alpha_0) = b/2R_1$; and
- $\{r_{ik,1}\}$ is the vector that determines the coordinates of the point P_i in the coordinate system $X_{ik}Y_{ik}Z_{ik}$, $(i = 1,2)$.

The vector $\{d_{12k}\}$ from point P_1 to P_2 in the k^{th} notch and in the $X_{1k}Y_{1k}Z_{1k}$ coordinate system is equal to:

$$\{d_{12k}\} = [A_3]^T [A_1]^T (\{R_{p2}\} - \{R_{p1}\}) \tag{5}$$

and its first time-derivative is developed:

$$\{\dot{d}_{12k}\} = [A_3]^T [\dot{A}_1]^T (\{R_{p2}\} - \{R_{p1}\}) + [A_3]^T [A_1]^T (\{\dot{R}_{p2}\} - \{\dot{R}_{p1}\}) \tag{6}$$

where $[\dot{A}_i] = [\tilde{\omega}_i][A_i]$;
$\{\dot{R}_{pi}\} = \{\dot{U}_{ci}\} + [\dot{A}_i](\{r_{ik,0}\} + [A_3(\gamma_k)]\{r_{ik,1}\})$;
$[\tilde{\omega}_i]$ is a skew-symmetric matrix associated with the vectors $\{\omega_i\} = [\Omega + \dot{\alpha}_i \quad \dot{\beta}_i \quad \dot{\gamma}_i]$, $(i = 1,2)$, respectively.

The elements of the vector of force acting in the k^th^ notch of the first and the second semi-coupling in the $X_{1k}Y_{1k}Z_{1k}$ coordinate system are given as:

$$F_{2ky} = \begin{cases} -k_y\left(d_{12k}(2) - \delta_y\right), & if \ d_{12k}(2) > \delta_y \\ -c_{y,fluid}\dot{d}_{12k}(2), & if \ -\delta_y \le d_{12k}(2) \le \delta_y \\ -k_y\left(d_{12k}(2) + \delta_y\right), & if \ d_{12k}(2) < -\delta_y \end{cases}$$

$$\begin{gathered} F_{2kx} = -f\left|F_{1ky}\right| sign\left(\dot{d}_{12k}(1)\right) \\ F_{2kz} = -f\left|F_{2ky}\right| sign\left(\dot{d}_{12k}(3)\right) \\ \{F_{2k}\} = \begin{bmatrix} F_{2kx} & F_{2ky} & F_{2kz} \end{bmatrix}^T \\ \{F_{1k}\} = -\{F_{2k}\} \end{gathered} \tag{7}$$

where δ_y is the gap between the k^{th} plate and the semi-coupling; k_y, k_z and $c_{y,\mathrm{fluid}}$ are coefficients of stiffness and damping of the k^{th} plate; f is the friction coefficient between the plate and the semi-coupling.

The following total force and moment vectors acting at points C_i in the global coordinate system are developed as follows:

$$\begin{gathered} \{F_{ci}\} = [A_i]\sum_{k=1}^{NZ}[A_3(\gamma_k)]\{F_{ik}\} \\ \{M_{ci}\} = [A_i]\sum_{k=1}^{NZ}[\tilde{r}_{k,ci,pi}][A_3(\gamma_k)]\{F_{ik}\} \end{gathered} \tag{8}$$

where $[\tilde{r}_{k,ci,pi}]$ is a skew-symmetric matrix associated with vectors:

$$\{r_{k,ci,pi}\} = \{r_{ik,0}\} + [A_3(\gamma_k)]\{r_{ik,1}\}$$

$$(i = 1,2)$$

The equations of motion of the coupling can be presented by the matrix equation:

$$[M_{coupl}(q_{coupl})]\{\ddot{q}_{coupl}\} + [G_{coupl}]\{\dot{q}_{coupl}\} = \{F_{coupl}(q_{coupl}, \dot{q}_{coupl})\} \tag{9}$$

where $[M_{\mathrm{coupl}}(q_{\mathrm{coupl}})]$, $[G_{\mathrm{coupl}}]$ and $[K_{\mathrm{coupl}}]$ are mass and gyroscopic matrices of the coupling, respectively (see appendix C); and $\{F_{coupl}(q_{coupl}, \dot{q}_{coupl})\}$ is the load vector of the coupling.

4 THE SOLUTION OF THE DYNAMIC EQUILIBRIUM EQUATION

The dynamic equilibrium equation for the structure is written as follows:

$$[M(q)]\{\ddot{q}\} + [C]\{\dot{q}\} + [K]\{q\} = \{P(t)\} + \{F(t, q, \dot{q})\} \tag{10}$$

where $[M(q)]$, $[C]$ and $[K]$ are the mass, damping and stiffness matrices; $\{P(t)\}$ is an externally applied load vector; $\{F(t, q, \dot{q})\}$ is a non-linear force vector; and $\{q\}, \{\dot{q}\}$ and $\{\ddot{q}\}$ are the displacement, velocity and acceleration vectors of the finite-element assemblage. In an implicit time-integration scheme, the equilibrium of the system (10) is considered at time $t+\Delta t$ to obtain the solution at time $t+\Delta t$. Iteration will be performed in the non-linear analysis. Using the Newton-Raphson iteration method, the main equilibrium equations turned out to be as follows:

$$\begin{gathered} [M]_{t+\Delta t, i-1}\{\ddot{q}\}_{t+\Delta t, i} + [C]_{t+\Delta t, i-1}\{\dot{q}\}_{t+\Delta t, i} \\ + \left([K]_{t+\Delta t, i-1} - [J]_{t+\Delta t, i-1}\right)\{\Delta q\}_i = \\ \{P\}_{t+\Delta t} + \{F\}_{t+\Delta t, i-1} \end{gathered} \tag{11}$$

$$\{q\}_{t+\Delta t, i} = \{q\}_{t+\Delta t, i-1} + \{\Delta q\}_i$$

where $[J]_{t+\Delta t,i-1}$ is Jacobian matrix, $[J]_{t+\Delta t,i-1} = [\partial\{F\}/\partial\{q\}_T]$.

In the Newmark-Beta time-integration scheme, the following assumptions are employed ([4] and [14]):

$$\begin{gathered} \{q\}_{t+\Delta t} = \{q\}_t + \frac{\Delta t}{2}\left(\{\dot{q}\}_t + \{\dot{q}\}_{t+\Delta t}\right) \\ \{\dot{q}\}_{t+\Delta t} = \{\dot{q}\}_t + \frac{\Delta t}{2}\left(\{\ddot{q}\}_t + \{\ddot{q}\}_{t+\Delta t}\right) \end{gathered} \tag{12}$$

The application of the relations of the equations (12) results in:

$$\{\ddot{q}\}_{t+\Delta t, i} = \frac{4}{\Delta t^2}\left(\{q\}_{t+\Delta t, i-1} - \{q\}_t + \{\Delta q\}_i\right) - \frac{4}{\Delta t}\{\dot{q}\}_t - \{\ddot{q}\}_t \tag{13}$$

and substituting into Equation (11) yields:

$$\left(\frac{4}{\Delta t^2}[M]_{t+\Delta t,i-1}+\frac{2}{\Delta t}[C]+[K]-[J]_{t+\Delta t,i-1}\right)\{\Delta q\}_i=\{P\}_{t+\Delta t}+$$
$$\{F\}_{t+\Delta t,i-1}-[M]\left(\frac{4}{\Delta t^2}\left(\{q\}_{t+\Delta t,i-1}-\{q\}_t\right)-\frac{4}{\Delta t}\{\dot{q}\}_t-\{\ddot{q}\}_t\right)-$$
$$[C]\left(\frac{2}{\Delta t}\left(\{q\}_{t+\Delta t,i-1}-\{q\}_t\right)-\{\dot{q}\}_t\right) \qquad (14).$$

The selection of an appropriate time step Δt is important for the accuracy of the simulation results. A time step of 10^{-5} s was selected to be used in the integration process.

5 RESULTS AND DISCUSSION

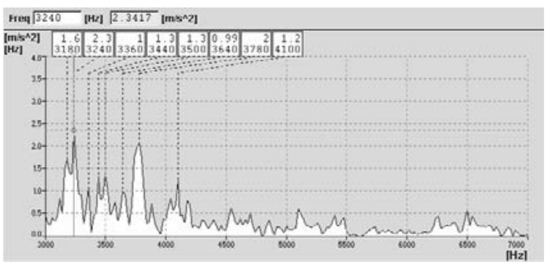
The values of the stiffness coefficients of the bearings are as follows: k_{yy} = 100.0×10^6 N/m, k_{yz} = 50.0×10^6 N/m, k_{zy} = –20.0×10^6 N/m, k_{zz} = 300.0×10^6 N/m, $c_{yy} = c_{zz}$ = 50.0×10^3 Ns/m, and $c_{yy} = c_{zz} = 0$. The steel's density and the elastic modulus are ρ = 7850 kg/m^3 and E = 210×10^9 N/m^2, respectively. The gap between the plate packet and the semi-coupling is equal to δ = 90×10^{-6} m. The coefficients of stiffness and the damping of the plate packet are k_y = 7.47×10^9 N/m, k_z = 1.050×10^9 N/m, c_y = 5.0×10^3 Ns/m, respectively, and the friction coefficient between the plate and the semi-coupling teeth is equal to f = 0.10. The radii of the coupling are R_0 = 0.424 m, and $R_1 = R_2$ = 0.374 m. The geometrical parameters of the coupling are $a_1 = a_2$ = 0.035 m, and b = 15.180×10^{-3} m. The mass, polar and transverse moments of the semi-coupling are $m_1 = m_2$ = 241 kg, J_{cp} = 2.15×10^{-3} m^4, and J_{cd} = 1.075×10^{-3} m^4. The journal diameter is equal to 0.300 m with the length 0.300 m; the bearing radial clearance "lemon type" is ~350×10^{-6} m. The turbines transferred to a 40 MW power load. The unbalances of the rotors are negligible.

The vibration caused by the TWC falls in the high-frequency region from 3000 Hz up to 7000 Hz, as shown in Fig. 4. Experimental results indicated that a more accurate parameter to evaluate the technical condition of the TWC is the 2nd bearing's absolute vibration acceleration in comparison with the 1st and 3rd bearings' vibration data [15]. The prevailing vibration acceleration amplitudes at ~4000 Hz frequency indicate that such a high-frequency vibration is caused by TWC plate packets meshing with teeth as in gear drivers (80 packets × 50 Hz = 4000 Hz).

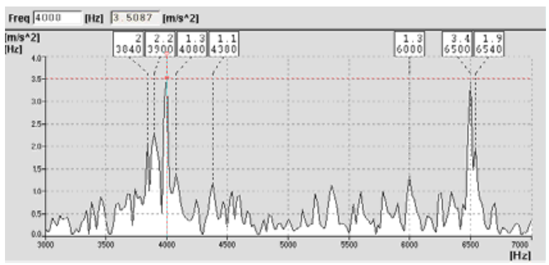
The vibration acceleration spectra acquired from the experimental monitoring of the 2nd bearing indicates a significant difference in the vibration amplitudes at frequencies of 3180 to 4100 Hz when the TWC runs with new plates (a), in comparison with the vibration intensity of the TWC with used damaged plates (b), as shown in Fig. 4. The TWC with damaged plates provides not only high-frequency (~4000 Hz) acceleration amplitudes but 6000 to 6500 Hz vibration accelerations. These data approved the diagnostics concept of the TWC condition evaluation with high-frequency acceleration monitoring of the 2nd bearing. The surfaces of the plates that are in contact with the MLPR were heavily damaged in comparison with the driven sides of the plates that are in contact with the HPR.

If the load is increased up to 53 MW, the vibration intensity of these frequencies slightly decreases. It can be interpreted as a more uniform transmission motion of the rotating system torque from the driving part of the coupling via the plate packets and teeth to the driven part of the coupling, since larger deformations of the plates in the TWC distribute the load among the plate packets more uniformly.

The experimental study of the kinetic orbits of the 2nd and 3rd bearing shafts indicates that the maximum value of the 2nd shaft displacement from the time-integrated mean position "zero"



a - new plates



b - used and damaged plates

Fig.4. *2nd bearing's vertical vibration accelerations spectra with TWC new plates (a) and with plates used in the exploitation and finally damaged (b)*

s_{max2}=158 µm is three times larger in comparison with the 3rd shaft s_{max3}=55 µm, as shown in Fig.5. It is impossible to measure the semi-couplings' orbits, but the HPR semi-coupling orbit repeats the 2nd shaft's motion as the MLPR semi-coupling orbit, i.e., the 3rd shaft.

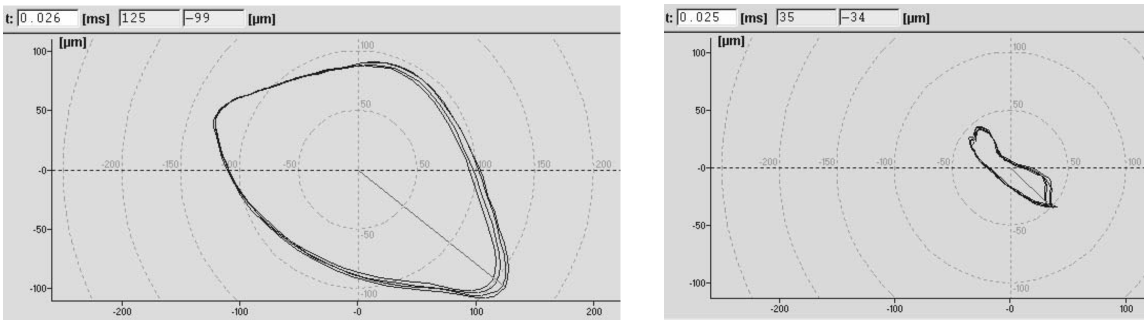
The data of the average position changes of the HPR and MLPR bearing's shafts is presented in Table 1. The changes in the HPR 3rd bearing shaft's average position in the bearing during 10 months in operation indicates the significant changes in the static radial loads acting on the MLPR. The 3rd bearing shaft position's changes in the fluid-film bearing versus time indicate a misalignment malfunction of the HPR-MLPR. The heavy damage to the plate's surfaces took place in the contact area with MLPR semi-coupling teeth (Fig. 1b), but not with HPR semi-coupling teeth. The HPR with the semi-coupling and the plate packets moves in a radial direction relative to the LMPR semi-coupling teeth. This motion provides additional friction between the plates and the LMPR teeth.

The theoretical simulation results of the rotating system model confirmed that the causality of the plate's damage was only slightly involved in the vibration displacement of the plates in the packets that caused the fretting corrosion of the plates. The simulated vibration displacements under a 40 MW power load during four full rotations indicated that the HPR and MLPR semi-couplings' vibration displacements in the *XYZ* coordinate system reached 140 µm in the *Y* (Fig.6) and *Z* directions.

Simulated vibration displacements of the first plate packet during a full four rotations in the $X_{1k}Y_{1k}Z_{1k}$ coordinate system at a 40 MW power load is shown in Fig. 7, and comprises maximum peak-peak values of s_{ppmax}=150 µm. The contact between the plates and the plate packet with the semi-coupling tooth is not constant, and the oil pressure in this gap is variable. Such an operating condition of the plate packet is caused by the relative motion of the mated contact surfaces and results in contact failures and fretting corrosion and increased damage to the plates. Due to continuous long-term operation, heavy damage to the plates is caused by the eccentric motion of HPR-MLPR due to the misalignment.

6 CONCLUSIONS

1. The designed theoretical model of the whole rotating system is based on the finite-element method and simulates the dynamics parameters of elements, and the vibration displacements of



2nd bearing's shaft, s_{max2}=158 µm 3rd bearing's shaft, s_{max3}=55 µm

Fig. 5. *The kinetic orbits of 2nd and 3rd bearings' shafts measured with proximity probes*

Table 1. *The average shaft positions in the bearings during 10 months of steady-state operation*

Machine Operation conditions	HPR bearings shafts' gaps, in µm				MLPR bearings shafts' gaps, in µm			
	1st bearing		2nd bearing		3rd bearing		4th bearing	
	XY displacement directions, when rotating system runs in CW direction							
	-1X	+1Y	-2X	+2Y	-3X	+3Y	-4X	+4Y
After overhaul	-135	+268	-151	+375	-276	+271	-130	+407
After 10 months	-195	+365	-167	+399	-349	+203	-143	+423

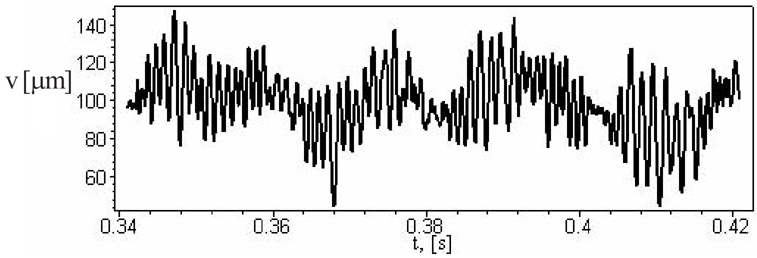


Fig. 6. *Vibration displacement $s_y(t)$ plot of the HPR semi-coupling during a full four rotations in the XYZ coordinate system under a 40 MW power load in the horizontal Y direction*

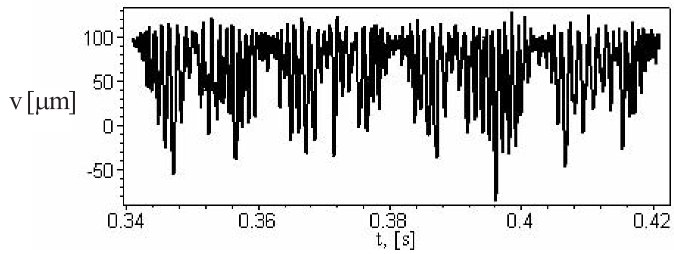


Fig. 7. *Vibration displacement plot of the first plate packet in the $X_{1k}Y_{1k}Z_{1k}$ coordinate system in the Y_{1k} direction*

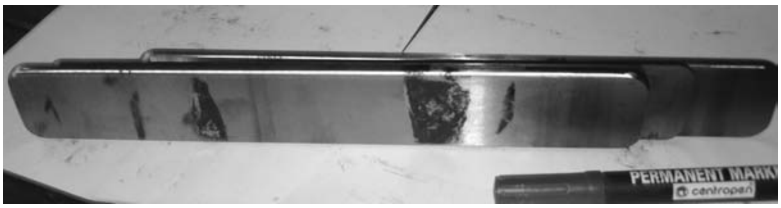


Fig. 8. *Photograph of the damaged plate of the TWC caused by the fretting corrosion and misalignment of the HPR - LMPR*

the tooth-wheel coupling elements that are impossible to measure in-situ on the running machine. The gyroscopic effect due to the rotor's spinning motion and the dynamic properties of the oil-film bearings were introduced in the model.

2. Theoretical and experimental research results identify the causality that provides heavy damage to the TWC plates: the vibration displacements of the rotors, plates and plate packets, and the misalignment of the HPR due to the MLPR.
3. The results indicated that an accurate parameter to evaluate the technical condition of the toothed-wheel coupling is the HPR 2nd bearing's absolute high-frequency vibration acceleration.

APPENDIX A

Individual shape functions given by Ref. [10] are:

$$N_1 = \frac{1}{1+\Phi}\left(1 - 3\xi^2 + 2\xi^3 + \frac{\Phi}{2}(1-\xi)\right)$$

$$N_2 = \frac{L}{1+\Phi}\left(\xi - 2\xi^2 + \xi^3 + \frac{\Phi}{2}\xi(1-\xi)\right)$$

$$N_3 = \frac{1}{1+\Phi}\left(3\xi^2 - 2\xi^3 + \Phi\xi\right)$$

$$N_4 = \frac{L}{1+\Phi}\left(-\xi^2 + \xi^3 + \frac{\Phi}{2}\xi(\xi-1)\right)$$

$$N_5 = 1-\xi; \quad N_6 = \xi$$

$$N_7 = \frac{-6\xi}{L(1+\Phi)}(1-\xi); \quad N_8 = \frac{1}{1+\Phi}\left(1 - 4\xi + 3\xi^2 + \Phi(1-\xi)\right)$$

$$N_9 = \frac{6\xi}{L(1+\Phi)}(1-\xi); \quad N_{10} = \frac{1}{1+\Phi}\left(-2\xi + 3\xi^2 + \Phi\xi\right)$$

$$[N] = \begin{bmatrix} N_1 & 0 & 0 & 0 & N_2 & N_3 & 0 & 0 & 0 & N_4 \\ 0 & N_1 & 0 & -N_2 & 0 & 0 & N_3 & 0 & -N_4 & 0 \end{bmatrix}$$

$$[N_\theta] = \begin{bmatrix} 0 & 0 & \frac{dN_5}{d\xi} & 0 & 0 & 0 & 0 & \frac{dN_6}{d\xi} & 0 & 0 \\ 0 & -\frac{dN_7}{d\xi} & 0 & \frac{dN_8}{d\xi} & 0 & 0 & -\frac{dN_9}{d\xi} & 0 & \frac{dN_{10}}{d\xi} & 0 \\ \frac{dN_7}{d\xi} & 0 & 0 & 0 & \frac{dN_8}{d\xi} & \frac{dN_9}{d\xi} & 0 & 0 & 0 & \frac{dN_{10}}{d\xi} \end{bmatrix}$$

Φ - the shear deformation parameter, $\Phi = 12EJ_d/kGAL^2$;
E, G - moduli of elasticity and shear respectively;
A - cross-sectional area;
J_d is the second moment of the cross-sectional area;
k - the shear correction factor depending on the shape of the cross-section;
ξ - the non-dimensional natural coordinate, $\xi = x/L$.

APPENDIX B

The stiffness matrix of the rotor's finite element is:

$$[K]=[K_{11}]+[K_{12}]+[K_{21}]+[K_{22}]+[K_3]$$

$$[K_{11}]=\int_0^L\left[\frac{dN_{1\beta}}{dx}\right]^T EJ_d\left[\frac{dN_{1\beta}}{dx}\right]dx$$

$$[K_{12}]=\int_0^L\left[\frac{dN_{2\beta}}{dx}\right]^T EJ_d\left[\frac{dN_{2\beta}}{dx}\right]dx$$

$$[K_{21}]=\int_0^L\left(\left[\frac{dN_v}{dx}\right]-[N_{2\beta}]\right)^T kGA\left(\left[\frac{dN_v}{dx}\right]-[N_{2\beta}]\right)dx$$

$$[K_{22}]=\int_0^L\left(\left[\frac{dN_w}{dx}\right]+[N_{2\beta}]\right)^T kGA\left(\left[\frac{dN_w}{dx}\right]+[N_{2\beta}]\right)dx$$

$$[K_3]=\int_0^L[N_{1\theta}]^T\, GJ_p[N_{1\theta}]dx$$

The total kinetic energy of the rotor's finite element in short form is:

$$T=\frac{1}{2}\rho LJ_p\Omega^2+\frac{1}{2}\{\dot q\}^T[M]\{\dot q\}+\Omega[P_1]\{\dot q\}-\Omega\{q\}^T[P_2]\{\dot q\}$$

where $[M]$ is the composite mass matrix of the rotor's finite element is:

$$[M]=[M_1]+[M_2]-[M_3(q$$

$$[P_1]=\int_0^1\rho LJ_p[N_{1\theta}]d\xi$$

$$[P_2]=\int_0^1\rho LJ_p[N_{2\theta}]^T[N_{3\theta}]d\xi$$

$$[M_1]=\int_0^1\rho L[N]^T[N]d\xi$$

$$[M_2]=2L\int_0^1[N_\theta]^T[D_1][N_\theta]d\xi$$

$$[M_3(q)]=\int_0^1[N_{1\theta}]^T\left(L\rho J_p\theta_y\right)[N_{3\theta}]d\xi$$

$$[D_1]=diag\begin{bmatrix}\rho J_p & \rho J_d & \rho J_d\end{bmatrix}$$

where A is cross-sectional area; ρ is density of the finite element; J_p, J_d are the polar and transverse moments, respectively; Ω is the constant angular velocity of rotor.

The gyroscopic matrix of the rotor's finite element is:

$$[G]=\Omega\left([P_2]-[P_2]^T\right)$$

APPENDIX C

The mass and gyroscopic matrices of the semi-coupling can be:

$$[M_c]=L_c([D_3]+2\beta[D_6])$$

$$[G_c]=L_c\Omega\left([D_4]-[D_4]^T\right)$$

$$[D_3]=\begin{bmatrix}\begin{bmatrix}\rho A_c & 0\\ 0 & \rho A_c\end{bmatrix} & [0]\\ [0] & [D_1]\end{bmatrix}$$

$$[D_4]=\rho J_{cp}\begin{bmatrix}0&0&0&0&0\\0&0&0&0&0\\0&0&0&0&0\\0&0&0&0&0\\0&0&0&-1&0\end{bmatrix}$$

$$[D_2]=[D_0]^T[D_1][D_0]$$

$$[D_0]=\begin{bmatrix}1&0&-\beta\\0&1&\alpha\\0&-\alpha&1\end{bmatrix}$$

$$[D_1]=\begin{bmatrix}\rho J_{cp}&0&0\\0&\rho J_{cd}&0\\0&0&\rho J_{cd}\end{bmatrix}$$

$$[D_6]=\rho J_{cp}\begin{bmatrix}0&0&0&0&0\\0&0&0&0&0\\0&0&0&0&-1\\0&0&0&0&0\\0&0&0&0&0\end{bmatrix}$$

where A_c, J_{cp} are the cross-sectional area, polar and transverse inertia moments of the semi-coupling, respectively.

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Authors' Address: Prof. Dr. Vytautas Barzdaitis
Kaunas University of Technology
Faculty of Mechanical and Mechatronics
K. Donelaičio St. 73
LT-44029 Kaunas, Lithuania
vytautas.barzdaitis@ktu.lt

Prof. Dr. Marijonas Bogdevicius
Vilnius Gediminas Technical Univ.
Faculty of Transport Engineering
Plytines St. 27
LT-10105 Vilnius, Lithuania
marius@ti.vtu.lt

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Metoda raziskave nelinearnih upogibov valjev pri rotacijskem ofsetnem tiskarskem stroju

A Method for Investigating the Nonlinear Bends of the Cylinders of a Web Offset Printing Station

Vytautas Kazimieras Augustaitis - Nikolaj Šešok - Igor Iljin
(Vilnius Gediminas Technical University, Lithuania)

Elastični pomiki valjev s plošč, valjev z gumijasto oblogo in tiskovnih valjev ter še posebej upogibov valjev, ki se pojavijo med vrtenjem valjev, pritiskajo drug ob drugega prek tanke elastične gumirane tkanine, ki pokriva valja z gumijasto oblogo (valji so nameščeni tako, da je v vsakem paru valjev vsaj en valj z gumijasto oblogo), močno vplivajo na delovanje rotacijskega ofsetnega tiskarskega stroja. Valji začnejo pritiskati drug ob drugega zaradi pomikanja njihovih ležajnih sklopov.

Prispevek opisuje metodo raziskovanja upogibov valjev in drugih elastičnih pomikov, pa tudi sprememb tlaka vzdolž valjev, ki pritiskajo drug ob drugega. Med raziskavo smo ocenili nelinearne značilnosti elastičnosti sklopov ležajev, sile teže in lege osi vrtenja valjev glede na njihove medsebojne vplive. Obravnavani problem smo rešili kot zahteven nelinearni statični problem, pri katerem smo na robovih delovnih površin valjev, ki pritiskajo drug ob drugega, poznali potrebne vrednosti tlaka, pri tem pa so bili pomiki sklopov ležajev, ki so potrebni za nastanek omenjenih vrednosti tlaka, na začetku neznani.

Z uporabo metode na specifičnem primeru smo pokazali, da so upogibi valjev bistveno vplivali na porazdelitev tlaka vzdolž dotikalnega predela valjev sodobnega tiskarskega stroja in s tem tudi na kakovost tiska.

(Ključne besede: rotacijsko ofsetno tiskanje, stroji tiskarski, valji, upogibanje, metode raziskovalne)

The elastic shifts of the plate, blanket and impression cylinders and, initially, the bends of the cylinders that appear during the rotation of the cylinders that are pressed against each other via the thin elastic cloth (blanket) that covers the blanket cylinders (the cylinders are located in such way that at least one blanket cylinder exists in each pair of cylinders) have a considerable impact on the operation of web offset printing stations. The cylinders are compressed by shifting their bearing assemblies.

The method of investigating the mentioned bends of the cylinders and other elastic shifts as well as the changes of the pressure along the cylinders that are pressed against each other are described. The nonlinear features of the elasticity of the assemblies of the cylinder bearings, the force of gravity and the location of the axes of the rotation of the cylinders with respect to each other are assessed. The problem under discussion is settled as a complex nonlinear static problem, where the required pressures at the edges of the working surfaces of the cylinders that are pressed against each other are provided and the shifts of the assemblies of the cylinder bearings that are required to ensure the said values of the pressures are unknown at the start.

On the application of the method to the specified example, it was shown that the bends of the cylinders noticeably impact on the distribution of the pressure along the contact zone of the cylinders in a modern printing press and on the quality of the prints.

(Keywords: web offset printing, printing press, cylinders, bending, investigation methods)

0 UVOD

V tiskarski industriji je močno razširjena raba rotacijskih ofsetnih strojev. Najpomembnejši del takšnega stroja je njegova tiskarska enota ([1] in [2]).

0 INTRODUCTION

Web offset printing presses are widely used in the printing industry. The most important part of such a press is its printing station ([1] and [2]).

Najpomembnejše komponente tiskarske enote vključujejo valja s ploščo, valja z gumijasto oblogo in (v nekaterih primerih) tiskovna valja, ki sta vpeta v kotalne ležaje v osrednjem delu stroja. Mehanizma za dovajanje barve in vlaženje sta prav tako pomembna, vendar o njiju tu ne bomo govorili. Površini valjev z gumijasto oblogo sta prevlečeni s posebno tanko elastično tesnilko (oblogo), ki ima gladko zunanjo površino. Vsi valji pritiskajo drug ob drugega po celotni dolžini in so v tiskarski enoti nameščeni tako, da ima v vsakem paru priležnih valjev vsaj en valj gumijasto oblogo. Zato pa valji pritiskajo drug ob drugega v smeri deformacije gumijaste obloge. V manjši meri lahko tlak krmilimo s spreminajnjem razdalje med osmi vrtenja valjev. To dosežemo z vrtenjem valjčnih ležajev, ki so montirani v izsrednih pušah. Med delovanjem tiskarske enote, ko se vsi valji (pritisnjeni drug ob drugega) vrtijo brez drsenja, se odtisi prenesejo iz navlaženih tiskarskih predlog, ki sta pritrjeni na valja za ploščo, na površino gumijaste obloge in od tam na neskončni trak, ki se v primeru obojestranskega tiskanja premika med vrtečima se valjema z gumijasto oblogo (ki pritiskata drug ob drugega), v primeru enostranskega tiskanja pa med valjem z gumijasto oblogo in tiskovnim valjem (ki tudi pritiskata drug ob drugega).

Pritisk valjev drug ob drugega prek elastične obloge povzroča absolutne in relativne elastične pomike valjev (te elastične pomike kasneje imenujemo kar pomiki). Ti pomiki sestojijo iz prečnih premih pomikov sklopov valjčnih ležajev, upogibanja valjev in kotnih pomikov (odstopanj) njihovih prerezov.

Tema tega prispevka je iskanje rešitve problema slabše kakovosti delovanja tiskarskega stroja – razvoj metod za računalniško podprto analitično raziskavo upogibanja obloge, plošče in tiskovnih valjev, ki ga povzroča pritiskanje valjev preko obloge. Ti upogibi imajo negativen učinek na kakovost tiskanja, pa tudi na njegovo izrabo in njegove dinamične značilnosti. Upogibi valjev so povezani z njihovimi pomiki in jih moramo zato raziskovati hkrati. Največ pozornosti smo posvetili relativnim upogibom valjev, ker ti povzročajo spremembe v tlaku med upognjenimi valji, takšne spremembe pa neposredno vplivajo na poslabšanje kakovosti tiskov.

Zdaj bomo namen prispevka opisali bolj podrobno. V strokovni literaturi nismo zasledili rešitev zgoraj opisanega problema, a je vendarle

The most important components of the printing station are the plate cylinders, the blanket cylinders and (in some cases) the impression cylinders, which are mounted in rolling bearings in the body of the equipment. The inking and moistening mechanisms are also important, but they are not discussed here. The surfaces of the blanket cylinders are coated with a special thin elastic gasket (blanket), the external surface of which is even. All the cylinders are pressed against each other along their generatrices and are positioned in the station in such a way that for any pair of adjacent cylinders at least one of them is a blanket cylinder. For this reason, all the cylinders are pressed against each other on a deformation of the blanket. The pressure is regulated in a narrow range by changing the distance between the axes of the cylinders' rotation. This is achieved by rotating the bearings of the cylinders, which are mounted in eccentric hubs. During the operation of the printing press, when all the cylinders (pressed against each other) are rotating without sliding, the prints are transferred from the inked printing forms fixed to the plate cylinders to the surface of the blanket of the blanket cylinders, and from there to the paper web that moves between two rotating blanket cylinders (pressed against each other), in the case of double-side printing, and between the blanket cylinder and the impression cylinder (pressed against it) in the case of printing on one side of the paper.

Pressing the cylinders against each other via the elastic blanket causes absolute and relative elastic shifts of the cylinders (such elastic shifts are subsequently referred to as shifts). These shifts consist of transversal rectilinear shifts of the assemblies of cylinder bearings, the bending of the cylinders, and the angular shifts (deviations) of their cross-sections.

The subject of this paper is the solution of the important problem of the quality of a printing station - the development of methods for a computer-aided analytical investigation of the bending of the blanket, the plate and the impression cylinders caused by their pressing against each other via the blanket. These bends adversely affect the quality of the printing as well as its exploitation and dynamic features. The cylinder bends are linked to their other shifts, so they should be investigated together. The most attention is paid to the relative bends of the cylinders, because they cause changes of pressure between the bent cylinders, and such changes directly reduce the quality of the prints.

The aim of the paper is now described in more detail. There have been no recent solutions to the above-mentioned problem described in any technical nujno najti ustrezne rešitve. Na primer, metode ocenjevanja upogibanja valjev so navedene v virih [3] do [5]. Le-te lahko uporabimo pri razlagi vpliva tovrstnega upogibanja na postopek tiskanja, ne pa tudi pri oceni vpliva upogibanja valjev, ki so v posebnih delih tiskarskega stroja.

Da bi razložili metode raziskave upogibanja valjev, navedenih v tem prispevku, in ocenili pomembnost upogibanja, smo uporabili močno razširjen rotacijski ofsetni tiskarski stroj za obojestransko tiskanje [2]; sestava njegovih valjev in shema njegove postavitve sta prikazani na sliki 1.

Obravnavana tiskarska postaja sestoji iz valjev s ploščo 1, 4 (s pritrjenima tiskarskima predlogama, ki tu nista prikazani), in valjev z gumijasto oblogo 2, 3, ki sta prevlečena z oblogama 5. Premeri valjev so označeni z D_d (kjer je d=1, …, 4 – zaporedna številka valja), njihove dolžine pa kot l_1. Valji imajo lahko odprtine z dolžino l_3 in premerom d_d.

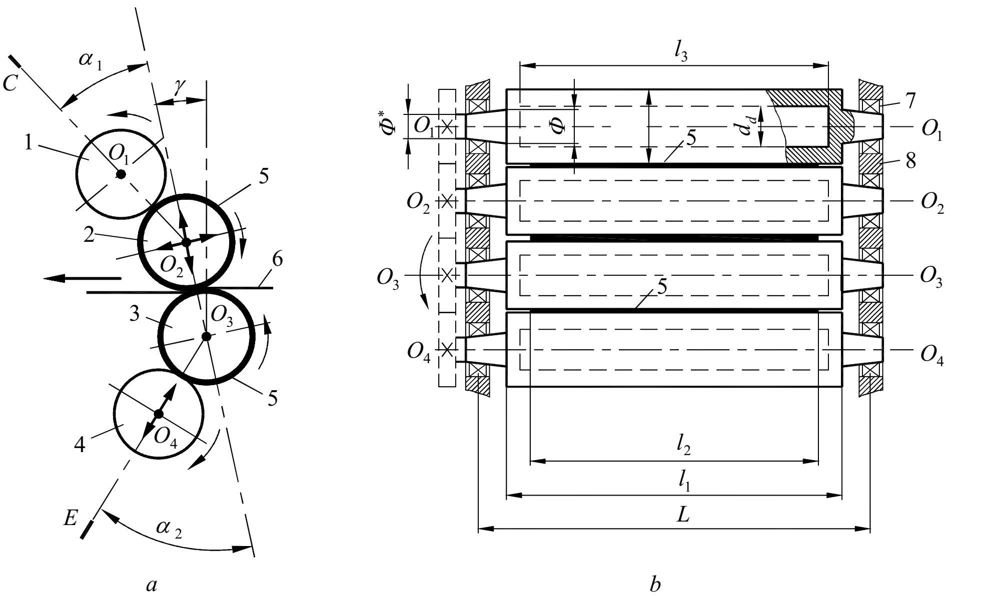
Vsi valji pritiskajo drug ob drugega v smeri deformacije oblog 5 z dolžino l_2; α_1, α_2, γ so koti osi vrtenja valjev. Valji se vrtijo v posebnih ležajih 7 z visoko togostjo in so montirani na stojalo 8 tiskarske enote ter so nameščeni na stožčastih vratovih s premeroma Φ oziroma Φ^*. Stiskanje oblog uravnavamo s

references; however, there is an urgent need to find such solutions. For example, methods of assessing the bending of cylinders are provided in references [3] to [5]. These can be used to explain the impact of such bending on the printing process, but not for assessing the impact of bending cylinders located in specific parts of the printing press on the printing process.

For an explanation of the methods of investigating the cylinder bending described here and an assessment of the importance of this bending, a widely used double-sided web offset printing station [2] was used. The structures of the cylinders and a scheme of the arrangement are shown in Fig. 1

The printing station under discussion consists of two plate cylinders 1, 4 (with the printing forms fixed on them and not shown here) and two blanket cylinders 2, 3, coated with blankets 5. The diameters of the cylinders are marked as D_d (where d=1, …, 4 – the consecutive number of a cylinder) and their lengths – as l_1. The cylinders can have l_3 long holes with the diameter d_d.

When all the cylinders are pressed against each other on a deformation of l_2 along the blankets 5, α_1, α_2, γ are the angles of the axes of rotation of the cylinders. The cylinders roll in special and precise high-rigidity bearings 7 that are mounted in the stand 8 of the printing station and put onto the conical necks of the cylinders with diameters Φ and Φ^*, respectively. The compression



Sl. 1. *Shema postavitve valjev s ploščo in valjev z gumijasto oblogo v rotacijskem ofsetnem stroju za obojestransko tiskanje: a – lege osi vrtenja valjev O_1–O_1,..., O_4–O_4; b – postavitev valjev glede na CO_2O_3E*

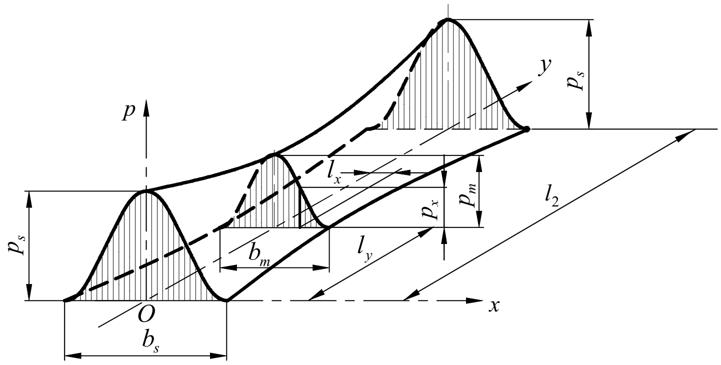
Fig.1. *The scheme of the assembly of the plate and blanket cylinders in the web offset double-side printing station: a – the location of the axes of the rotation of cylinders O_1–O_1,..., O_4–O_4; b – the evolvent of the cylinders according to CO_2O_3E*

premikanjem sklopov ležajev valja 2 v dveh pravokotnih smereh ter premikanjem ležajev valja 4 v eni smeri (na sliki 1 so smeri krmiljenja prikazane s puščicami). Med pritiskanjem se valji (ki jih prek zobnikov obrača električni motor) vrtijo, ne da bi drgnili ob delovne površine, pri tem pa se med valjema z gumijasto oblogo 2 in 3 pomika papirni trak 6. Tiskanje poteka na obeh straneh traku. Delovna površina valja je tisti del površine, ki je prevlečen z oblogo (pri valjih z gumijasto oblogo) in del, ki je v stiku z oblogo (pri valjih za ploščo in tiskovnih valjih).

Ko valji pritiskajo drug ob drugega, se obloga deformira in med valji nastanejo pasovno oblikovani dotikalni predeli. V obravnavanem primeru (slika 1) obstajajo trije dotikalni predeli ($j = 3$): med valjema 1 in 2, 2 in 3 ter 3 in 4. Na katerikoli točki j-tega stičnega predela je stiskanje obloge Δ_j. Največje stiskanje obloge $\Delta_{m,j}$ vzdolž j-tega dotikalnega predela (j=1, 2, 3) in v prečni smeri nastane približno v sredini omenjenega predela. Opazili smo majhen odmik $\Delta_{m,j}$ od sredine dotikalnega pasu in v smeri, ki je nasprotna smeri premih hitrosti površin vrtečih se valjev, ki pritiskajo drug ob drugega. ([4] in [5]). Tukaj, kakor tudi v primerih [3] do [5], tega odmika nismo upoštevali. Vrednost $\Delta_{m,j}$ se spreminja zaradi upogibanja valjev: vrh $(\Delta_{m,j})_{max} = \Delta_{s,j}$ najdemo na robovih delovnih površin valjev, dol pa v sredini (v primeru, ko se valja upogibata v nasprotnih

of the blankets is regulated by a shifting of the assemblies of the bearings of cylinder 2 in two perpendicular directions and the shifting of cylinder 4 in one direction (Fig. 1, the directions of the regulation are shown by arrows). During compression, the cylinders (rotated by electric engine via tooth gears) are rolling without sliding against their working surfaces and draw the paper tape 6 between the blanket cylinders 2 and 3. The prints are made on both sides of the tape. The working surface of a cylinder is the part of its surface that is coated with a blanket (in blanket cylinders) or the part having contact with it (in the plate and impression cylinders).

When pressing the cylinders against each other, the blanket between them is deformed and band-shaped contact zones appear between them. In the case under discussion (Fig. 1), there are three contact zones ($j = 3$): between the cylinders 1 and 2, 2 and 3, 3 and 4. At any point of the j-th contact zone the compression of the blanket is Δ_j. The maximum compression $\Delta_{m,j}$ of the blanket along the j–th band of a zone (j=1, 2, 3) in a radial direction is obtained approximately in the middle of the mentioned band. The small deviation $\Delta_{m,j}$ from the middle of the band and in the direction opposite to the one of the linear speeds of the surfaces of rotating cylinders pressed against each other was observed ([4] and [5]). Here, as in [3] to [5], this deviation is not taken into account. The value of $\Delta_{m,j}$ changes because of the bending of the cylinders: the maximum $(\Delta_{m,j})_{max} = \Delta_{s,j}$ is found at the ends of the working surfaces of the cylinders and the minimum is found in the middle (in the case of bending of the cylinders in opposite directions). The pressure p_j between the cylinders depends



Sl. 2. *Porazdelitev tlaka p v j-tem dotikalnem predelu (indeks "j" smo tu opustili); b_s, p_s – širina dotikalnega predela in tlak na njegovih robovih; b_m, p_m – sta enaka na razdalji l_y od začetka dotikalnega predela; p_x – tlak na katerikoli točki odseka b_m na razdalji l_x od točke največjega tlaka p_m*

Fig. 2. *The distribution of the pressure p in the j-th contact zone of the cylinders (the index "j" has been omitted here); b_s, p_s – the width of the contact zone and the pressure at its ends; b_m, p_m – the same in l_y distance from the beginning of the zone; p_x – pressure in any point of the section b_m in l_x distance from the point of maximum pressure p_m*

smereh). Tlak p_j med valji je odvisen od stiskanja obloge Δ_j. Način porazdelitve tlaka p_j po j-tem dotikalnem predelu je prikazan na sliki 2.

Na podlagi izmerjenih podatkov ([5] in [6]) vemo, da med tlakoma p_j, $p_{m,j}$, napetostima σ_j, $\sigma_{m,j}$ v tlačenem delu obloge in njenima stiskoma Δ_j, $\Delta_{m,j}$ obstaja naslednje razmerje:

on the compression of the blanket Δ_j. The character of the distribution of pressure p_j in the j–th contact zone is shown in Fig. 2.

It is known from experimental data ([5] and [6]) that the following interrelation between the pressures p_j, $p_{m,j}$, and the tensions σ_j, $\sigma_{m,j}$ in the compressed part of the blanket and its compressions Δ_j, $\Delta_{m,j}$ exists:

$$p_j = \sigma_j = E\left(\frac{\Delta_j}{c}\right)^{\eta}; \quad p_{m,j} = \sigma_{m,j} = E\left(\frac{\Delta_{m,j}}{c}\right)^{\eta} \qquad (1).$$

E in c sta Youngov modul in nedeformirano debelino obloge. Nespremenljivi koeficient η = 1,2 do 1,5 za rotacijsko ofsetno tiskanje ([5] in [6]):

Here, E and c are the Young's modulus and the non-deformed thickness of the blanket. The constant coefficient η=1.2 to 1.5 for offset web printing ([5] and [6]):

$$\Delta_j = \Delta_{m,j}\left(1 - \frac{4\,l_{x,j}^2}{b_j^2}\right) \qquad (2)$$

$$\Delta_{m,j} = \frac{b_j^2}{8B_j}; \; b_j = 2\sqrt{2B_j\Delta_{m,j}} \quad (\mathrm{j} = 1, 2, 3) \qquad (3)$$

$$B_1 = \frac{D_1(D+2c)}{2(D_1+D+2c)}; \; B_3 = \frac{D_4(D+2c)}{2(D_4+D+2c)}; \; B_2 = \frac{D+2c}{2}; \; D = D_2 = D_3 \qquad (4).$$

Nespremenljivi koeficiet η in običajni Youngov modul obloge E sta uporabljena tako kakor v virih [4] do [6]. Da bi dobili vrhunski tisk, tlak $p_{m,j}$ ne sme prekoračiti največje $(p_{m,j})^*_{\max}$ in najmanjše $(p_{m,j})^*_{\min}$ dovoljene omejitve. Ti dve omejitvi ustrezata razliki med dovoljenima stiskoma obloge.

The constant coefficient η and the conventional Young's modulus of the blanket E are set as in [4] to [6]. In order to obtain high-quality prints, the pressure $p_{m,j}$ should not overstep the maximum $(p_{m,j})^*_{\max}$ and minimum $(p_{m,j})^*_{\min}$ permissible limits. These limits correspond to the difference between the permissible compressions of the blankets

$$\Delta d_m^* = (\Delta_{m,j})^*_{\max} - (\Delta_{m,j})^*_{\min} \qquad (5).$$

Vrednosti dovoljenih tlakov in stiskov so odvisne od vrste obloge, pričakovane kakovosti tiska in od drugih dejavnikov. Najpogosteje jih dobimo s preizkusi po določitvi naslednjih odvisnosti: tlak med valji in stisk obloge.

Če uporabimo enačbi po [4] in [5], sta η in E [6]:

The values of the permissible pressures and compressions depend on the type of blanket, the required quality of the prints and other factors. Most frequently, they are found in an experimental way after the formation of the following dependences: the pressure between the cylinders and the compression of the blanket.

Using equations from [4] and [5], the expressions for η and E [6] can be written as:

$$\eta = \frac{\lg\left[(p_{m,j})^*_{\max} \big/ (p_{m,j})^*_{\min}\right]}{\lg\left\{\left[(\Delta_{m,j})^*_{\min} + \Delta d_m^*\right] \big/ (\Delta_{m,j})^*_{\min}\right\}} \qquad (6)$$

$$E = c^{\eta}(p_{m,j})^*_{\max} \Big/ \left[(\lambda_{m,j})^*_{\min} + \Delta d_m^*\right] = c^{\eta}(p_{m,j})^*_{\min} \big/ (\Delta_{m,j})^*_{\min} \qquad (7).$$

Iz enačbe (1) izpeljemo naslednje razmerje med $p_{m,j}$ in $\Delta_{m,j}$:

From (1), the interrelation between $p_{m,j}$ and $\Delta_{m,j}$ is found to be:

$$\Delta_{m,j} = \left(\frac{p_{m,j}}{E}\right)^{1/\eta} c \qquad (8).$$

Porazdelitev tlaka p po j-tem dotikalnem predelu je prikazana na sliki 2.

Tlak $p_{m,j}$ je določen kot prvi pogoj. Da bi zagotovili njegovo potrebno vrednost, morajo valji pritiskati drug ob drugega z močjo, ki povzroča potrebno vrednost največjega stiska obloge $\Delta_{m,j}$. Zaželeno bi bilo, da je $p_{m,j}$ = konst, vendar to ni mogoče zaradi upogibanja valjev. Potrebni $p_{m,j}$ (kakor tudi $\Delta_{m,j}$) je lahko povprečna vrednost tlaka $p_{m,j}$ oziroma njegova vrednost na kateremkoli prerezu valjev, na primer, znotraj valja ali na robovih delovnih površin. V tem prispevku predpostavljamo, da sta potrebni največji tlak med valji in ustrezni največji stisk obloge določena na robovih delovnih površin valjev, torej je tlak $p_{s,j}$ stisk obloge $\Delta_{s,j}$.

Stisk obloge povzročajo pomiki sklopov ležajev; ti pomiki se ne ujemajo z $\Delta_{s,j}$, ker med pomikanjem sklopi ležajev in robovi valjev postanejo deformirani. Soodnos med pomiki sklopov ležajev in $\Delta_{s,j}$ za katerokoli dvojico valjev, ki pritiskata drug ob drugega, dobimo z izračunom pomikov vseh valjev tiskarske enote, ko ocenimo nelinearnost odziva elastičnosti obloge.

0.1 Cilj raziskave

Cilj prispevka je razvoj analitičnih, računalniško podprtih matematičnih metod, ki jih lahko uporabimo za izračun in raziskavo zakonitosti deformacij valjev s ploščo, valjev z gumijasto oblogo in tiskovnih valjev (pri rotacijskem tiskarskem stroju), ki pritiskajo drug ob drugega prek obloge, pa tudi tlaka na dotikalnih predelih valjev. Omenjene metode prav tako omogočijo določitev vpliva deformacije valjev na analizo postopka tiskanja in določitev vrednosti pomikov sklopov ležajev, ki povzročijo specifične vrednosti stiska obloge $\Delta_{s,j}$ in tlaka $p_{s,j}$.

Rezultate raziskave lahko uporabimo pri oblikovanju in izboljšavi tiskarskih enot in pri oceni kakovosti novih tiskarskih strojev še preden jih začnemo uporabljati.

Razvijanje omenjenih metod je bilo zahtevno, ker je odnos med stiskalnimi silami valjev in stiskom oblog, ki jih povzročijo valji, nelinearen in ker pomiki ležajev, ki omogočijo potrebno stiskanje oblog $\Delta_{s,j}$, niso vnaprej znani. Vendar pa je nujno potrebno, da ugotovimo

The distribution of the pressure p in the j-th contact zone is shown in Fig. 2.

The pressure $p_{m,j}$ is set in the requirements. In order to ensure the required value of the pressure, the cylinders are pressed against each other with a force that causes the required value of the maximum compression $\Delta_{m,j}$ of the blanket. It is desirable for $p_{m,j}$ = const, but this is impossible because of the bending of the cylinders. The required $p_{m,j}$ (as well as $\Delta_{m,j}$) can be considered as the average value of the pressure $p_{m,j}$ or its value in any cross-section of the cylinders, for example, inside the cylinders or at the edges of their working surfaces. In this paper, it is considered that the required maximum pressure between the cylinders and the corresponding maximum compression of the blanket are determined at the edges of the working surfaces of the cylinders, i.e., they are the pressures $p_{s,j}$ and the blanket's compressions are $\Delta_{s,j}$.

The compression of the blanket is caused by shifts of the assemblies of cylinder bearings; these shifts do not coincide with $\Delta_{s,j}$, because the assemblies of bearings and the ends of the cylinders are deformed during the shifting. The interrelation between the shifts of the assemblies of bearings and $\Delta_{s,j}$ for any pair of cylinders that are pressed against each other is found by calculating the shifts of all the cylinders of the printing station when the non-linearity of the response of the blanket's elasticity is assessed.

0.1 Object of the Paper

The object of the paper is to develop analytical computer-aided calculation methods that are usable for the calculation and investigation of the regularities of deformations of the plate, blanket and impression cylinders (in the web printing press) that are pressed against each other via the blanket as well as the pressure on the contact zones of the cylinders; a determination of the impact of the deformation of the cylinders on an investigation of the printing process; a determination of the values of shifts of the bearings assemblies of such cylinders that cause the set values of blanket compression $\Delta_{s,j}$ and pressure $p_{s,j}$.

The results of the paper may be applied to designing and improving printing stations and an assessment of the quality of the acquired printing presses before starting their exploitation.

The development of the methods was complicated, because the interrelation of the forces of the compression of the cylinders and the compression of the blankets caused by them is non-linear, and the shifts of the assemblies of cylinder bearings that ensure the required compression of the blankets $\Delta_{s,j}$ are not njihove vrednosti, kajti, kakor bomo pokazali v tem prispevku, je prav zaradi tega problema naša raziskava drugačna od običajnih iskanj rešitve sistema nelinearnih enačb.

1 SHEMA IZRAČUNOV

Shema izračunov, ki se nanaša na tiskarsko enoto, predstavljeno na sliki 1, je prikazana na sliki 3. Shema kaže, da so absolutni pomiki valjev izračunani, njihove medsebojne razlike pa lahko uporabimo za določitev relativnih pomikov valjev in s tem vrednosti $\Delta_{m,j}$, $\Delta_{s,j}$, $p_{m,j}$, $p_{s,j}$, ki nas najbolj zanimajo.

V shemi izračunov je bila elastičnost obloge in ležajev ocenjena z uporabo diskretnih elastičnih elementov (njihovi odzivi so opisani z ustreznimi koeficienti togosti) in z uporabo metode končnih elementov, ki izloči pomike valjev. Analogne sheme izračunov smo oblikovali tudi za sklope valjev, ki so drugače razporejeni in so sestavljeni iz drugačnega števila posameznih valjev.

Valji so razstavljeni na ploščate končne elemente, ki so oblikovani kot valji in prisekani stožci (slika 3, c, d). Z uporabo enačb smo ocenili upogibe in druge pomike valjev v ravninah, ki potekajo skozi naslednje teoretične osi vrtenja nedeformiranih valjev (te osi kasneje imenujemo osi vrtenja valjev): valj 1 – ravnina osi vrtenja valja 1 in valja 2 je skladna z osjo koordinat x_1; valj 4 – ravnina osi vrtenja valjev 4 in 3 je skladna z osema koordinat x_2, x_3 in dve ravnini, ki ležita pravokotno na to ravnino – ena gre zkozi os vrtenja valja 2, druga pa skozi os vrtenja valja 3 (prva ravnina ima os koordinat y_2 in druga ima os koordinat y_3).

Tako so absolutni pomiki valja 1, in tudi valja 4 ocenjeni v eni ravnini (pomiki valja 1 v smeri osi koordinat x_1 in pomiki valja 4 v smeri osi x_4), pomiki valjev 2 in 3 pa v dveh pravokotnih ravninah (pomiki valja 2 v smeri osi koordinat x_2, y_2 in pomiki valja 3 v smeri osi koordinat x_3, y_3).

Predpostavljamo, da ima tiskarska enota simetrično ravnino, ki poteka skozi središčne točke valjev (točke O_1,...,O_4) in leži pravokotno na njihove osi vrtenja (sl. 3, b). Zato so vse vrednosti, ki opisujejo valje, njihove pomike in sile, ki delujejo na valje v obeh polovicah valjev, ki jih deli ravnina, enake.

known in advance. It is necessary to find them and this, as we will show here, causes a difference in the problem under discussion from the usual problem of the solution of a system of non-linear equations.

1 THE SCHEME OF CALCULATIONS

The scheme of calculations, corresponding to the printing station, presented in Fig. 1, is shown in Fig. 3. According to this scheme the absolute shifts of the cylinders are calculated and the corresponding differences are used to find the relative shifts of the cylinders and the values $\Delta_{m,j}$, $\Delta_{s,j}$, $p_{m,j}$, $p_{s,j}$ that are our main interest.

In the scheme of the calculations the elasticity of the blanket and the cylinder bearings was assessed by using discrete elastic elements (their responses are described with the corresponding coefficients of stiffness) and by applying the use of a method of finite elements that discretized shifts of the cylinders. Analogous schemes of the calculation are formed for other layouts and numbers of cylinders.

The cylinders are disintegrated to plane finite elements shaped as cylinders and truncated cones (Fig. 3, c, d). Using the equations, the bends and other shifts of the cylinders are assessed in planes, passing through the following theoretical axes of rotation of non-deformed cylinders (these axes are subsequently referred to as the axes of rotation of the cylinders): cylinder 1 – the plane of the axes of rotation of it and cylinder 2 according to the axis of the coordinates x_1; cylinder 4 - the plane of the axes of rotation of cylinders 4 and 3 according to the axes of the coordinates x_2, x_3 and in two planes perpendicular to this plane – one of them goes through the axis of rotation of cylinder 2 and the other - through the axis of rotation of cylinder 3 (the first plane includes the axis of coordinates y_2 and the second plane – the axis of coordinates y_3).

Thus, the absolute shifts of cylinder 1, like that of cylinder 4, are assessed in one plane (the shifts of cylinder 1 in the direction of the axis of coordinates x_1, and the shifts of cylinder 4 in the direction of the axis x_4) and the shifts of cylinders 2 and 3, in two perpendicular planes (the shifts of cylinder 2 in the direction of the axes of coordinates x_2, y_2 and the shifts of cylinder 3 in the direction of the axes of coordinates x_3, y_3).

It is considered that the printing station has a plane of symmetry that passes through the middle points of the cylinders (the points O_1,...,O_4) and is perpendicular to their axes of rotation (Fig. 3, b). Because of this, all the values describing the cylinders, their shifts and the forces impacting on the cylinders in both halves of the cylinders divided by the plane are identical.

Vrednosti stiska oblog $\Delta_{s,j}$ (kadar so znane, so vrednosti tlaka $p_{s,j}$ izračunane z enačbo (1)) dobimo z uporabo nadzornih točk $A_1,...,A_4$ v enotah končnih elementov, ki so na levem robu delovnih površin valjev (sl. 3, b). Spremembe pri razdaljah A_1A_2, A_2A_3, A_3A_4, ki jih dobimo z izračunom, so enake $\Delta_{s,j}$ upoštevaje dovoljeno napako. Zaradi simetrije je dovolj, da ugotovimo navedene spremembe le na eni strani valja.

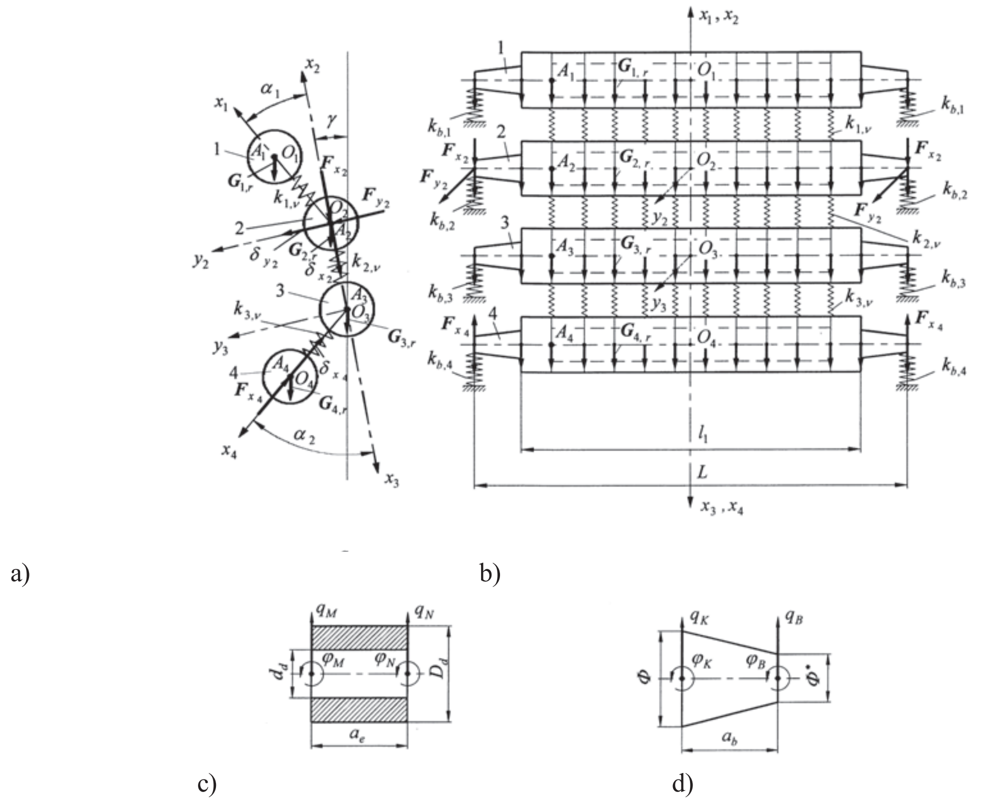
Ležaji valjev so v približku enaki linearnim elastičnim elementom, katerih koeficienti togosti $k_{b,1},...,k_{b,4}$ so izračunani tako kakor v viru [7] (sodobni tiskarski stroji imajo valjaste kotalne ležaje, elastičnost njihovih sklopov pa je v približku premočrtna).

Plast obloge med valjema smo simulirali z diskretnimi nelinearnimi elastičnimi elementi, ki povezujejo delovne površine dvojice valjev, ki pritiskajo drug ob drugega ob dotikališčih končnih

The values of the compression of the blankets $\Delta_{s,j}$ (when they are known, the values of the pressure $p_{s,j}$ are found according to Formula (1)) are found using the control points $A_1,...,A_4$ in the units of the finite elements, situated at the left-hand edge of the working surfaces of the cylinders (Fig. 3, b). Changes of the distances A_1A_2, A_2A_3, A_3A_4, found in the calculation, are equal to $\Delta_{s,j}$, with a permissible error. Because of the symmetry, it is sufficient to find these changes at only one end of the cylinders.

The bearings of the cylinders are approximated with linear elastic elements whose coefficients of stiffness $k_{b,1},...,k_{b,4}$ are calculated according to [7] (in modern printing stations, cylindrical rolling bearings are used, and the elasticity of their assemblies is approximately rectilinear).

The layer of blanket between the cylinders is simulated with discrete non-linear elastic elements that connect the working surfaces of pairs of cylinders that are pressed against each other at the junctions of the finite



Sl. 3. *Shema izračunov za tiskarski stroj, prikazan na sliki 1: a – pogled s strani; b – postavitev glede na osi vrtenja valjev; c, d – tipi končnih elementov, ki simulirajo valje;* $q_M,...,q_B$, $\varphi_M,...,\varphi_B$ *– linearne in kotne koordinate, ki določajo lege sklopov končnih elementov*

Fig. 3. *The scheme of calculations of the printing station shown in Fig. 1: a – the view from the end; b – the cylinders;* $q_M,...,q_B$, $\varphi_M,...,\varphi_B$ *– the linear and angular coordinates that define the positions of the assembles of the finite elements*

elementov (sl. 3, b). Te elemente definiramo s koeficienti togosti $k_{j,\nu} = k_{1,\nu}$; $k_{2,\nu}$; $k_{3,\nu}$, ki so odvisni od največjih vrednosti stiskanja obloge (ν - zaporedno število dotikališč končnih elementov na oblogi (v primeru s sl. 3: ν = 1, 2,..., 10) v j-tem dotikalnem pasu). Vsak od njih določi togost z_ν, dolgega valjastega izreza obloge. Vrednosti z_ν so odvisne od dolžin a_e končnih elementov na oblogi (sl. 3, b, c).

Po enačbah (1) do (4) in (8) pa tudi virih ([4] do [6]) smo izpeljali naslednje izraze za koeficiente togosti elastičnih elementov, ki simulirajo oblogo:

elements (Fig. 3, b). These elements are defined by the coefficients of stiffness, $k_{j,\nu} = k_{1,\nu}$; $k_{2,\nu}$; $k_{3,\nu}$, that depend on the maximum values of the compression of the blanket (ν - the consecutive number of junctions of the finite elements covered by the blanket (in this case shown in Fig. 3: ν= 1, 2,..., 10) in the j-th contact band). Each of them determines the stiffness of z_ν, the long cylindrical cut of the blanket. The values of z_ν depend on the lengths a_e of the finite elements covered with the blanket (Fig. 3, b, c).

On the basis of the Formulas (1) to (4) and (8) as well as the references ([4] to [6]) we found the following expressions of the coefficients of stiffness of the elastic elements that simulate the blanket to be:

$$k_{j,\nu} \cong \alpha_j \sqrt{8B_j}\; E z_\nu \psi c^{-\eta} \left(\Delta_{m,j}\right)_\nu^{\eta-0,5} \quad (j = 1, 2, 3) \qquad (9).$$

Koeficient $\alpha_1 = \alpha_3 = 1$ ($j = 1, 3$) in $\alpha_2 = 0{,}5$ (j= 2). Glede na [4] in [5] dobimo naslednji izraz:

The coefficient $\alpha_1 = \alpha_3 = 1$ ($j = 1, 3$) and α_2 = 0.5 ($j = 2$). According to [4] and [5]:

$$\psi \cong \frac{1}{12}\left(1 + 2\cdot 0{,}75^{\eta} + 4\cdot 0{,}9375^{\eta} + 4\cdot 0{,}4375^{\eta}\right) \qquad (10).$$

Torej so vsi koeficienti togosti $k_{j,\nu}$ funkcije vrednosti stiskanja obloge $(\Delta_{m,j})_\nu$.

Papirni trak 6 se pomika med valjema z gumijasto oblogo 2 in 3 (sl. 1). Lahko bi dokazali, da ima elastičnost stisnjenega papirja zanemarljiv učinek, zato je tu ne upoštevamo. Po potrebi bi elastičnost papirja lahko ocenili z ustreznim zmanjšanjem koeficientov togosti obloge med valjema 2 in 3.

Masa valjev lahko doseže nekaj sto kilogramov, zato jo moramo upoštevati. V shemi izračunov (sl. 3, a, b) je masa valjev v približku enaka silam mase $G_{d,r}$ (r = 1, 2, … – zaporedne številke dotikališč končnih elementov), ki delujejo na stikališča končnih elementov.

Pri obravnavanem tiskarskem stroju se oblogi napneta zaradi pomikov sklopov ležajev drugega valja za razdalji ($-\delta_{x2}$), δ_{y2} in zaradi pomikov sklopov ležajev četrtega valja za razdaljo ($-\delta_{x4}$) (sl. 3, a; vrednosti δ_{x2} in δ_{x4} sta negativni, ker pomikanje poteka v smereh, ki sta nasprotni smerem pozitivnih smeri osi koordinat x_2 in x_4). To dogajanje ustvarja naslednje sile pomikanja valjev 2 in 4, ki učinkujejo na priključke ležajev valjev (sl. 3, a, b):

Therefore, all the coefficients of stiffness $k_{j,\nu}$ are functions of the values of the compression of the blanket $(\Delta_{m,j})_\nu$.

The paper tape 6 moves between the blanket cylinders 2 and 3 (Fig. 1). It can be shown that the elasticity of its compression has a negligible impact, so it is not taken into account here. If necessary, the elasticity of the paper can be assessed by the corresponding reduction of the coefficients of stiffness of the blanket between the cylinders 2 and 3.

The weight of the cylinders can reach some hundreds of kilograms, so it should be taken into account. In the scheme of the calculations (Fig. 3, a, b), the weight of the cylinders is approximated with the forces of gravity $G_{d,r}$ (r = 1, 2, … - the consecutive numbers of junctions of the finite elements) acting at the junctions of the finite elements.

In the printing station under discussion the blankets are stretched by shifting the assemblies of bearings of the second cylinder distances ($-\delta_{x2}$), δ_{y2} and the assemblies of the bearings of the fourth cylinder a distance ($-\delta_{x4}$) (Fig. 3, a; the values δ_{x2} and δ_{x4} are negative, because the shifting is performed in directions opposite to the set positive directions of the axes of coordinates x_2 and x_4). This generates the following forces of the shifting of cylinders 2 and 4 that affect the cylinders in the fittings of their bearings (Fig. 3, a, b):

$$F_{x_2} = k_{b,2}\,\delta_{x_2};\;\; F_{y_2} = k_{b,2}\,\delta_{y_2};\;\; F_{x_4} = k_{b,4}\,\delta_{x_4} \qquad (11).$$

Vrednosti pomikov δ_{x2}, δ_{y2}, δ_{x4} in tudi sil F_{x2}, F_{y2}, F_{x4} so funkcije elastičnih pomikov valjev.

The values of the shifts δ_{x2}, δ_{y2}, δ_{x4} as well as of the forces F_{x2}, F_{y2}, F_{x4} are functions of the elastic shifts of the cylinders.

2 ALGORITEM IZRAČUNOV

2.1 Glavne značilnosti algoritma

Na kratko bomo opisali zaporedje izračunov. Najprej smo ugotovili absolutne pomike valjev v smereh osi koordinat $x_1,\ldots,x_4$, ki so prikazani na sl. 3, a, b. Te pomike opišemo s posplošenimi Lagrangevimi koordinatami (v nadaljevanju jih imenujemo kar koordinate) $q_1,q_2,\ldots,q_n$ (skupaj sestavljajo vektor q). Nato, z uporabo ustreznih razlik med omenjenimi koordinatami, relativne pomike med valji najdemo na ravninah, ki potekajo skozi osi vrtenja valjev 1 in 2, 3 in 4 (pomiki valjev 2 in 3 se preslikajo na te ravnine), ter skozi osi vrtenja valjev 2 in 3.

Vrednosti koordinat q lahko najdemo v rešitvi enačb oblikovanih na njihovi osnovi. Če so vrednosti pomikov sklopov ležajev δ_{x2}, δ_{y2}, δ_{x4} znane, lahko omenjene koordinate dobimo z naslednjim nelinearnim sistemom algebraičnih enačb:

2 THE ALGORITHM OF THE CALCULATIONS

2.1 The Principal Features of the Algorithm

The sequence of the calculations will be briefly described. First of all, the absolute shifts of the cylinders in the directions of the axes of coordinates $x_1,\ldots,x_4$ shown in Fig. 3, a, b were found. These shifts are described by using the generalized Lagrange coordinates (subsequently, the coordinates) $q_1,q_2,\ldots,q_n$ (altogether they form the vector q). Then, by using the corresponding differences of the said coordinates, the relative shifts between the cylinders are found in the planes that pass through the axes of the rotation of cylinders 1 and 2, 3 and 4 (the shifts of cylinders 2 and 3 are projected onto these planes), and through the axes of rotation of cylinders 2 and 3.

The values of the coordinates of q can be found in the solution of the equations formed on their basis. If the values of the shifts δ_{x2}, δ_{y2}, δ_{x4} of the assemblies of the cylinder bearings are known, the said coordinates can be found from the following non-linear system of algebraic equations:

$$\left[\mathbf{K}(q_1,\ldots,q_n)\right]\boldsymbol{q}=\boldsymbol{F}\left(\delta_{x_2},\delta_{y_2},\delta_{x_4}\right)+\boldsymbol{F}_G \qquad (12).$$

Tu del elementov matrike togosti [**K**] vsebuje koeficiente tromosti gumijaste obloge $k_{j,v}$, ki so funkcije vrednosti stiskanja obloge $\Delta_{m,j}$ pa tudi koordinat q.

Kljub temu pa je oblikovanje in reševanje enačb tipa (12) zapleteno, ker so vrednosti stiska oblog $\Delta_{s,j}=\Delta_{s,j}(q_1,\ldots,q_n)$, ki so odvisne od vrednosti pomikov δ_{x2}, δ_{y2}, δ_{x4}, sicer znane, ne poznamo pa tudi vrednosti samih δ_{x2}, δ_{y2}, δ_{x4}; prav tako je neznan analitični medsebojni odnos med temi vrednostmi. Zato so neznani tudi ustrezni izrazi komponent vektorja $\boldsymbol{F}$.

Algoritem, ki je predstavljen spodaj, predlagamo kot mogočo rešitev sistema enačb tipa (12) z uporabo iterativne metode izračuna, ki smo ga razvili prav za ta namen: z njegovo izpeljavo so linearizirani sistemi enačb rešeni ob vsaki ponovitvi in oblikovanje nelinearnega sistema enačb (12) sploh ni potrebno.

Izračun poteka v več stopnjah. Na vsaki stopnji smo preučili naslednji linearni sistem algebraičnih enačb, pridobljenih z lineariziranjem sistema (12) (metode lineariziranja so opisane v odstavku 2.2):

Here, a part of the elements of the matrix of stiffness [**K**] includes the coefficients of stiffness of the blanket $k_{j,v}$ that are functions of the values of the compression $\Delta_{m,j}$ of the blanket as well as of the coordinates of q.

However, the formation and solution of the equations of type (12) is complicated, because the values of the compression of the blankets $\Delta_{s,j}=\Delta_{s,j}(q_1,\ldots,q_n)$ that depend on the values of the shifts δ_{x2}, δ_{y2}, δ_{x4} are known, but not the values of δ_{x2}, δ_{y2}, δ_{x4} themselves, and the analytical interrelation between these values is also unknown. So, the specific expressions of the components of the vector $\boldsymbol{F}$ are unknown as well.

The algorithm described below is proposed as a solution of the systems of (12)-type equations by using the iterative method of calculation that was developed for this purpose: on its realization, the linearized systems of equations are solved on each iteration and the formation of a non-linear system of Equations (12) is not required.

The calculation is performed in several stages. At each stage, the following rectilinear system of algebraic equations obtained on the linearization of the system (12) was examined (the methods of linearization are described in paragraph 2.2):

$$\left[\mathbf{K}^{(i)}\right]\boldsymbol{q}^{(i)}=\boldsymbol{F}^{(i)}\left(\delta_{x_2}^{(i)},\delta_{y_2}^{(i)},\delta_{x_4}^{(i)}\right)+\boldsymbol{F}_G \qquad (13).$$

Tu so elementi matrike togosti [$\mathbf{K}^{(i)}$] koeficienti togosti gumijaste obloge $k_{j,\nu}^{(i)}$ (njihov izračun je opisan v odstavku 2.2), pa tudi koeficienti togosti sklopov ležajev valjev, ki kažejo deformacije valjev. Vsi elementi matrike [$\mathbf{K}^{(i)}$] so nespremenljive.

Vektor sil $\boldsymbol{F}^{(i)}$, ki pomikajo valje v i-tem približku, je funkcija približne vrednosti pomikov sklopov ležajev $\delta_{x_2}^{(i)}, \delta_{y_2}^{(i)}, \delta_{x_4}^{(i)}$, ki smo jo dobili med izračunom. Vsebuje le tri nespremenljive neničelne dvojice komponent (dve enaki komponenti za vsak pomik):

Here, the elements of the matrix of stiffness [$\mathbf{K}^{(i)}$] are the coefficients of the stiffness of the blanket $k_{j,\nu}^{(i)}$ (their calculation is described in paragraph 2.2) as well as the coefficients of the stiffness of the assemblies of bearings and the cylinders that reflect the deformations of the latter. All the elements of the matrix [$\mathbf{K}^{(i)}$] are constants.

The vector $\boldsymbol{F}^{(i)}$ of the forces shifting the cylinders in the i-th approximation is a function of the approximate values of shifts of the assemblies of bearings $\delta_{x_2}^{(i)}, \delta_{y_2}^{(i)}, \delta_{x_4}^{(i)}$ found in the course of the calculation. It includes only three constant non-zero pairs of components (the same two components for each shift):

$$F_{x_2}^{(i)} = k_{b,2} \cdot \left(-\delta_{x_2}^{(i)}\right); \quad F_{y_2}^{(i)} = k_{b,2}\delta_{y_2}^{(i)}; \quad F_{x_4}^{(i)} = k_{b,4} \cdot \left(-\delta_{x_4}^{(i)}\right) \tag{14}$$

Tudi komponente vektorja sil težnosti $\boldsymbol{F}_G$ so nespremenljive.

Z uporabo računalniško podprte metode, ki sledi shemi izračunov, smo oblikovali sistem enačb (13), ki je prikazan na sliki 3. Za njegovo oblikovanje lahko uporabimo različne algoritme, vendar pa priporočamo naslednji algoritem, ki smo ga razvili. Najprej za posamezne valje oblikujemo enačbe podsistema našega sistema (ne celotnega sistema); oblikujemo tudi elastične elemente, ki simulirajo gumijasto oblogo in so umetno ločeni od enega valja. Tako v obravnavanem primeru sestavimo naslednje ločene sisteme enačb šestih podsistemov: enačbe valja 1 z elastičnimi elementi, ki simulirajo gumijasto oblogo in so ločeni od valja 2 (njihovi koeficienti togosti so $k_{1,\nu}^{(i)}$)) – te enačbe opišejo pomike valja v smeri x_1; enačbe valja 2, ki opišejo pomike valja v smeri x_2, z elastičnimi elementi, ločenimi od valja 3 (njihovi koeficienti togosti so $k_{2,\nu}^{(i)}$); enačbe valja 2 brez elastičnih elementov, ki opišejo pomike valja v smeri y_2; dva analogna sistema enačb brez elastičnih elementov, ki posamično opišeta pomike valja 3 v smereh x_3 in y_3; enačbe valja 4, ki opišejo pomike valja v smeri x_4 z elastičnimi elementi, ločenimi od valja 3 (njihovi koeficienti togosti so v). Poleg tega oblikujemo tudi linearne algebraične povezovalne enačbe, ki opišejo povezave prostih koncev elastičnih elementov; ti simulirajo gumijasto oblogo, pri čemer so valji umetno ločeni od elastičnih elementov.

Potem ko razvijemo vseh šest sistemov enačb podsistemov ter povezovalne enačbe, jih združimo v enoten sistem (13) z uporabo računalniško podprte metode. Takšna metoda oblikovanja enačb, ki

The components of the vector $\boldsymbol{F}_G$ of the forces of gravity are constants as well.

The system of Equations (13) was formed by using a computer-aided method following the scheme of calculations and is provided in Fig. 3. Various algorithms can be used for its formation; however, we recommend the following algorithm that was developed by us. First, equations of the subsystems of the system (not of the whole system) are formed for separate cylinders, as are the elastic elements simulating the blanket that are artificially separated from one of the cylinders. In such a way, in the example under discussion, the following separate systems of equations of the six subsystems are formed: the equations of cylinder 1 with the elastic elements simulating the blanket that are separated from cylinder 2 (their coefficients of stiffness are $k_{1,\nu}^{(i)}$) – these equations describe its shifts in the direction x_1; the equations of cylinder 2 that describe its shifts in the direction x_2 with elastic elements separated from cylinder 3 (their coefficients of stiffness are $k_{2,\nu}^{(i)}$); the equations of cylinder 2 without elastic elements that describe its shifts in the direction y_2; two analogous systems of equations without elastic elements that separately describe the shifts of cylinder 3 in the directions x_3 and y_3; the equations of cylinder 4 that describe its shifts in the direction x_4 with the elastic elements separated from cylinder 3 (their coefficients of stiffness are $k_{3,\nu}^{(i)}$). In addition, linear algebraic link equations are formed; they describe the links of the free ends of the elastic elements simulating the blanket with the cylinders artificially separated from them.

After the development of all six systems of the equations of subsystems and the link equations, they are united into a single system (13) using the computer-aided method. Such a method for the formation of the uporablja posamezne podsisteme valjev, je koristna, ker z njo lahko na podlagi poznanih enačb podsistemov valjev, preprosto oblikovanih povezovalnih enačb in mnogih drugih zapletenih enačb, ustvarimo celoten sistem na način, ki lahko vključuje različna števila valjev in njihove medsebojne lege. Ta metoda razvoja enačb je podrobno opisana v virih [8] in [9].

Sedaj predpostavimo, da so vrednosti pomikov sklopov ležajev, kakor tudi komponente vektorja $\boldsymbol{F}^{(i)}\left(\delta_{x_2}^{(i)},\delta_{y_2}^{(i)},\delta_{x_4}^{(i)}\right)$, znane. Ko rešimo sistem nelinearnih algebraičnih enačb (13), ugotovimo, da potekajo absolutni pomiki v nadzornih točkah $A_1,...,A_4$ v smereh osi koordinat $x_1^{(i)},...,x_4^{(i)},y_2^{(i)},y_3^{(i)}$ v i-tem približku. Zdaj lahko ugotovimo, da so spremembe razdalj A_1A_2, A_2A_3, A_3A_4, tj. spremembe stiskov obloge med nadzornimi točkami v i-tem približku naslednje:

equations by using separate subsystems of the cylinders is useful, because on the basis of known equations of the subsystems of cylinders and the easily formed link equations and many more complicated equations the whole system can be formed in a way that involves various numbers of cylinders and their locations with respect to each other. This method of the development of the equations is described in detail in [8] and [9].

Let us suppose that the values of the shifts of the assemblies of bearings are known and the components of the vector $\boldsymbol{F}^{(i)}\left(\delta_{x_2}^{(i)},\delta_{y_2}^{(i)},\delta_{x_4}^{(i)}\right)$ are known as well. Then, after having a solution of the system of non-linear algebraic Equations (13), we find that the absolute shifts of the control points are $A_1,...,A_4$ in the directions of the axes of coordinates $x_1^{(i)},...,x_4^{(i)},y_2^{(i)},y_3^{(i)}$ in the i-th approximation. Then, the changes of the distances A_1A_2, A_2A_3, A_3A_4, i.e., the compressions of the blanket between the control points in the i-th approximation are found to be:

$$\begin{aligned}
\Delta_{s,A_1A_2}^{(i)} &= q_{A_1,x_1}^{(i)} - \left(q_{A_2,x_2}^{(i)}\cos\alpha_1 + q_{A_2,y_2}^{(i)}\sin\alpha_1\right)\\
\Delta_{s,A_2A_3}^{(i)} &= q_{A_2,x_2}^{(i)} + q_{A_3,x_3}^{(i)}\\
\Delta_{s,A_3A_4}^{(i)} &= q_{A_3,x_3}^{(i)}\cos\alpha_2 + q_{A_3,y_3}^{(i)}\sin\alpha_2 - q_{A_4,x_4}^{(i)}
\end{aligned} \qquad (15).$$

Absolutne linearne pomike $q_{A_1,x_1}^{(i)},...,q_{A_4,x_4}^{(i)}$ v točkah $A_1,...,A_4$ in v smereh ustreznih osi koordinat smo dobili v postopku rešitve sistema (13). Namesto enačb (15) lahko zapišemo naslednje:

Where the absolute rectilinear shifts $q_{A_1,x_1}^{(i)},...,q_{A_4,x_4}^{(i)}$ of the points $A_1,...,A_4$ in the directions of the relevant axes of the coordinates are found during the solution of the system (13). Instead of Equations (15) we can write the following:

$$\begin{aligned}
\Delta_{s,A_1A_2}^{(i)} &= \gamma_{A_1A_2,x_2}^{(i)}\cdot\left(-\delta_{x_2}^{(i)}\right) + \gamma_{A_1A_2,y_2}^{(i)}\cdot\delta_{y_2}^{(i)} + \gamma_{A_1A_2,x_4}^{(i)}\cdot\left(-\delta_{x_4}^{(i)}\right) + x_{A_1A_2,G}^{(i)}\\
\Delta_{s,A_2A_3}^{(i)} &= \gamma_{A_2A_3,x_2}^{(i)}\cdot\left(-\delta_{x_2}^{(i)}\right) + \gamma_{A_2A_3,y_2}^{(i)}\cdot\delta_{y_2}^{(i)} + \gamma_{A_2A_3,x_4}^{(i)}\cdot\left(-\delta_{x_4}^{(i)}\right) + x_{A_2A_3,G}^{(i)}\\
\Delta_{s,A_3A_4}^{(i)} &= \gamma_{A_3A_4,x_2}^{(i)}\cdot\left(-\delta_{x_2}^{(i)}\right) + \gamma_{A_3A_4,y_2}^{(i)}\cdot\delta_{y_2}^{(i)} + \gamma_{A_3A_4,x_4}^{(i)}\cdot\left(-\delta_{x_4}^{(i)}\right) + x_{A_3A_4,G}^{(i)}
\end{aligned} \qquad (16).$$

Enačbe (13) so linearne in pri njihovih rešitvah moramo upoštevati načelo nalaganja. Zato so enačbe (16), ki smo jih dobili z rešitvijo enačb (13), veljavne in koeficienti $\gamma^{(i)}$ z ustreznimi indeksi bodo ostali nespremenjeni za vse končne vrednosti $\delta_{x_2}^{(i)}$, $\delta_{y_2}^{(i)}$, $\delta_{x_4}^{(i)}$, $x_{A_1A_2,G}^{(i)},...,x_{A_3A_4,G}^{(i)}$ (vključno z ničelnimi vrednostmi). To značilnost enačb (16) lahko uporabimo pri izračunu koeficientov $\gamma^{(i)}$ in komponent $x_{A_1A_2,G}^{(i)},...,x_{A_3A_4,G}^{(i)}$, kakor tudi pri izračunu pomikov ležajev v i-tem približku. To bo izvedeno takole.

The equations (13) are linear and the principle of superposition is valid for their solution. Because of this, the equalities (16), found from the solution of the equations (13), are valid and the coefficients $\gamma^{(i)}$ with the relevant indices will not change on any finite values of $\delta_{x_2}^{(i)}$, $\delta_{y_2}^{(i)}$, $\delta_{x_4}^{(i)}$, $x_{A_1A_2,G}^{(i)},...,x_{A_3A_4,G}^{(i)}$ (including zero values). This feature of the equations (16) is applied to a calculation of the coefficients $\gamma^{(i)}$ and the components $x_{A_1A_2,G}^{(i)},...,x_{A_3A_4,G}^{(i)}$ and then to a calculation of the shifts of the bearings in the i-th approximation. This will be carried out as follows.

1. Izračun koeficientov $\gamma^{(i)}$ (predpostavljamo, da je $F_G=0$):

Koeficiente $\gamma_{A_1A_2,x_2}^{(i)},\gamma_{A_2A_3,x_2}^{(i)},\gamma_{A_3A_4,x_2}^{(i)}$ izračunamo po rešitvi sistema (13) in enačb (16), ki so oblikovane na predpostavki, da je $\delta_{y_2}^{(i)}=\delta_{x_4}^{(i)}=0$;

1. Calculation of the coefficients $\gamma^{(i)}$ (it is considered that $F_G=0$):

The coefficients $\gamma_{A_1A_2,x_2}^{(i)},\gamma_{A_2A_3,x_2}^{(i)},\gamma_{A_3A_4,x_2}^{(i)}$ are calculated on the basis of solutions of system (13) and the equations (16) obtained on the consideration

$\delta^{(i)}_{x_2}=1$. V tem primeru je sistem deformiran z dvema enakima posplošenima silama $F_{x2} = k_{b,2}$, ki vplivata na robove valja 2 in sta edini neničelni komponenti vektorja $F^{(i)}$. Koeficienti, ki jih moramo določiti, kar je razvidno iz enačb (16), za omenjene razmere, bodo enaki vrednostim $\Delta^{(i)}_{s,A_1A_2}, \Delta^{(i)}_{s,A_2A_3}$ in $\Delta^{(i)}_{s,A_3A_4}$.

Koeficiente $\gamma^{(i)}_{A_1A_2,y_2}, \gamma^{(i)}_{A_2A_3,y_2}, \gamma^{(i)}_{A_3A_4,y_2}$ izračunamo na način, ki je analogen sistemu (13), $\delta^{(i)}_{x_2}=\delta^{(i)}_{x_4}=0$ in $\delta^{(i)}_{y_2}=1$ (na sistem vplivata dve posplošeni sili $F_{y2} = k_{b,2}$), koeficiente $\gamma^{(i)}_{A_1A_2,x_4}, \gamma^{(i)}_{A_2A_3,x_4}, \gamma^{(i)}_{A_3A_4,x_4}$ pa dobimo s predpostavko, da je $\delta^{(i)}_{x_2}=\delta^{(i)}_{y_2}=0$ in $\delta^{(i)}_{x_4}=1$ (na sistem vplivata dve posplošeni sili $F_{x4} = k_{b,4}$).

2. Komponente $x^{(i)}_{A_1A_2,G}, x^{(i)}_{A_2A_3,G}, x^{(i)}_{A_3A_4,G}$, ki se pojavijo zaradi sile teže, ki delujejo na valje, dobimo iz enačb (13) in (16), če upoštevamo, da je $\delta^{(i)}_{x_2}=\delta^{(i)}_{y_2}=\delta^{(i)}_{x_4}=0$ in $F_G \neq 0$. Po izračunu, dobljenem na tej osnovi, bodo komponente enake vrednostim $\Delta^{(i)}_{s,A_1A_2},\ldots,\Delta^{(i)}_{s,A_3A_4}$.

3. Ko dobimo koeficiente $\gamma^{(i)}_{A_1A_2,x_2},\ldots,\gamma^{(i)}_{A_3A_4,x_4}$ in komponente, ki se pojavijo zaradi sile teže, enačbe (16) uporabimo za določitev približnih pomikov sklopov ležajev valjev. Za ta namen približne vrednosti stiskanja gumijastih oblog $\Delta^{(i)}_{s,A_1A_2},\ldots,\Delta^{(i)}_{s,A_3A_4}$ na levi strani enačb (16) nadomestimo z natančnimi potrebnimi vrednostmi stiskanja gumijastih oblog $\Delta_{s,A1\,A2}$, $\Delta_{s,A2\,A3}$, $\Delta_{s,A3\,A4}$. Približne vrednosti pomikov sklopov ležajev nato dobimo iz naslednjega sistema enačb:

that $\delta^{(i)}_{y_2}=\delta^{(i)}_{x_4}=0$; $\delta^{(i)}_{x_2}=1$. In such a case, the system is deformed by two equal generalized forces $F_{x2} = k_{b,2}$ affecting the ends of cylinder 2, which are the only non-zero components of the vector $F^{(i)}$, and the coefficients to be found, as can be seen from the equations (16), on the said conditions, will be equal to the values of $\Delta^{(i)}_{s,A_1A_2}, \Delta^{(i)}_{s,A_2A_3}$ and $\Delta^{(i)}_{s,A_3A_4}$.

The coefficients $\gamma^{(i)}_{A_1A_2,y_2}, \gamma^{(i)}_{A_2A_3,y_2}, \gamma^{(i)}_{A_3A_4,y_2}$ are calculated in a analogous way to that in System (13), $\delta^{(i)}_{x_2}=\delta^{(i)}_{x_4}=0$ and $\delta^{(i)}_{y_2}=1$ (the system is affected by two generalized forces $F_{y2} = k_{b,2}$) and the coefficients $\gamma^{(i)}_{A_1A_2,x_4}, \gamma^{(i)}_{A_2A_3,x_4}, \gamma^{(i)}_{A_3A_4,x_4}$ – on the consideration that $\delta^{(i)}_{x_2}=\delta^{(i)}_{y_2}=0$ and $\delta^{(i)}_{x_4}=1$ (the system is affected by two generalized forces $F_{x4} = k_{b,4}$).

2. The components $x^{(i)}_{A_1A_2,G}, x^{(i)}_{A_2A_3,G}, x^{(i)}_{A_3A_4,G}$ caused by the force of gravity acting on the cylinders are found from the equations (13) and the equalities (16), taking into account that $\delta^{(i)}_{x_2}=\delta^{(i)}_{y_2}=\delta^{(i)}_{x_4}=0$ and $F_G \neq 0$. Based on the mentioned conditions, they will be equal to the values of $\Delta^{(i)}_{s,A_1A_2},\ldots,\Delta^{(i)}_{s,A_3A_4}$.

3. After finding the coefficients $\gamma^{(i)}_{A_1A_2,x_2},\ldots,\gamma^{(i)}_{A_3A_4,x_4}$ and the components caused by the forces of gravity, the equalities (16) are used to determine the approximate shifts of the assemblies of bearings of the cylinders. For this purpose, the approximate values of the compressions of the blankets $\Delta^{(i)}_{s,A_1A_2},\ldots,\Delta^{(i)}_{s,A_3A_4}$ on the left-hand side of the equalities (16) are replaced with the exact required values of the compression of the blankets $\Delta_{s,A1\,A2}$, $\Delta_{s,A2\,A3}$, $\Delta_{s,A3\,A4}$. Then the approximate values of the shifts of the assemblies of the bearings are found from the following system of equations:

$$\begin{cases} -\gamma^{(i)}_{A_1A_2,x_2}\delta^{(i)}_{x_2} + \gamma^{(i)}_{A_1A_2,y_2}\delta^{(i)}_{y_2} - \gamma^{(i)}_{A_1A_2,x_4}\delta^{(i)}_{x_4} = \Delta_{s,A_1A_2} - x^{(i)}_{A_1A_2,G} \\ -\gamma^{(i)}_{A_2A_3,x_2}\delta^{(i)}_{x_2} + \gamma^{(i)}_{A_2A_3,y_2}\delta^{(i)}_{y_2} - \gamma^{(i)}_{A_2A_3,x_4}\delta^{(i)}_{x_4} = \Delta_{s,A_2A_3} - x^{(i)}_{A_2A_3,G} \\ -\gamma^{(i)}_{A_3A_4,x_2}\delta^{(i)}_{x_2} + \gamma^{(i)}_{A_3A_4,y_2}\delta^{(i)}_{y_2} - \gamma^{(i)}_{A_3A_4,x_4}\delta^{(i)}_{x_4} = \Delta_{s,A_3A_4} - x^{(i)}_{A_3A_4,G} \end{cases} \quad (17).$$

2.2 Stopnje izračuna

Na podlagi ugotovitev iz poglavja 2.1, pomike valjev izračunamo v naslednjih stopnjah.

1. stopnja (ničti približek, $i = 0$)

a) Predpostavimo, da so valji absolutno togi. Če so potrebne vrednosti stiskanja gumijastih oblog $\Delta_{s,A1\,A2}$, $\Delta_{s,A2\,A3}$, $\Delta_{s,A3\,A4}$ znane in jih vnesemo v enačbo (8) namesto $\Delta_{m,j}$, bomo dobili ničti približek koeficientov togosti elastičnih elementov obloge $k^{(0)}_{1,\nu}, k^{(0)}_{2,\nu}, k^{(0)}_{3,\nu}$.

2.2 The stages of calculation

Following the statements provided in subparagraph 3.1, the shifts of the cylinders will be calculated in the following stages.

The Stage 1 (the zero approximation, $i = 0$)

a) The cylinders are considered absolutely stiff. If the required values of the compression of the blankets $\Delta_{s,A1\,A2}$, $\Delta_{s,A2\,A3}$, $\Delta_{s,A3\,A4}$ are known and they are written into formula (8) instead of $\Delta_{m,j}$, the coefficients of stiffness $k^{(0)}_{1,\nu}, k^{(0)}_{2,\nu}, k^{(0)}_{3,\nu}$ of the elastic elements of the blanket are found in the zero approximation.

b) Potem ko izračunamo navedene koeficiente togosti, z uporabo teh koeficientov in drugih znanih parametrov obravnavanega sistema sestavimo matriko koeficientov togosti [K$^{(0)}$] v ničtem približku.
c) Nato izračunamo koeficiente $\gamma^{(0)}_{A_1A_2,x_2},...,\gamma^{(0)}_{A_3A_4,x_4}$ iz enačbe (16) in komponente $x^{(0)}_{A_1A_2,G}$, $x^{(0)}_{A_2A_3,G},x^{(0)}_{A_3A_4,G}$, ki se pojavijo zaradi sile teže, ki deluje na valje.
d) Iz enačb (17) dobimo vrednosti $\delta^{(0)}_{x_2},\delta^{(0)}_{y_2},\delta^{(0)}_{x_4}$, nato pa sestavimo vektor $\boldsymbol{F}^{(0)}\left(\delta^{(0)}_{x_2},\delta^{(0)}_{y_2},\delta^{(0)}_{x_4}\right)$ z neničelnimi komponentami (14).
e) Rešimo sistem ničtega približka (13), pri katerem upogibe in druge pomike valjev ocenimo v ničtem približku:

b) After the calculation of the said coefficients of stiffness is completed, the matrix of the coefficients of stiffness [K$^{(0)}$] in the zero approximation is formed by using that coefficient and other known parameters of the system under discussion.
c) Then the coefficients $\gamma^{(0)}_{A_1A_2,x_2},...,\gamma^{(0)}_{A_3A_4,x_4}$ from the equality (16) and the components $x^{(0)}_{A_1A_2,G},x^{(0)}_{A_2A_3,G},x^{(0)}_{A_3A_4,G}$ caused by the force of gravity acting on the cylinders are calculated.
d) The values of $\delta^{(0)}_{x_2},\delta^{(0)}_{y_2},\delta^{(0)}_{x_4}$ are found from the equations (17) and then the vector $\boldsymbol{F}^{(0)}\left(\delta^{(0)}_{x_2},\delta^{(0)}_{y_2},\delta^{(0)}_{x_4}\right)$ with non-zero components (14) is formed.
e) The system of zero approximation (13), where bends and other shifts of the cylinders are assessed in the zero approximation, is solved:

$$\left[\mathbf{K}^{(0)}\right]\boldsymbol{q}^{(0)}=\boldsymbol{F}^{(0)}\left(\delta^{(0)}_{x_2},\delta^{(0)}_{y_2},\delta^{(0)}_{x_4}\right)+\boldsymbol{F}_G \tag{18}$$

in dobimo ničte približke vrednosti koordinat $q^{(0)}$.

and the values of the coordinates $q^{(0)}$ in the zero approximation are found.

2. stopnja (prvi približek, *i*=1)

a) Ko poznamo vrednosti koordinat $q^{(0)}$, tj. ko ocenimo ničte približke pomikov valjev, vključno z upogibi, lahko z enačbo (9) bolj natančno določimo koeficiente togosti elastičnih elementov gumijaste obloge, nato pa tudi bolj natančno definiramo matriko koeficientov togosti [K$^{(1)}$].
b) Zdaj izračunamo bolj natančno definirane koeficiente $\gamma^{(1)}_{A_1A_2,x_2},\ldots,\gamma^{(1)}_{A_3A_4,x_4}$ iz enačb (16) pa tudi komponente $x^{(1)}_{A_1A_2,G},x^{(1)}_{A_2A_3,G},x^{(1)}_{A_3A_4,G}$.
c) Iz enačb (17) dobimo približne vrednosti pomikov sklopov ležajev $\delta^{(1)}_{x_2},\delta^{(1)}_{y_2},\delta^{(1)}_{x_4}$, nato pa sestavimo vektor $\boldsymbol{F}^{(1)}\left(\delta^{(1)}_{x_2},\delta^{(1)}_{y_2},\delta^{(1)}_{x_4}\right)$ z neničelnimi komponentami (14).
d) Sestavimo naslednji sistem enačb:

The Stage 2 (the first approximation, *i*=1)

a) When the values of the coordinates $q^{(0)}$ are known, i.e., after an assessment of the shifts of the cylinders in the zero approximation, including the bends, the coefficients of the stiffness of the elastic elements of the blanket are more closely defined by using formula (9) and then the more closely defined matrix of the coefficients of stiffness [K$^{(1)}$] is formed.
b) The more closely defined coefficients $\gamma^{(1)}_{A_1A_2,x_2}$, $\ldots,\gamma^{(1)}_{A_3A_4,x_4}$ included into the equalities (16) as well as the components $x^{(1)}_{A_1A_2,G},x^{(1)}_{A_2A_3,G},x^{(1)}_{A_3A_4,G}$ are then calculated.
c) The approximate values of the shifts of the assemblies of bearings $\delta^{(1)}_{x_2},\delta^{(1)}_{y_2},\delta^{(1)}_{x_4}$ are found from Equations (17), and the vector $\boldsymbol{F}^{(1)}\left(\delta^{(1)}_{x_2},\delta^{(1)}_{y_2},\delta^{(1)}_{x_4}\right)$ with non-zero components (14) is then formed.
d) The following system of equations is formed:

$$\left[\mathbf{K}^{(1)}\right]\boldsymbol{q}^{(1)}=\boldsymbol{F}^{(1)}\left(\delta^{(1)}_{x_2},\delta^{(1)}_{y_2},\delta^{(1)}_{x_4}\right)+\boldsymbol{F}_G \tag{19}$$

Ko ga rešimo, dobimo vrednosti prvega približka koordinat $q^{(1)}$.

After its solution, the values of the coordinates $q^{(1)}$ in the first approximation are found.

Naslednje stopnje (i=2, 3, ...)

S približnimi vrednostmi koordinat $q^{(i-1)}$, dobljenih na predhodni stopnji, bolj natančno definiramo koeficiente togosti gumijaste obloge in sestavimo matriko [$\mathbf{K}^{(i)}$]. Nato dobimo koeficiente $\gamma^{(i)}_{A_1A_2,x_2},...,\gamma^{(i)}_{A_3A_4,x_4}$ iz enačb (16) in komponente $x^{(i)}_{A_1A_2,G},x^{(i)}_{A_2A_3,G},x^{(i)}_{A_3A_4,G}$, iz enačb (17) pa dobimo

Further stages (i=2, 3, ...)

By using the approximate values of the coordinates $q^{(i-1)}$ found at the previous stage, the coefficients of the stiffness of the blanket are more closely defined and the matrix [$\mathbf{K}^{(i)}$] is formed. Then, the coefficients $\gamma^{(i)}_{A_1A_2,x_2},...,\gamma^{(i)}_{A_3A_4,x_4}$ included in the equalities (16), and the components $x^{(i)}_{A_1A_2,G},x^{(i)}_{A_2A_3,G},x^{(i)}_{A_3A_4,G}$ are vrednosti pomikov $\delta_{x_2}^{(i)}, \delta_{y_2}^{(i)}, \delta_{x_4}^{(i)}$. Po oblikovanju vektorja $\boldsymbol{F}^{(i)}\left(\delta_{x_2}^{(i)},\delta_{y_2}^{(i)},\delta_{x_4}^{(i)}\right)$, dobimo sisteme enačb, kjer je i = 2, 3, 4 itn. V postopku iskanja rešitve vsakega od sistemov bolj natančno definiramo vrednosti koordinat $q^{(i)}$.

Na teh stopnjah z izračunom nadaljujemo, dokler razlike med koordinatama $q^{(i)}$ in $q^{(i-1)}$, komponentami vektorjev, pomiki sklopov ležajev valjev in drugimi vrednostmi, ki smo jih dobili med potekom zadnjih dveh stopenj, ne postanejo manjše od 1%. Ko se to zgodi, prenehamo z izračunavanjem.

Iz dobljenih rezultatov dobimo potrebne vrednosti pomikov ležajev valjev pa tudi zakonitost razporeditve deformacij valjev ter tlaka med valjema, ki pritiskata drug ob drugega preko gumijaste obloge.

Z uporabo teh metod v praksi smo ugotovili, da dosežemo zadostno natančnost rezultatov (ko napaka znaša manj kot 1%), ko je i = 4,5.

3 PRIMER

Z uporabo tu opisane metode smo raziskali upogibe valjev tiskarskega stroja “Compacta 213”, izdelanega v Nemčiji, in razporeditev tlaka vzdolž njihovih delovnih površin. Obdelali smo pomike polnih valjev in cevastih valjev. Parametri valjev so bili naslednji (isti za vse valje) L =1,384 m; l_1 =1,04m; l_2 =1,00m; l_3 =0,96m; D_d =0,2m; $k_{b,1}=k_{b,2}=k_{b,3}=k_{b,4}$ =1,7·10^{6}N/m; E =500MPa; η=1,5; α_1 = 30°; α_2 = 50°; γ = 35°; spreminjali smo premer odprtin valjev d_d, vsak valj pa smo razdelili na 13 končnih elementov (sl. 3).

Po opravljeni raziskavi smo ugotovili, da relativni upogibi valjev z gumijasto oblogo in valjev s ploščo, ki pritiskajo drug ob drugega, lahko znašajo do 24 mm in povzročijo 14 do 31 odstotkov sprememb v tlaku vzdolž delovnih površin valjev. To lahko vidno poslabša kakovost tiska. Pokazali smo, da ni praktično uporabljati cevaste valje s tankimi stenami (valji tiskarskega stroja “Compacta 213” so polni).

Da bi poenotili razporeditev tlaka, lako uporabljamo valje, ki imajo izbočene ali vbočene delovne površine, ne pa valjastih površin. Za ta namen smo, z uporabo posebnega programa, spremembe v premeru valjev določili tako, da smo zagotovili premo dotikalno površino valjev. V takšnem primeru je tlak vzdolž dotikalnih pasov valjev nespremenljiv.

found, and the values of the shifts $\delta_{x_2}^{(i)}, \delta_{y_2}^{(i)}, \delta_{x_4}^{(i)}$ are calculated from the equations (17). Then, after the formation of the vector $\boldsymbol{F}^{(i)}\left(\delta_{x_2}^{(i)},\delta_{y_2}^{(i)},\delta_{x_4}^{(i)}\right)$, systems of equations, where i = 2, 3, 4 and so on, are formed. In the course of finding a solution to each of these systems, the values of the coordinates $q^{(i)}$ are more closely defined.

The calculation at these stages is continued until the differences between the coordinates $q^{(i)}$ and $q^{(i-1)}$, the components of the vectors, the shifts of the assemblies of cylinder bearings and other values found during the two last stages become less than 1%. In such a case the calculation is then stopped.

From the obtained results the necessary values of the shifts of the cylinder bearings as well as the regularities of the distribution of deformations of the cylinders and the pressure between the cylinders that are pressed against each other via blankets are found.

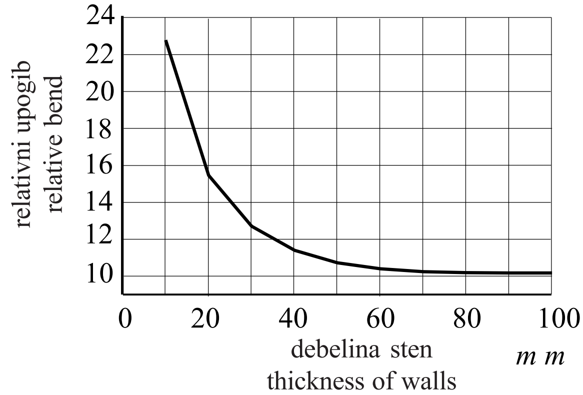
By using these methods in practice we found that sufficient accuracy (when the error is less than 1%) is achieved when i = 4.5.

3 THE EXAMPLE

By using the method described herein, the bends of the cylinders of the “Compacta 213” printing station, made in Germany, and the distribution of the pressure along their working surfaces were investigated. The shifts of the solid and the pipe-type cylinders were investigated. The parameters of the cylinders were as follows (the same for all cylinders): L =1.384 m; l_1 =1.04m; l_2 =1.00m; l_3 =0.96m; D_d =0.2m; $k_{b,1}=k_{b,2}=k_{b,3}=k_{b,4}$ =1.7·10^{6}N/m; E =500MPa; η =1.5; α_1 = 30°; α_2 = 50°; γ = 35°; the diameter d_d of cylinder holes was varied, and each cylinder was divided into 13 finite elements (see Fig. 3).

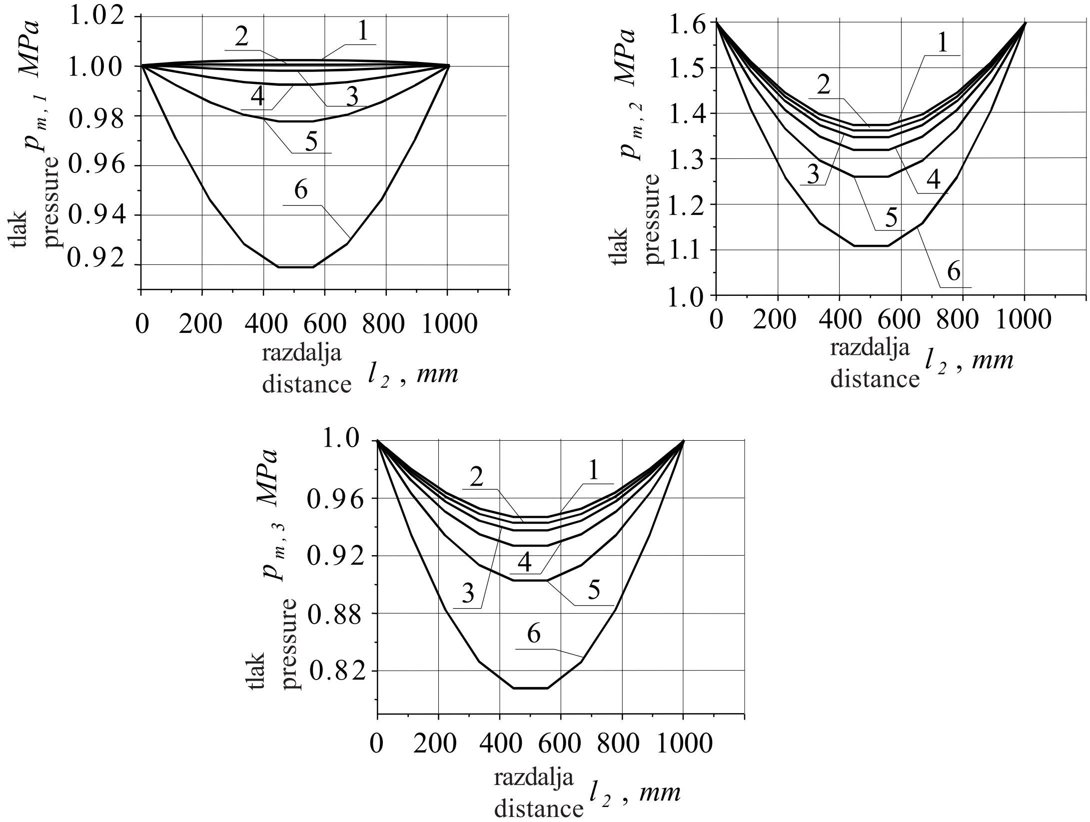
After the completion of the investigation it was found that the relative bends of the blanket and the plate cylinders that were pressed against each other can be up to 24 μm and cause 14–31% changes in the pressure along the working surfaces of the cylinders. This may noticeably deteriorate the quality of the prints. It was shown that it is not useful to use pipe-type cylinders with thin walls (the cylinders of the “Compacta 213” printing press are solid).

In order to equalize the distribution of the pressure, the working surfaces of the cylinders can be made convex or concave, but not cylindrical. For this purpose, by using the special programme, changes of the diameters of the cylinders are chosen in such way so as to ensure a rectilinear surface of the contact of the cylinders. In such a case, the pressure along the contact bands of the cylinders remains constant.



Sl. 4. *Največji relativni upogib med delovnima površinama valjev z gumijasto oblogo 2, 3 pri različnih debelinah sten votlih valjev 1, 2, 3, 4 (so enake za vse štiri valje)*

Fig. 4. *The maximum relative bend between the working surfaces of the blanket cylinders 2, 3 on the various widths of the walls of the hollow cylinders 1, 2, 3, 4 (the same for all four cylinders)*



Sl. 5. *Porazdelitev tlaka $p_{m,j}$ vzdolž delovnih površin valjev, ki pritiskajo drug ob drugega, za primer ocenjenih upogibov valjev: a – med valjem s ploščo 1 in valjem z gumijasto oblogo 2; b – med valjema z gumijasto oblogo 2 in 3; med valjem z gumijasto oblogo 3 in valjem s ploščo 4; 1 – polni valji; 2 – votli valji z debelino sten 50 mm; 3 – 40 mm, 4 – 30 mm, 5 – 20 mm, 6 – 10 mm*

Fig. 5. *The distribution of the pressure $p_{m,j}$ along the working surfaces of cylinders that are pressed against each other on an assessment of the bends of the cylinders: a – between the plate cylinder 1 and the blanket cylinder 2; b – between the blanket cylinders 2 and 3; between the blanket cylinder 3 and the plate cylinder 4; 1 – solid cylinders; 2 – hollow cylinders with the width of the walls 50 mm; 3 – 40 mm, 4 – 30 mm, 5 – 20 mm, 6 – 10 mm*

Primeri rezultatov izračunov so prikazani na slikah 4 in 5.

Examples of the results of the calculation are shown in Fig. 4 and 5.

4 SKLEPI

1. Predlagali smo metodo raziskave upogibov in drugih elastičnih pomikov valjev s ploščo, valjev z gumijasto oblogo in tiskovnih valjev rotacijskega ofsetnega tiskarskega stroja, ki pritiskajo drug ob drugega prek gumijaste obloge, pa tudi spremembe v tlaku vzdolž valjev. Obravnavani problem smo razrešili kot zahteven nelinearni statični problem, pri katerem imamo na začetku potreben tlak na robovih delovnih površin valjev, ki pritiskajo drug ob drugega, medtem pa so pomiki sklopov ležajev, ki omogočijo potrebne vrednosti tlaka, neznane.
2. S predlagano metodo smo ocenili nelinearne značilnosti elastičnosti obloge, upogibe valjev, elastičnost sklopov ležajev, silo teže in lege osi vrtenja valjev glede na njihove medsebojne lege.
3. Z uporabo metode na specifičnem primeru smo pokazali, da upogibi valjev bistveno vplivajo na porazdelitev tlaka vzdolž dotikalnega predela valjev v tiskarskem stroju in na kakovost tiska.

4 CONCLUSIONS

1. A method of investigating the bends and other elastic shifts of the plate, blanket and impression cylinders of the web offset printing press that are pressed against each other via blankets as well as the changes of the pressure along the cylinders was proposed. The problem under discussion was settled as a complex nonlinear static problem, where in the beginning the required pressures at the edges of the working surfaces of the cylinders that are pressed against each other are provided, and the shifts of the assemblies of the cylinder bearings required to ensure the said values of the pressures are unknown.
2. In the proposed method, the nonlinear features of the elasticity of the blanket's material, the bends of the cylinders, the elasticity of the assemblies of their bearings, the force of gravity and the location of the axes of the rotation of the cylinders with respect to each other are assessed.
3. With the application of the method to the specified example, it was shown that the bends of the cylinders noticeably impacted on the distribution of pressure along the contact zone of the cylinders in the printing press and on the quality of the prints.

5 VIRI
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Naslov avtorjev:
prof. dr. Vytautas Kazimieras Augustaitis
prof. dr. Nikolaj Šešok
prof. dr. Igor Iljin
Vilnius Gediminas Technical University
Faculty of Mechanics
Department of Printing Machines
J. Basanavičiaus 28
LT-2009 Vilnius, Lithuania
pgses@me.vtu.lt

Authors' address:
Prof. Dr. Vytautas Kazimieras Augustaitis
Prof. Dr. Nikolaj Sheshok
Prof. Dr. Igor Iljin
Vilnius Gediminas Technical University
Faculty of Mechanics
Department of Printing Machines
J. Basanavičiaus 28
LT-2009 Vilnius, Lithuania
pgses@me.vtu.lt

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Energijska analiza pridelave oljne ogrščice za potrebe proizvodnje biodizla na Hrvaškem

An Energy Analysis of Rapeseed Production for Biodiesel in Croatia

Dubravko Filipović - Tajana Krička
(University of Zagreb, Croatia)

Preden smo na Hrvaškem začeli s izdelavo biodizla iz olja ogrščice, smo izvedli energijsko analizo pridelave tega pridelka. Podatke, potrebne za energijsko analizo, smo v obdobju 2001 do 2003 zbirali na podlagi raziskav, izvedenih na kmetijah v Slavoniji, ki je glavno kmetijsko področje na Hrvaškem. Za izračun dovedene in odvedene energije smo rezultate raziskave uporabili skupaj z energijskimi vrednostmi iz primerjalne literature. Povprečen pridelek oljne ogrščice je bil 3195 kg/ha, njena povprečna vsebnost olja pa 42,5%. Če upoštevamo, da smo oljno ogrščico uporabili za izdelavo biodizla in za krmo živali, je znesek celotne odvedene energije znašal 74,01 GJ/ha. Celotna dovedena energija, potrebna za pridelavo oljne ogrščice, je bila 23,44 GJ/ha, pri čemer je bilo največ te energije namenjene porabi gnojil in goriva. Energijsko razmerje pri pridelavi oljne ogrščice je bilo 3,16, čisti energijski prirastek je bil 50,56 GJ/ha, in energijska donosnost je bila 49,23 L/GJ. Predlagali smo tudi nekatere možnosti varčevanja z energijo v postopku pridelave oljne ogrščice.

(Ključne besede: oljna ogrščica, analize energijske, varčevanje z energijo, biodizel)

Before starting with biodiesel production from rapeseed oil in Croatia, an energy analysis of the production was made. The data for the energy analysis were gathered from investigations on farms in the Slavonia region, the main arable production area in Croatia, in the period 2001 to 2003. For the calculation of energy outputs and inputs, the results of this investigation were used together with energy values from the literature. The average rapeseed yield and oil content was 3195 kg/ha and 42.5%, respectively. Considering rapeseed oil for biodiesel and meal for animal feed, the total energy outputs were 74.01 GJ/ha. Total energy inputs for the rapeseed production were 23.44 GJ/ha, and the major energy inputs were fertilizers and fuel. The energy ratio of the rapeseed production was 3.16, the net energy gain was 50.56 GJ/ha, and the energy productivity was 49.23 L/GJ. Some possibilities for energy saving in rapeseed production were also suggested.

(Keywords: rapeseed, energy analysis, energy saving, biodiesel)

0 UVOD

Zaradi omejenih virov fosilnih goriv in problemov z emisijami toplogrednih plinov v ozračje postajajo dandanes nadomestna goriva za dizelske motorje vse bolj pomembna. Biodizel je nadomestno tekoče gorivo narejeno iz obnovljivih bioloških virov, kot na primer rastlinska olja. Do prvih pobud za rabo biodizla je prišlo leta 1981 v Južni Afriki in leta 1982 v Avstriji, Nemčiji in na Novi Zelandiji. Že leta 1985 so v majhnem poskusnem obratu v Avstriji poskusno naredili biodizel iz olja ogrščice, leta 1990 pa je prva kmetijska zadruga pričela s pridelavo oljne ogrščice

0 INTRODUCTION

Alternative fuels for diesel engines are becoming increasingly important today due to the limited resources of fossil fuels and problems with the emission of greenhouse gases into the atmosphere. Biodiesel is an alternative liquid fuel, which is made from renewable biological sources such as plant oils. The first biodiesel initiatives were reported in 1981 in South Africa and then in 1982 in Austria, Germany and New Zealand. As early as 1985, a small pilot plant in Austria tested the production of biodiesel from rapeseed oil, and in 1990 the first farm-v tržne namene. Naslednji pomembni korak je bila uveljavitev prvega standarda glede goriv za biodizel, ON C 1190, ki ga je leta 1991 potrdil avstrijski inštitut za standardizacijo in s tem zagotovil visoko kakovost tako narejenega goriva. Velik korak naprej je bil narejen leta 1996, ko so se v Franciji in Nemčiji pojavili veliki industrijski obrati in ko je bil ustanovljen Evropski odbor za biodizel, strokovna organizacija, ki vključuje vse večje izdelovalce biodizla [1]. Poleg omenjenih držav se biodizel gospodarsko uporablja kot nadomestno gorivo tudi v drugih evropskih državah, na primer v Belgiji, na Češkem, Madžarskem, v Italiji in na Švedskem [2]. Čeprav sicer večina sodobnih raziskav na področju uporabe nadomestkov dizelskih goriv poteka v razvitih državah, pa so potrebe po teh nadomestkih in možnosti za njihovo izdelavo precej večje v državah v razvoju [3]. Podobno kakor druge evropske države, Hrvaška nima ustreznih zalog virov fosilnih goriv, kar vodi v odvisnost od uvoza nafte, potrebne za zadovoljitev potreb po bencinu in dizelskem gorivu na področju prevoza. Zadovoljiva ponudba biodizla bi zato pripomogla k zmanjšanju tovrstne odvisnosti. Hkrati pridelovanje oljčnic velja za eno najboljših izbir v kmetijstvu, saj se je pridelava za prehrambne namene zmanjšala v skladu z evropsko kmetijsko politiko. Ta okoliščina omogoča razvoj novih panog, na primer agroenergijske industrije, s tem pa tudi zaposlovanje in pokrajinski razvoj [4].

V Evropi je najpomembnejša surovina za izdelavo biodizla oljna ogrščica, tej sledita sončnica in soja [5]. Glede na vir [1] je oljna ogrščica odlična za pridelavo v evropskem podnebju, biodizelsko gorivo, pridobljeno iz te rastline, pa se zelo približa značilnostim običajnega dizelskega goriva. Poleg tega nove različice oljne ogrščice z dvojno ničlo (00) vsebujejo manj ko 0,1% eruka kisline in 8 do 12 mol/g glukosinolata, zaradi česar sta izboljšana kakovost olja in krme [6]. Leta 2001 smo na Hrvaškem začeli projekt izdelave biodizla iz olja ogrščice, saj lahko uporaba biodizla kot nadomestnega goriva Hrvaški prinese gospodarske, okoljevarstvene in energijske koristi. Zaradi majhnih gospodarskih donosov glavnih poljščin na Hrvaškem, koruze in pšenice, so mnogi kmetje pokazali zanimanje za pridelavo oljne ogrščice. Eden napomembnejših pogojev za odločitev, ali je pridelek oljne ogrščice primeren za proizvodnjo biodizla, je zahteva, da, v primerjavi z energijo, ki smo jo porabili za pridelek, omogoča pozitivni energijski donos, tj. da s pridelavo

ers' cooperative started with commercial production. Another important step was establishing the first fuel standard for biodiesel, ON C 1190, in 1991 by the Austrian Standardisation Institute, so ensuring the high quality of this fuel. A big step forward was made in 1996 with start of large industrial-scale plants in France and Germany, and also with the foundation of the European Biodiesel Board as a professional organization for all major biodiesel producers [1]. In addition to the above-mentioned countries, biodiesel has been in commercial use as an alternative fuel in other European countries, like Belgium, Czech Republic, Hungary, Italy and Sweden [2]. Although much of the recent research and development in the production and use of diesel-fuel substitutes has been carried out in developed countries, the need for such substitutes and the potential for their production are much greater in developing countries [3]. Croatia, like many other European countries, does not have adequate reserves of fossil-fuel resources, which implies a dependency on petroleum imports in order to provide for the demands for petrol and diesel fuel in the transport sector. The satisfaction of this demand with biodiesel would contribute to reducing this dependency. On the other hand, oil crops have been considered as one of the best alternatives in the agricultural sector, whose production for food purposes has been limited by the PAC, thus allowing the development of new industries such as the agroenergy industry, which facilitates employment opportunities and regional development [4].

The most important raw material for biodiesel production in Europe is rapeseed, followed by sunflower and soybean [5]. According to [1], rapeseed oil is an ideal raw material for the European climate, and biodiesel fuel produced from this plant oil comes very close to the properties of conventional diesel fuel. In addition, the new double-zero (00) rapeseed varieties contain less than 0.1% erucic acid and 8 to 12 mol/g glucosinolate, which results in good oil and meal quality [6]. In 2001 Croatia started a project for biodiesel production from rapeseed oil, since using biodiesel as an alternative fuel can have economic, environmental and energy benefits for Croatia. Due to the small economic returns of the main arable crops in Croatia, maize and wheat, many farmers showed an interest in rapeseed production. One of most important requirements for an oilseed crop to be considered for biodiesel production is that it provides a positive energy return compared with the energy used for ustvarimo več energije kakor je porabimo ([7] in [8]). Preden smo na Hrvaškem pričeli z izdelavo biodizla iz oljne ogrščice, smo opravili energijsko analizo pridelave oljne ogrščice na Hrvaškem. Na podlagi podatkov iz primerjalne literature smo izrabili tudi nekaj možnosti za prihranek energije.

production, i.e., that the production yields more energy than it spends ([7] and [8]). Before starting with biodiesel production from rapeseed oil in Croatia, an energy analysis of rapeseed production in Croatia was made. Some possibilities for energy saving were made based on the literature data.

1 METODOLOGIJA ENERGIJSKE ANALIZE

Obstaja več metodologij energijske analize, ki se precej razlikujejo glede na svoja izhodišča in postopke. V tem prispevku poročamo o uporabi energijske analize, ki jo je leta 1974 predlagala Mednarodna zveza inštitutov za sodobne raziskave, pojasnila pa sta jo Ortiz-Canavate in Hernanz [9]. V tem postopku energijske analize za vsak dejavnik pridelovalnega postopka ugotovimo potrebno količino neobnovljive energije. Celotno energijo postopka ugotovimo s seštevkom delnih vnosov energije, ki so vezani na posamezne korake v postopku, ne da bi jim pri tem določili kakovostni faktor. Celotno energijo na pridelovalno enoto (hektar) ugotovimo s seštevkom delnih energij posameznih vnosov, ki se nanašajo na določeno pridelovalno enoto. Energija je lahko dovedena neposredno in posredno. Neposredno dovedena energija ($EI_{neposredna}$) vključuje energijo dizelskega goriva, porabljenega pri postopku pridelave oljne ogrščice. Posredno dovedena energija ($EI_{posredna}$) pa vključuje energijo porabljeno za izdelavo kmetijske mehanizacije, gnojil, pesticidov in semen, pa tudi energijo, potrebno za delo kmetovalcev. Namakanja nismo upoštevali, ker le-tega nismo uporabljali v poljedelstvu obravnavanega področja. Celotno dovedeno energijo ($EI_{celotna}$) zato laho izrazimo z enačbo:

1 METHODOLOGY OF THE ENERGY ANALYSIS

There are several methodologies for energy analysis; these are quite different in terms of their basis and approaches. In this paper, the energy analysis suggested in 1974 by the International Federation of Institutes for Advanced Studies and explained by Ortiz-Canavate and Hernanz [9], was applied. In this procedure, the energy analysis relies on assigning non-renewable energy amounts to each factor in the production. The total energy in a process is established by adding the partial energies associated with each step, without assigning a quality factor. The total energy per production unit (hectare) is established by the addition of the partial energies of each input, referred to the unit of production. The energy inputs are classified as direct and indirect. Direct energy inputs (EI_{direct}) include the energy of the diesel fuel consumed in rapeseed production. Indirect energy inputs ($EI_{indirect}$) include the energy for the production of farm machinery, fertilizers, pesticides and seeds, and the energy for the labor of the agricultural workers. Irrigation was not considered, because irrigation was not applied in the arable production in this region. Thus, the total energy input (EI_{total}) can be expressed with:

$$EI_{celotna/total} = EI_{neposredna/direct} + EI_{posredna/indirect} \quad [MJ/ha] \qquad (1).$$

1.1 Neposredno dovedena energija

Neposredno dovedeno energijo na hektar smo določili z izmero traktorjeve porabe goriva, potrebnega za vsako opravljeno delo na polju, ali za potreben prevoz z uporabo prostorninskega sistema. Za izračun neposredno dovedene energije pri pridelavi oljne ogrščice smo upoštevali podatke o povprečni porabi dizelskega goriva za vsako opravljeno delo na polju in za vsak prevoz. Glede na vir [9] je energiji, potrebni za prevoz goriva do kmeta (9,1 MJ/L) treba dodati še kurilno vrednost dizelskega goriva in energijo, ki je potrebna za rafiniranje surove nafte (38,7 MJ/L). Tako je celoten

1.1 Direct energy inputs

The direct energy inputs per hectare were determined by measuring the tractor's fuel consumption for completing each field operation over the known area or transported distance by applying a volumetric system. For calculating the direct energy inputs in rapeseed production, the average diesel-fuel consumption data for each field operation and transportation was taken into account. According to [9], it is necessary to add the diesel fuel's heating value and the energy needed to refine the crude oil (38.7 MJ/L) to the energy required to transport it to the farmer (9.1 MJ/L). Therefore, the total energy strošek energije za porabo enega litra dizelskega goriva 47,8 MJ/L. Stroške maziva smo šteli med stroške vzdrževanja. Če je N celotno število opravljenih del pri pridelavi oljne ogrščice, in je E_n strošek energije na hektar za vsako od teh opravljenih del, lahko celotno neposredno dovedeno energijo izračunamo z enačbo:

cost to consume a liter of diesel fuel is 47.8 MJ/L. The lubricant costs were calculated as maintenance costs. If N is the total number of operations in rapeseed production, and E_n is the energy cost per hectare for each of these operations, the total direct energy inputs can be calculated with:

$$EI_{neposredna/direct} = \sum_{n=1}^{N} E_n \quad [\mathrm{MJ/ha}] \tag{2}$$

1.2 Posredno dovedena energija

Posredno dovedena energija za kmetijsko mehanizacijo vključuje energijo potrebno za pripravo surovin (jeklo in drugo) in za proizvodni postopek, energijo za prevoz mehanizacije do porabnika in energijo za vzdrževanje in popravila mehanizacije. Glede na vir [10] je srednja vrednost za prva dva koraka ocenjena na znesek 87,0 MJ/kg, za prevoz 8,8 MJ/kg in za vzdrževanje ter popravila 47,9 MJ/kg (skupaj 143,7 MJ/kg). Za ovrednotenje posredno dovedene energije, potrebne za delovanje mehanizacije na hektar, moramo poznati maso rabljene mehanizacije, njeno dobo trajanja in njen delovni čas na hektar. Energijske vrednosti gnojil, pesticidov in semena pomenijo energijo, potrebno za pridelavo, pakiranje in dostavo končnih izdelkov. Za izračun dovedene energije gnojil N, P_2O_5 in K_2O smo uporabili vrednosti 78,1 MJ/kg, 17,4 MJ/kg in 13,7 MJ/kg [11]. Delujoča sestavina uporabljenih pesticidov (devrinol in triflurolex) je bila trifluralin, ki potrebuje energijo v znesku 171,4 MJ/kg [12]. Znesek stroškov energije za semena oljne ogrščice, 200 MJ/kg, smo povzeli po viru [13]. Za potrebe naše raziskave smo posejali semena različice Bristol, čigar uporabljeni odmerek je bil 4,6 kg/ha. Energijo, potrebno za delo kmetovalcev, smo izračunali tako, da smo čas, potreben za posamezno opravilo, spremenili v stroške energije za vloženo delo, pri čemer je bil spreminjevalni količnik 2,3 MJ/h [14]. Delovno učinkovitost smo določili z izmero časa, potrebnega za vsako delo, opravljeno na obravnavanem področju, ali za prevoz na določeni razdalji. Tako lahko celotno posredno dovedeno energijo izrazimo z enačbo:

1.2 Indirect energy inputs

The indirect energy inputs for the farm machinery included the energy required to produce the raw materials (steel and others) and the manufacturing process, the energy for the transportation of the machinery to the consumer, and the energy for the machinery's maintenance and repairs. According to [10], the mean value for the first two steps was assumed to be 87.0 MJ/kg, for transportation, 8.8 MJ/kg, and for maintenance and repairs, 47.9 MJ/kg (total 143.7 MJ/kg). To evaluate the indirect energy inputs for machinery per hectare it is necessary to know the weight of the used machinery, its working life and the working time per hectare. The energy values for fertilizers, pesticides and seeds included the energy for production, packaging and distribution of the final products. When calculating the energy inputs for N, P_2O_5 and K_2O fertilizers, the values of 78.1 MJ/kg, 17.4 MJ/kg and 13.7 MJ/kg, respectively, were used [11]. The active ingredient of the applied pesticides (Devrinol and Triflurolex) was trifluralin, which required an energy of 171.4 MJ/kg [12]. The energy cost for rapeseed seeds of 200 MJ/kg was taken from [13]. In this investigation, the rapeseed variety Bristol was sowed with an application rate of 4.6 kg/ha. The energy for human labor was calculated by converting the time for each operation into labor energy costs, with a conversion factor of 2.3 MJ/h [14]. The work rate was determined by measuring the time to finish each field operation of a known area or transport distance. Thus, the total indirect energy inputs can be expressed as follows:

$$EI_{posredna/indirect} = E_{mehanizacija/machinery} + E_{gnojila/fertilizers} + E_{pesticidi/pesticides} + E_{semena/seeds} + E_{delo/labour} \quad [\mathrm{MJ/ha}] \tag{3}$$

1.3 Odvedena energija (EO)

Odvedeno energijo pri pridelavi oljne ogrščice smo ocenili z množenjem količin izdelkov, olja za biodizel in krme za živali, z njihovimi energijskimi

1.3 Energy outputs

The energy outputs for rapeseed production were estimated by multiplying the quantities of the products, the oil for the biodiesel and the meal for vrednostmi. Glede na vir [15] so energijske vrednosti olja, pridobljenega iz oljne ogrščice in krme, 37,6 MJ/kg in 15,0 MJ/kg. Če odpadke pridelka obravnavamo kot energijo biomase, je njihova energijska vrednost 12,5 MJ/kg. V postopku naše raziskave smo odvedeno energijo izračunali s seštevkom povprečnih količin pridelka oljne ogrščice, deležem olja v semenih in količino odpadkov pridelka:

the animal feed by their energy values. According to [15], the energy values of the rapeseed oil and the meal are 37.6 MJ/kg and 15.0 MJ/kg, respectively. If crop residues are taken as biomass energy, their energy value is 12.5 MJ/kg. The energy outputs during this investigation were calculated with an average rapeseed yield, oil content in the seed and quantity of crop residues:

$$EO_{celotna/total} = E_{olje/oil} + E_{krma/meal} (+ E_{odpadki/residues}) \quad [MJ/ha] \tag{4}$$

1.4 Energijska izravnava

Da bi v tej raziskavi določili energijsko izravnavo pridelave oljne ogrščice, smo izračunali energijsko razmerje, čisti energijski prirastek in energijsko donosnost. Energijsko razmerje (ER) smo definirali kot razmerje med kurilno vrednostjo dobljenih izdelkov in vso energijo, dovedeno v postopku pridelave:

1.4 Energy balance

In this analysis, for determining the energy balance of the rapeseed production, the energy ratio, the net energy gain and the energy productivity were calculated. The energy ratio (ER) is defined as the ratio between the calorific heat of the output products and all the energy inputs that take part in the production processes:

$$ER = EO_{celotna/total} / EI_{celotna/total} \tag{5}$$

Čisti energijski prirastek (NEG) pomeni razliko med celotno odvedeno energijo in celotno dovedeno energijo, ki se nanaša na enoto pridelave, tj. 1 hektar:

The net energy gain (NEG) is the difference between the total energy outputs and the total energy inputs, related to the unit of production, i.e. 1 hectare:

$$NEG = EO_{celotna/total} - EI_{celotna/total} \quad [MJ/ha] \text{ ali/or } [GJ/ha] \tag{6}$$

Energijska donosnost (EP) je merilo količine izdelka (v tem primeru olja ogrščice; Q_{olje}) pridobljenega na enoto dovedene energije in pomeni kurilno vrednost končnega izdelka:

The energy productivity (EP) is a measure of the quantity of a product (in this case rapeseed oil, Q_{oil}) obtained per unit of input energy, and this is the calorific value of the final product:

$$EP = Q_{olje/oil} / EI_{celotna/total} \quad [L/GJ] \tag{7}$$

1.5 Poskusna polja in mehanizacija

Podatke, potrebne za energijsko analizo, smo v obdobju 2001 do 2003 zbirali z raziskovanjem na desetih izbranih kmetijah z značilno pridelavo oljne ogrščice v Slavoniji, ki je glavna obdelovalna površina na Hrvaškem. Vse te kmetije so izvajale enoten način pridelovanja. Povprečna velikost polj, ki smo jih zajeli v analizo, je bila 5,2 ha, kar je za to področje običajno. Povprečna razdalja od kmečkega poslopja do polja je bila 3,6 km, povprečna razdalja med poljem in skladiščem pa je bila 21,8 km. Preglednica 1 prikazuje mehanizacijo in pripomočke, ki so jih kmetovalci uporabljali v času naše raziskave.

1.5 Experimental fields and machinery

The data for the energy analysis were gathered by investigations on ten selected farms with typical rapeseed production in the Slavonia region, the main arable production area in Croatia, in the period 2001 to 2003. An identical production system was applied in all of the farms. The average size of the fields considered in the analysis was 5.2 ha, which is representative for this region. The average distance from the farmyard to the field was 3.6 km, while from the field to the storage facility the average distance was 21.8 km. The machines and the implements included in this investigation are presented in Table 1.

Preglednica 1. *Stroji in pripomočki za delo na polju in za transport*
Table 1. *Machine and implements for field operations and transport*

Opravila Operation	Mehanizacija Machinery
vlečenje in vožnja pulling and driving	traktor – moč motorja 55 kW tractor – 55 kW engine power
oranje ploughing	rigolni plug - 3 brazde mouldboard plough - 3 furrows
priprava semenskih gred seedbed preparation	kombajn za obdelovanje semenske grede - 3 m seedbed combine implement - 3 m
uporaba gnojil fertilizer application	centrifugalni razdeljevalnik gnojil - 620 l centrifugal fertilizer distributor - 620 l
uporaba pesticidov pesticide application	razpršilnik - 600 l field sprayer - 600 l
vnos pesticidov pesticide incorporation	kultivator - 3,2 m field cultivator - 3.2 m
setev sowing	sejalni stroj - 3 m planter - 3 m
žetev harvesting	žetveni kombajn - 4 m combine harvester - 4 m
prevoz transport	prikolica – 3000 kg trailer – 3000 kg

2 REZULTATI IN RAZPRAVA

Neposredno dovedena energija pri pridelavi oljne ogrščice je znašala 5346,43 MJ/ha (2), kar pomeni 22,8% celotne dovedene energije. Za opravila na polju je bilo porabljenih 80,6% omenjene dovedene energije, 19,4% energije pa za prevoz. Pri delih na polju je bilo največ energije porabljene za oranje, in sicer 1419,66 MJ/ha ali 33,0% (preglednica 2), poraba energije za obdelovanje zemlje (oranje in priprava semenskih gred) pa je znašala 49,1% celotne energije, potrebne za opravila na polju. Ti rezultati so podobni podatkom iz vira [16], po katerem je 55 do 65 odstotokov neposredno porabljene energije na polju povezanih z obdelovanjem zemlje. Kakor poroča vir [17], sta oranje in z njim povezana priprava semenskih gred energijsko izrazito zahtevni opravili, ki terjata največjo dovedeno energijo. Le-to lahko zmanjšamo z uporabo manj zahtevnega obdelovalnega sistema, na primer najmanjšega obdelovanja.

Druga največja poraba energije za delo na polju je povezana z žetvijo, za katero so kmetovalci potrebovali 1042,04 MJ/ha energije (24,2%). Z boljšo izrabo časa lahko bistveno povečamo energijsko donosnost žetve in zmanjšamo stroške energije. Glede na vir [18] lahko izrabo časa definiramo kot zmožnost, da neko dejavnost izvedemo ob času, ko pričakujemo optimalno kakovost in kolikost pridelka.

2 RESULTS AND DISCUSSION

The direct inputs in the rapeseed production were 5346.43 MJ/ha (Eq. 2), and this represented 22.8% of the total energy inputs. From this input, 80.6% of the energy was spent in field operations and 19.4% in transport. The greatest energy consumer of field operations was ploughing, with 1419.66 MJ/ha or 33.0% (Table 2), while soil tillage (ploughing and seedbed preparation) consumed 49.1% of the total energy requirement for the field operations. These results are close to the data of [16], who reported that 55-65% of the direct field energy consumption should be accounted for by the soil tillage. According to [17], ploughing and the associated seedbed preparation were very energy-intensive operations and required the highest energy inputs. These can be reduced by adopting less-intensive tillage systems, such as a minimum tillage system.

The second-ranking energy consumer in the field operation was harvesting, with 1042.04 MJ/ha (24.2%). The energy productivity of harvesting can be substantially increased, and the energy costs decreased by an improvement in timeliness. According to [18], timeliness can be defined as the ability to perform an activity at such a time when the quality and quantity of the product are optimized.

Preglednica 2. *Neposredno dovedena energija za dela na polju pri pridelavi oljne ogrščice*
Table 2. *Direct energy inputs for field operations in rapeseed production*

Opravila na polju Field operation	Delovna učinkovitost Work rate ha/h	Poraba goriva Fuel consumption L/ha	Dovedena energija Energy input MJ/ha
oranje ploughing	0,32	29,7	1419,66
priprava semenskih gred seedbed preparation	2,17	12,3	587,94
uporaba gnojil-3 x fertilizer application-3 times	4,80	8,7	415,86
uporaba pesticidov -2 x pesticide application-2 times	4,32	5,6	267,68
vnos pesticidov pesticide incorporation	2,82	8,1	387,18
setev sowing	1,08	3,9	186,42
žetev harvesting	1,20	21,8	1042,04
skupaj total		90,1	4306,78

Izračun stroškov energije, potrebne za prevoz, vezan na povprečno veliko polje (5,2 ha), je temeljil na desetih vožnjah traktorja z ustreznimi kmetijskimi orodji, eni vožnji žetvenega kombajna od kmečkega poslopja do polja in nazaj, ene vožnje traktorja s prikolico za potrebe prevoza materiala na polje in šest voženj traktorja s prikolico za potrebe prevoza oljne ogrščice od polja do skladišča. Za potrebe prevoza je znašala celotna dovedena energija na hektar 1039,65 MJ/ha ali 19,4% neposredno dovedene energije za potrebe pridelave oljne ogrščice (preglednica 3).

Calculation of the transportation energy costs for an average field (5.2 ha) was based on ten movements of the tractor with implements and one movement of a combine harvester from the farmyard to the field and back, one movement of the tractor with a trailer for the transportation of the input materials to the field and six movements of the rapeseed yield by tractor with the trailer from the field to the storage facility. The total energy input in transport per hectare was 1039.65 MJ/ha, 19.4% of the direct energy inputs in the rapeseed production (Table 3).

Preglednica 3. *Neposredno dovedena energija za potrebe prevoza za povrečno polje na hektar*
Table 3. *Direct energy inputs in the transport for an average field and per hectare*

Mehanizacija Machinery	Celotna razdalja Total distance km	Povprečna hitrost Average speed km/h	Celotna poraba goriva Total fuel consumption L	Celotna dovedena energija Total energy input MJ	Dovedena energija na hektar Energy input per hectare MJ/ha
traktor + orodje tractor + implement	72,0	16,5	22,7	1085,06	208,67
traktor + prikolica tractor + trailer	268,8	19,2	82,6	3948,28	759,28
žetveni kombajn combine harvester	7,2	10,6	7,8	372,84	71,70
skupaj total			113,1	5406,18	1039,65

Poznamo ukrepe, s katerimi lahko prihranimo energijo, potrebno za kmetijski prevoz: primerna izbira traktorja in prikolice, njuno pravilno vzdrževanje, primeren način vožnje in dobro načrtovanje, s katerim lahko zmanjšamo število posameznih voženj. Pomembni so tudi še naslednji dejavniki: uporaba koncentriranih gnojil in pesticidov, odstranitev vlage iz materiala, ki ga prevažamo, odstranitev odpadkov, ki jih ni treba prevažati [19].

Posredno dovedeno energijo potrebno za mehanizacijo smo izračunali iz teže vse uporabljene mehanizacije, njene dobe trajanja in delovnega časa na hektar (preglednica 4), in je znašala 5,2% celotne dovedene energije. Največ dovedene energije je bilo potrebne za žetveni kombajn (41,8%) in za traktor (37,4%), medtem ko so vsi preostali stroji porabili le 20,8% energije za mehanizacijo. Uporaba velikih orodij je do neke mere pripomogla k večji izrabi goriva in manjšemu naporu kmetovalcev, vseeno pa so se stroški energije v tem primeru povečali [20]. Količino dovedene energije za potrebe kmetijske mehanizacije je zato mogoče zmanjšati le s primernim vzdrževanjem, s katerim dosežemo daljšo dobo trajanja strojev in povečamo površino, ki jo stroji letno obdelajo.

Preglednica 5 prikazuje vrednosti dovedene energije za potrebe gnojil, pesticidov in semen. Največ

Some measures to save energy in agricultural transportation could include the proper choice of tractor and trailer, their proper maintenance, good driving habits and good planning to reduce the number of trips. This is also important in relation to energy savings in transport, the use of concentrated fertilizers and pesticides, the removal of moisture from commodities and the elimination of residues that do not need to be transported [19].

The indirect energy input for machinery was calculated from the weight of the used machinery, its working life and the working time per hectare (Table 4), and it was found to be 5.2% of the total energy input. The largest energy input was for the combine harvester (41.8%) and the tractor (37.4%), while all the other machines used only 20.8% of the energy for the machinery. Using larger equipment provides a degree of improved fuel efficiency and reduced labor input, but in general, energy costs increase [20]. So, the energy input for farm machinery can be reduced by proper maintenance, for a longer working life of the machines, and a greater acreage on which it is used annually.

The energy inputs for fertilizers, pesticides and seeds are presented in Table 5. The major

Preglednica 4. *Dovedena energija za potrebe kmetijske mehanizacije*
Table 4. *Energy inputs for farm machinery*

Mehanizacija Machinery	Teža Weight kg	Doba trajanja Working life h	Dovedena energija Energy input MJ/h	Obratovalni čas Working time h/ha	Dovedena energija Energy input MJ/ha
traktor tractor	3980	12000	47,66	9,50	452,77
rigolni plug mouldboard plough	450	2000	32,33	3,13	101,19
kombajn za obdelovanje grede seedbed implement	980	2000	70,41	0,46	32,39
centrifugalni razdeljevalnik centrifugal distributor	250	1500	23,95	0,63	15,09
razpršilnik field sprayer	325	1500	31,14	0,46	14,32
kultivator field cultivator	375	2000	26,94	0,36	9,70
sejalni stroj planter	435	1500	41,67	0,93	38,75
žetveni kombajn combine harvester	11000	3000	526,90	0,96	505,82
prikolica trailer	1250	12000	14,97	2,69	40,27
skupaj total					1210,30

Preglednica 5. *Dovedena energija za potrebe gnojil, pesticidov in semen*
Table 5. *Energy inputs for fertilizers, pesticides and seeds*

Material	Uporabljena količina Application rate kg/h	Energijska vrednost Energy value MJ/kg	Dovedena energija Energy input MJ/ha
gnojila / fertilizers			
N gnojilo N fertilizer	138	78,1	10777,80
P_2O_5 gnojilo P_2O_5 fertilizer	117	17,4	2035,80
K_2O gnojilo K_2O fertilizer	176	13,7	2411,20
skupaj / total			15224,80
pesticidi / pesticides			
Devrinol	2,7	171,4	462,78
Triflurolex	1,5	171,4	257,10
skupaj / total			719,88
semena / seeds			
različica Bristol Bristol variety	4,6	200,0	920,00

posredno dovedene energije je bilo porabljene za gnojila, tj. 15224,80 MJ/ha (64,9% celotne dovedene energije) in največji delež te energije je bil potreben za dušikovo gnojilo (70,8% celotne energije za gnojila in 46,0% celotne dovedene energije). Glede na vir [20] je skoraj tretjina vse energije, porabljene v kmetijstvu, namenjena dušikovemu gnojilu. Menimo, da ni mogoče bistveno zmanjšati rabo gnojil, ker bi to vodilo do zmanjšanega pridelka [8]. Stroške energije za potrebe mineralnih gnojil bi lahko zmanjšali s povečano uporabo komposta ter trave in detelje pri kolobarjenju posevkov [21].

Vrednost dovedene energije, potrebne za pesticide, je znašala 719,88 MJ/ha, kar pomeni 3,1% celotne dovedene energije. Celotna energija, potrebna za pesticide, je manjša od drugih vrednosti dovedene energije, je pa vloga pesticidov zelo pomembna, saj bi bil pridelek močno poškodovan, če ne bi pravilno izvajali nadzora nad plevelom, škodljivci in boleznimi. Zato pri načrtovanju zaščite pridelkov zmanjšanje energije ne sme biti glavno vodilo, saj so okoljski in gospodarski vidiki v tem primeru bolj pomembni [22].

Dovedena energija, potrebna za sejanje semen, je bila 920 MJ/ha (uporabljena količina je bila 4,6 kg/h), kar je 3,9% celotne dovedene energije. V tem primeru ne moremo pričakovati kakršnegakoli znatnega zmanjšanja stroškov energije, saj je bila pri prejšnjih različicah oljne ogrščice uporabljena količina precej večja, 8 do 10 kg/h [23].

Energija, potrebna za delo kmetovalcev, pomeni neznaten del dovedene energije z deležem, ki je bil manjši od 0,1% celotne dovedene energije (preglednica 6).

indirect energy input was for fertilizers, 15224.80 MJ/ha (64.9% of total energy inputs), and the greatest part of this input was for nitrogen fertilizer (70.8% of fertilizers and 46.0% of total energy inputs). According to [20], nearly one-third of all the energy used in agriculture is for nitrogen fertilizer. However, a significant reduction in the use of fertilizers is not considered feasible as it would decrease production yields [8]. Reducing energy costs for mineral fertilizers could be possible with the increased utilization of manure and grass-clover crops as part of crop rotation [21].

The energy input for pesticides was 719.88 MJ/ha, which was 3.1% of the total energy input. The overall energy required for pesticides is lower than other inputs, although it is of great significance because the yield can be seriously affected if the control of weeds, pests and disease is not performed properly. Therefore, energy reduction is not the main factor to be taken into account in planning crop protection, because environmental and economic considerations are more important [22].

The energy input for seeds was 920 MJ/ha (the application rate was 4.6 kg/h), 3.9% of the total energy input. In this input, no significant reduction of energy costs could be expected, because earlier rapeseed varieties had much greater application rates, 8 to 10 kg/h [23].

Labor was an insignificant energy input, with less than 0.1% of the total (Table 6). The greatest part of labor was taken up by ploughing

Preglednica 6. *Dovedena energija za potrebe dela kmetovalcev pri pridelavi oljne ogrščice*
Table 6. *Energy inputs for labor in rapeseed production*

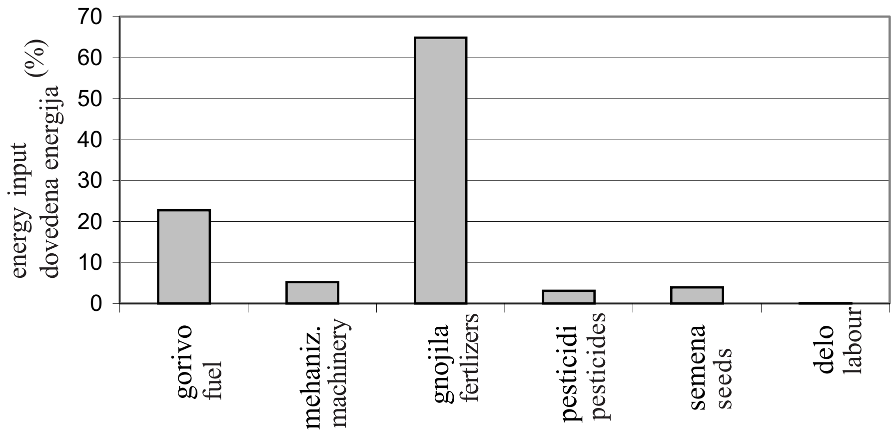
Opravila Operation	Delovni čas Working time h/ha	Dovedena energija Energy input MJ/ha
oranje ploughing	3,13	7,20
priprava semenskih gred seedbed preparation	0,46	1,06
uporaba gnojil-3 x fertilizer application-3 times	0,63	1,45
uporaba pesticidov -2 x pesticide application-2 times	0,46	1,06
vnos pesticidov pesticide incorporation	0,36	0,83
setev sowing	0,93	2,14
žetev harvesting	0,96	2,21
prevoz transport	2,69	6,18
skupaj total	9,62	22,13

Največ energije je bilo potrebne za oranje (32,5%) in prevoz (27,9%). Celotna posredno dovedena energija pri pridelavi oljne ogrščice je znašala 18.097,11 MJ/ha (enačba 3). Znesek celotne dovedene energije pri pridelavi oljne ogrščice pa je tako znašal 23.443,54 MJ/ha (enačba 1). Slika 1 kaže deleže posameznih vnosov energije.

Preglednica 7 prikazuje vso odvedeno energijo pri pridelavi oljne ogrščice. Povprečen pridelek vseh kmetij, dosežen v raziskovalnem obdobju, je bil 3.195 kg/ha (s 7,0% deležem vlage), kar je zelo dobro, saj je bil v istem obdobju povprečni pridelek oljne ogrščice na Hrvaškem 2.140 kg/ha [24].

(32.5%) and transport (27.9%). The total indirect energy input in rapeseed production was 18097.11 MJ/ha (Eq. 3). So, the total energy input in rapeseed production was 23443.54 MJ/ha (Eq. 1). Figure 1 shows the percentage of all the energy inputs.

The energy outputs in rapeseed production are presented in Table 7. The average yield of all the farms in the investigated period was 3195 kg/ha (with a 7.0% moisture content), which was very good, because the average yield of rapeseed in Croatia in the same period was 2140 kg/ha [24]. Some [25]



Sl. 1. *Dovedena energija pri pridelavi oljne ogrščice*
Fig. 1. *Energy inputs in rapeseed production*

Preglednica 7. *Odvedena energija pri pridelavi oljne ogrščice*
Table 7. *Energy outputs in rapeseed production*

Pridelek Item	Donos Yield kg/ha	Energijska vrednost Energy value MJ/kg	Odvedena energija Energy output MJ/ha
semena / seeds	3195		
olje / oil	1154	37,6	43390,40
krma / meal	2041	15,0	30615,00
skupaj / total			74005,40
odpadki / crop residues	5130	12,5	64125,00

Nekateri [25] menijo, da je povečanje pridelka mogoče, če uporabljamo nove hibride oljne ogrščice, ki so zdaj že na voljo. Povprečni delež olja v semenih je bil 42,5%, pri 85% ekstrakciji olja je ta pridelek vseboval 1.154 kg olja in 2.041 kg krme. Pri takšnem pridelku je celotna odvedena energija znašala 74.005,40 MJ/ha (4). Če odpadke pridelka oljne ogrščice obravnavamo kot energijo biomase, je njihova odvedena energija znašala 64.125 MJ/ha.

Če upoštevamo oboje, olje ogrščice, namenjeno proizvodnji biodizla, in krmo za živali, je energijsko razmerje pri pridelavi oljne ogrščice bilo 74.005,40/ 23.443,54 = 3,16 (5). Neto energijski prirastek pri pridelavi oljne ogrščice je bil 74.005,40 - 23.443,54 = 50.561,86 MJ/ha ali 50,56 GJ/ha (6). Energijska donosnost pri pridelavi oljne ogrščice za potrebe biodizla je bila 1.154/ 23,44 = 49,23 L/GJ (7). Te vrednosti energijske izravnave so zelo podobne evropskim povprečnim vrednostim [15], ki kažejo na energijsko razmerje 3,15 in neto energijski prirastek 49,27 GJ/ha pri pridelavi oljne ogrščice, pri čemer običajno v izračunu niso upoštevani odpadki pridelka. Vir [26] je poročal o neto energijskem prirastku 43,4 GJ/ha ter o razmerju med dovedeno in odvedeno energijo 2,8, ugotovljenem pri pridelavi oljne ogrščice, iz navedenega je sklepal, da v prihodnje biogoriva lahko obravnavamo kot pomembne vire obnovljive energije. V primerjavi z drugimi postopki proizvodnje goriva iz obnovljivih kmetijskih virov, oljna ogrščica ponuja dobro razmerje med dovedeno in odvedeno energijo in precejšen energijski donos. Glede na vir [27] je učinkovita raba energije tudi eden od pogojev za razvoj trajnostnega kmetijstva, saj omogoča denarne prihranke, ohranjanje fosilnih virov in zmanjšano onesnaževanje.

3 SKLEP

Oljno ogrščico imamo lahko za primeren vir priprave biodizla na Hrvaškem, saj v primerjavi z energijo, potrebno za njeno pridelavo, omogoča

suggested that an increase in yield seems possible by using new rapeseed hybrids, which are now available. The average oil content in the seed was 42.5%, and with an extraction rate of 85%, this yield is made up of 1154 kg of oil and 2041 kg of meal. According to this yield, the total energy output was 74005.40 MJ/ha (Eq. 4). If we consider rapeseed crop residues as the biomass energy, their energy output was 64125 MJ/ha.

Considering the rapeseed oil for biodiesel and the meal for animal feed, the energy ratio of the rapeseed production was 74005.40/23443.54 = 3.16 (Eq 5). The net energy gain of the rapeseed production was 74005.40 - 23443.54 = 50561.86 MJ/ha, or 50.56 GJ/ha (Eq. 6). The energy productivity of the production of rapeseed oil for biodiesel was 1154/23.44 = 49.23 L/GJ (Eq. 7). These values of energy balance are very close to the average values for Europe [15]. They showed an energy ratio of 3.15 and a net energy gain of 49.27 GJ/ha in rapeseed production, normally without using crop residues in the calculation. Source [26] reported a net energy gain of 43.4 GJ/ha, and an energy output/input ratio of 2.8 in the production of rapeseed oil, and concluded that biofuels can be considered as an important source of renewable energy in the future. In comparison with other processes for fuel production from renewable agricultural resources, rapeseed oil gives a good output/input energy ratio and a substantial net energy gain. According to [27], efficient energy use is also one of the conditions for sustainable agriculture because it allows financial savings, the preservation of fossil-fuel resources and a reduction in air pollution.

3 CONCLUSION

Rapeseed can be considered for biodiesel production in Croatia because it provides a positive energy return compared with the energy used to energijski donos. Ko smo raziskovali možnosti za uporabo oljne ogrščice za izdelavo biodizla in za krmo živali, je bilo energijsko razmerje pri pridelavi oljne ogrščice 3,16, neto energijski domos 50,56 GJ/ha in energijska donosnost 49,23 L/GJ. Večina dovedene energije je bila potrebna za rabo gnojil, 64,9%, ter za porabo goriva, 22,8%. Hkrati smo ugotovili, da je bil za delo kmetovalcev potreben le neznaten delež energije, tj. manj ko 0,1% celotne dovedene energije. Ker bi zmanjšanje uporabe gnojil povzročilo tudi zmanjšan pridelek, je treba najprej iskati možnosti varčevanja z dovedeno energijo pri porabi goriva, potrebnega za opravila na poljih, predvsem pri obdelovanju zemlje, in pri prevozu. Pridelava oljne ogrščice za proizvodnjo biodizla je hrvaškim kmetovalcem dobra zamenjava in lahko Hrvaški prinese gospodarske, okoljevarstvene in energijske koristi.

produce it. Considering rapeseed oil for biodiesel and meal for animal feed, the energy ratio of the rapeseed production was 3.16, the net energy gain was 50.56 GJ/ha, and the energy productivity was 49.23 L/GJ. The major energy inputs were fertilizers, with 64.9%, and fuel consumption, with 22.8% of the total energy input. On the other hand, labor was an insignificant energy input, with less than 0.1% of the total. Because the reduction of fertilizers would decrease production yields, the reduction of energy input might first be possible with regard to fuel consumption for field operations, especially in soil tillage and transport. The production of rapeseed oil for biodiesel can be a good alternative for Croatian farmers and can have economic, environmental and energy benefits for Croatia.

4 VIRI
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Naslov avtorjev: prof. dr. Dubravko Filipović
prof.dr. Tajana Krička
Univerza v Zagrebu
Agronomska fakulteta
Svetošimunska 25
10000 Zagreb, Hrvaška
dfilipovic@agr.hr
tkricka@agr.hr

Authors' Address: Prof. Dr. Dubravko Filipović
Prof. Dr. Tajana Krička
University of Zagreb
Faculty of Agriculture
Svetošimunska 25
10000 Zagreb, Croatia
dfilipovic@agr.hr
tkricka@agr.hr

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Poročila - Reports

Odmevi iz varilske stroke - Welding News

Dne 26. maja 2005 sta Društvi za varilno tehniko Ljubljane in Dolenjske (v ustanavljanju) v Dvorani za komunikacije podjetja Revoz priredili sedmi Dan varilne tehnike. Da je bil dan varilne tehnike umeščen v industrijsko okolje, je več razlogov:

- Tam je zbranih največ ljudi, ki se dnevno ukvarjajo z reševanjem varilskih problemov in so tudi živo zainteresirani za seznanjanje z novostmi na področju varilnih postopkov, opreme, proizvodnih sistemov, izobraževanja ter zaščite in varnosti.
- Želeli smo izpostaviti spremembe, do katerih je prišlo v zadnjih letih na področju varjenja v Republiki Sloveniji, kar je še posebej izrazito prav na Dolenjskem. Gre za vztrajno zmanjševanje deleža težkih varjencev in števila ročnih varilcev, ob sočasnem povečevanju uporabe avtomatiziranih in robotiziranih postopkov varjenja v hitro razvijajoči se množični proizvodnji. Prav v podjetju Revoz se te spremembe kažejo v kar največji meri.
- Sledili so zamisli, ki je bila nakazana in deloma udejanjena že na četrtem Dnevu varilne tehnike, namreč, da se po možnosti taka prireditev izvede v podjetju, kjer so postopki varjenja med ključnimi izdelovalnimi tehnologijami. V nasprotju z izvedbo 4. Dneva, ko je predstavitev v zborniku objavljenih referatov potekala na Fakulteti za pomorstvo in promet v Portorožu in je razstavno-družabni del potekal v tovarni Tomos v Kopru, se je na 7. Dnevu celotno dogajanje odvijalo na enem mestu.
- Tako smo obšli načrtovane blokade nekaterih kolegov varilcev, ki jim ob vsesplošnem "uspešnem lastninjenju" manjka samo še podreditev družtvene dejavnosti.

Da so kolegi v podjetju Revoz ob vsej prezasedenosti, ki jo imajo s pripravami na zagon proizvodnje novega avtomobila, bili pripravljeni sprejeti ponujeno pobudo obeh društev, sta v največji meri zaslužna član Industrijske direkcije in vodje kakovosti tovarne avtomobilov Revoz mag. Dušan Dimovski in vodja oddelka zunanjih komunikacij Nevenka Bašek.

Prav zato so uvodne besede na sedmem Dnevu varilne tehnike pripadle mag. Dušanu Dimovskemu, ki se je v imenu vodstva podjetja in v svojem imenu zahvalil organizatorjem za popularizacijo varilske stroke in za skrb pri prenosu znanja iz akademskih krogov v industrijsko okolje, saj je v njihovem podjetju varjenje ena od pomembnejših proizvodnih tehnologij. Ta razmišljanja je zaokrožil z naslednjimi besedami:

"Povezovanje razvojno-izobraževalnih institucij in industrije je postala nuja in edina možna pot, če želimo dvigniti kakovostno raven obeh segmentov. Raziskave, ki so same sebi namen in nimajo nikakršne povezave s potrebami industrije, so nekoristne in kot takšne obsojene na neuspeh. Revoz je tovarna, ki je in želi biti še bolj vpeta v slovenski prostor. Kar se tiče kakovosti, storilnosti in prilagodljivosti, sodimo v sam svetovni vrh avtomobilskih proizvajalcev. Na ta način lahko naše pojavljanje v slovenskem prostoru in povezovanje z različnimi znanstveno-raziskovalnimi skupinami pomeni velik izziv slovenskim strokovnjakom v smislu dvigovanja njihove in naše strokovne rasti. Ravno zato smo v podjetju Revoz z veseljem sprejeli pobudo DVT Ljubljana, da v našem podjetju organiziramo srečanje strokovnjakov varilske stroke iz Slovenije."

Predsednik DVT Ljubljana se je zahvalil mag. Dimovskemu za izrečene besede, predvsem pa vsem iz tovarne Revoz, ki so tako ali drugače pomagali pri izvedbi Dneva varilne tehnike, ter v spomin na ta dogodek tovarni Revoz predal zahvalno plaketo Društva za varilno tehniko Ljubljana in varjeno skulpturo starodobnega Renaultovega avtomobila avtorja Aleksandra Arharja. V zahvalo za pomoč pri organizaciji dneva in postavitvi razstave, je predsednik sekcije Plamen mag. Janez Kranjc vodji oddelka zunanjih komunikacij Nevenki Bašek in Ireni Vidic izročil dve varjeni marjetici avtorja Jožeta Lisca.

Z zahvalnimi plaketami in varjenimi skulpturami avtomobilov novejše generacije avtorja Ljuba Malenška so bili nagrajeni tudi preostali pokrovitelji: Cimos Koper, Fotokop Komen, Gorenje - Indop Velenje, MDM Ljubljana, Motoman Ribnica, Techning Hajdina in Šolski center iz Novega mesta.

V nadaljevanju je v imenu Ministrstva za visoko šolstvo, znanost in tehnologijo spregovoril mag. Jože Zrimšek:

"Dovolite, da vam ob tej priložnosti tudi predstavim Ministrstvo za visoko šolstvo, znanost

Predsednik DVT Ljubljana izroča članu Industrijske direkcije in vodji kakovosti tovarne mag. Dušanu Dimovskemu spominsko darilo

Pozdravni nagovor prof. dr. Viktorja Prosenca udeležencem Dneva varilne tehnike

in tehnologijo, ki je z reorganizacijo državne uprave nastalo:

- *z ločitvijo bivšega Ministrstva za šolstvo, znanost in šport na Ministrstvo za šolstvo in šport ter Ministrstvo za visoko šolstvo, znanost in tehnologijo,*
- *s pridružitvijo Direktorata za tehnologijo z Ministrstva za gospodarstvo ter*
- *s pridružitvijo Direktorata za informacijsko družbo z bivšega Ministrstva za informacijsko družbo.*

Ena od prednostnih nalog novega Ministrstva za visoko šolstvo, znanost in tehnologijo je tudi ta, da skrbi za hitrejši pretok znanja med izobraževalnimi in raziskovalnimi inštitucijami in gospodarstvom, kar sodi v ukrepe za pospeševanje tehnološkega razvoja in inovativnosti predvsem v malih in srednjih podjetjih. Predvsem z navedenega vidika Ministrstvo za visoko šolstvo, znanost in tehnologijo pozdravlja in podpira dogodke kakršen je Dan varilne tehnike, ki sta ga organizirala Društvo za varilno tehniko Ljubljana in Društvo za varilno tehniko Dolenjske v ustanavljanju. Varjenje pomeni enega od ključnih postopkov v podjetjih s področja kovinske industrije, zato je ključnega pomena, da zaposleni med drugim obvladajo tehnične in tehnološke novosti na področju varilnih tehnik, varilnih naprav in avtomatov, novih materialov in zaščitnih sredstev ter zagotavljanja potrebne kakovosti in standardizacije.

Želim, da bi vsi udeleženci prejeli čim več znanja in izkušnje iz strokovnih predavanj v okviru dneva varilne tehnike in nova spoznanja kar najbolje uporabili za tehnološki napredek in izboljšave v vsakdanji praksi v organizacijah in podjetjih, iz katerih prihajate. Tehnološki napredek in razvoj sta po mojem trdnem prepričanju ob ustrezni praksi med ključnimi pogoji, ki vodijo k ustvarjanju večje dodane vrednosti v proizvodnji in posledično k večji gospodarski rasti."

V imenu dekana Fakultete za strojništvo je navzoče pozdravil izr. prof. dr. Janez Diaci:

"Tokratno srečanje je že sedmo po vrsti in sedem je število, ki ima v mnogih kulturah prav poseben - simboličen pomen. Če ta simbolika kaj velja tudi v tem primeru, potem lahko z gotovostjo rečemo, da se je to srečanje popolnoma uveljavilo v slovenskem prostoru kot najpomembnejši forum za izmenjavo izkušenj, mnenj in pogledov na stanje in smeri razvoja varilne tehnike.

Poseben čar teh srečanj je v tem, da se na njih srečujemo in družimo strokovnjaki z vseh področij varilne tehnike: raziskovalci iz inštitucij znanja, razvojniki in tržniki iz podjetij, ki proizvajajo tovrstno opremo in uporabniki te opreme. Čeprav nam sodobne tehnologije omogočajo široko izbiro različnih vrst komunikacije na daljavo, pa ostajajo osebni stiki nenadomestljivi tudi na področju izmenjave strokovnih vsebin. Takšen forum, kot so Dnevi varilne tehnike, omogoča najbolj neposredno izmenjavo svežih informacij med ljudmi, ki živijo z varilno tehniko in to mogoča hiter prenos dosežkov iz raziskovalnih in razvojnih laboratorijev v prakso, pa tudi dosežkov iz ene prakse v drugo. V času hitrih in korenitih sprememb, ki ga živimo, postaja takšen hiter in neoviran prenos pravih informacij vse bolj pomemben.

V veliko zadovoljstvo nam je, da je bila zamisel o tovrstnem srečanju, ki se je rodila in *razvila v akademskem okolju (prvi dve srečanji sta bili na Univerzi v Mariboru oz. na Univerzi v Ljubljani), tako toplo sprejeta tudi s strani strokovnjakov iz gospodarstva. Današnji Dan varilne tehnike je tako že drugi, ki poteka v industrijskem okolju. To je še posebej dragoceno za nas, ki prihajamo iz akademskega okolja, da imamo priložnost spoznati probleme in praktične rešitve prav v okolju, kjer nastajajo. Iz tega se nam porajajo vzpodbude in izzivi za nadaljevanje našega raziskovalnega in tudi pedagoškega dela.*

Dovolite mi, da zaključim ta pozdravni nagovor z mislijo, ki se mi zdi pomembna za vse nas kot skupnost, odgovorno za razvoj naše stroke. Z mislijo, ki se mi zdi še posebej aktualna v času, ki ga živimo.

Opažamo, da se v Sloveniji mladi ljudje vse premalo odločajo za izobraževanje in vstop v tehnične poklice. To za razvoj naših strok ni dobro. Zato mislim, da mora postati naša trajna skrb, da predstavljamo ob vsaki priložnosti, ki se ponudi, tehniške poklice v pravi luči, zlasti z vidika možnosti zaposlitve, razvoja profesionalne kariere in možnosti osebne rasti. To je potrebno še posebej za to, da premagamo nekatere stereotipe, ki so se oblikovali v preteklosti. Tehniški poklici se, kot tudi vse drugo, v današnjem času vse bolj spreminjajo. Vse več je v njih potrebnega znanja in kreativnega dela, vse manj pa je (pogojno rečeno) "umazanega ročnega dela".

Iščimo torej poti, po katerih bomo pripeljali v naše vrste več mladih ljudi. Njihova energija, zagnanost in kreativnost so ključnega pomena za razvoj naših strok."

Posebno prijazen nagovor z zgodovinskim orisom dogajanj na področju varilne tehnike je imel prof. dr. Viktor Prosenc, upokojeni profesor varilskih predmetov in nekdanji dekan Fakultete za strojništvo. Dejstvo, da je imel priložnost spregovoriti pred predstavniki tovarne Revoz v Sloveniji, je izrabil tudi za to, da je pohvalil kakovost, zanesljivost in udobnost avtomobilov te tovarne, saj jih vozi že več ko 40 let. Začel je z Renaultom R10 v sredini šestdesetih let, največ kilometrov je prevozil z Renaultom R12, najdlje pa vozi R5.

Ob tej priložnosti se mu je sekcija Plamen pri DVT Ljubljana oddolžila za njegov odziv na vabilo in mu podarila varjeno skulpturo Drevo in knjiga avtorja Jožeta Lisca, ki je darilo izročil osebno ter obdarjencu in prisotnim razložil sporočilo umetnine: *"Drevo se žalosti izgubi listja in radosti svojim plodovom!"*

Sledile so zgoščene predstavitve 32 strokovnih referatov, ki so v Zborniku Dneva varilne tehnike zbrani po tematskih sklopih:

- Varjenje v avtomobilski industriji - 5 prispevkov
- Varilske tehnologije - 5 prispevkov
- Varivost - 12 prispevkov
- Spremljanje in zagotavljanje kakovosti - 8 prispevkov ter
- Zaščita in varnost - 2 prispevka.

V zborniku formata A4 sta predstavljeni še dve poljudni poročili: Poročilo ob petletnem jubileju sekcije umetnostnih varilcev Plamen avtorja Janeza Kranjca in Pogled na restavriranje starodobnih avtomobilov avtorjev Ivana Polajnarja in Janeza Zajca.

V avli razstavnega prostora se je z izdelki, panoji in prospekti predstavilo 30 domačih razstavljavcev varilne opreme in naprav, med njimi ABB, Akrapovič, Balder, Brinox, Casagrande, Castolini, Elektrode, Esab Flach, EV, Fotona, IMP - NDT, Ingvar, IPRO, Institut za varilstvo, Iskra varjenje, Iskra zaščite, Kočevar Thermotron, Linde plini, Litostroj, Messer, Metalna, Pilih TOX, Sanks, Spectro Martini, Team Trade, Trimo, Varstroj, VIRS, WELD, Zavar. Med temi razstavljavci gre izpostaviti svetovno znanega slovenskega proizvajalca zaščitnih varilskih stekel s tekočimi kristali Balder, katerega osnovni izdelek je v celoti plod domačega znanja.

Za večino udeležencev so bili najbolj odmeven del spremljajočega dogajanja ogledi proizvodnje, ki so potekali hkrati s predavanji v več manjših skupinah, pod vodstvom in z izčrpnimi strokovnimi obrazložitvami zastopnikov Revoza.

Poseben pečat celotnemu dnevu varilne tehnike je dala sekcija likovnih oblikovalcev v kovini Plamen z namensko narejenimi skulpturami za nagrado pokroviteljem (miniaturne izvedbe avtomobilov) in s postavljeno razstavo varjenih skulptur dveh članov sekcije: Ljubota Malenška in Jožeta Lisca.

Ob koncu dneva je bila še okrogla miza s celovito analizo dogodkov v zvezi z izvedbo Dneva varilne tehnike in kritično osvetlitvijo dogajanj v stroki. Ob tej priložnosti je predsednik društva DVT izpostavil tudi poslanstvo, ki ga z druženjem strokovnih kadrov in s seznanjanjem udeležencev o trenutnem stanju in trendih razvoja v varilski stroki opravlja sedaj že tradicionalni Dan varilne tehnike.

Celovit oris pogleda na dogajanje v okviru Dneva varilne tehnike je podal zastopnik podjetja Cimos g. Igor Puhar:

"Vsebina referatov kaže na to, da je trend bodočega razvoja varjenja v vse večji uporabi laserjev in hibridnih postopkov, kot je npr. kombinacija laserskega varjenja z obločnim varjenjem. Hvalevredno je, da obstajajo visokotehnološke uporabe tudi v Sloveniji, kar je moč videti tudi na montažni liniji podjetja Revoz, kjer v laserski celici varijo notranjo in zunanjo stranico vrat avtomobila Renault Clio.

Zanimivi so bili tudi referati, ki niso obravnavali osrednje tematike varjenja v avtomobilski industriji. Skoraj vse predstavljene rešitve in izkušnje se namreč lahko prenese tudi v avtomobilsko proizvodnjo.

Pri ogledu proizvodnje je bilo zlasti zanimivo videti varilno celico proizvajalca ABB s štirimi roboti, v katerih se vari dno avtomobila.

Na dnevu varilne tehnike smo uspešno izmenjali izkušnje s kolegi tudi z drugih varilnih področij, tako da bomo tisti, ki delamo v podjetjih, ki so dobavitelji za potrebe avtomobilske industrije, izšli iz tega srečanja bogatejši za nova spoznanja.

Prepričan sem, da bomo na naslednjem dnevu varilne tehnike lahko kolege seznanili z izsledki raziskovalne naloge neprekinjenega nadzora uspešnosti kakovosti pretaljevanja, ki smo se je v podjetju Cimos ravnokar lotili. Prepričan sem tudi, da bomo te izsledke koristno uporabili zlasti pri robotiziranem varjenju."

Da je bil 7. dan varilne tehnike resnično poseben, gre zasluga tudi članu organizacijskega odbora Štefanu Oreškoviču, ki je o dogajanju obvestil Radio in televizijo Slovenije. Posneta in predvajana so bila dogajanja ob Dnevu varilne tehnike in objavljeni intervjuji s predstavnikom tovarne Revoz mag. Dušanom Dimovskim, predsednikom Društva za varilno tehniko Ljubljana dr. Ivanom Polajnarjem in predstavnikom Fakultete za strojništvo prof. dr. Janezem Diacijem. O samem dogajanju pa je poročal tudi *Dolenjski list*.

Ivan Polajnar, predsednik DVT Ljubljana

Osebne vesti - Personal Events

80 let dr. Jožeta Hlebanje, zaslužnega profesorja Univerze v Ljubljani - 80th Anniversary of Dr. Jože Hlebanja, Professor Emeritus of the University of Ljubljana

V šolskem letu 1959/1960 je prof. Franček Kovačec pripeljal na svoje uvodno predavanje iz Strojnih elementov mladega inženirja in nam ga predstavil kot docenta, ki ga bo nadomestil pri predmetu. To je bil moj prvi stik s dipl.inž. Jožetom Hlebanjo. Prof. Kovačec je našo generacijo študentov strojništva v trenutku povsem prepustil novincu. Ker smo bili zadnja generacija pred napovedano reformo študija, nas je bilo v letniku ogromno, iz vse Jugoslavije, veliko kolegov je bilo starejših od učitelja. Bili smo kar malo prestrašeni, kako bomo absorbirali 180 ur težkega predmeta brez izkušenega prof. Kovačca. Že ob koncu semestra dvomov ni bilo več. Doc. Jože Hlebanja je v kratkem času nam v pomoč spisal skripte, nas pripeljal do izpita in nas v velikem številu pritegnil v svojo projektno skupino. Tako smo začeli z dolgoletnim skupnim delom. Za vse nas, ki smo se kakorkoli srečali s prof. Jožetom Hlebanjo, je to sodelovanje obdobje polno dela, uspehov in doživetij. Teh 46 let je prehitro minilo. 80 let pa je še kar nekaj več življenja. Žal so bila ta mladostna leta, ki bi morala biti najlepša, polna strahu, groze in trpljenja.

Jože Hlebanja je bil rojen 31.3.1926 na Srednjem vrhu nad Martuljkom. Osnovno šolo v Kranjski Gori in nižjo srednjo šolo na Jesenicah je končal leta 1940. Takoj se je vpisal na Tehniško srednjo šolo, Oddelek za strojništvo v Ljubljani. Kot Gorenjec in pod nemško okupacijo je v Ljubljani lahko končal samo 1. razred, v letih od 1941 do 1943 pa je 2. in 3. razred končal v Celovcu. Spomladi 1943 je začel aktivno in organizirano sodelovati v NOB, januarja 1944 pa se je pridružil partizanskim operativnim enotam. Nemci so ga v zasedi ujeli, ga zasliševali na Jesenicah in zaprli v Begunje. Ko so partizani junija 1944 minirali sedež gestapa na Bledu, je bil Jože Hlebanja med tistimi sotrpini, ki so jih Nemci že odpeljali na streljanje, pa potem le-to le ni bilo izvedeno. Avgusta so ga odpeljali v Mauthausen, kjer je vse do osvoboditve aktivno deloval med zaporniki v zaščitni enoti taboriščnikov. Junija 1945 se je z glavnim transportom vrnil domov, soorganiziral odbor internirancev Mauthausna, pri katerem sodeluje še dandanes in se takoj vključil v obnovo porušenih Železarn Jesenice. Do poletja 1946 je končal Srednjo tehniško šolo in se v jeseni vpisal na Tehniško fakulteto, Oddelek za strojništvo v Ljubljani. Med študijem je bil med leti 1947 do 1949 demonstrator pri prof. dr. Antonu Kuhlju. Leta 1949 se je priključil projektni skupini pri prof. Frančku Kovačecu in zato odložil študijske obveznosti tako, da je diplomiral šele leta 1952. Postal je asistent za transportne naprave in jeklene konstrukcije. Po treh letih se je kot projektant zaposlil v Metalni Maribor. Podjetje mu je omogočilo prakso in specializacijo v dveh tovarnah na Dunaju, kjer so gradili žičnice. Kot projektant v Metalni je izdelal projekte za veliko število dvigal, mostnih in pristaniških žerjavov, njegova specialnost pa so bile žičnice. Pod njegovim vodstvom in z njegovimi projekti je postala Metalna vodilna za gradnjo žičnic v širši regiji. Nekatere od njegovih konstrukcij obratujejo še dandanes. Z imenom Jožeta Hlebanje so povezane tovorne žičnice v rudnikih Velenje, Šoštanj, v TE Šoštanj, osebna žičnica na Pohorje, Sljeme pri Zagrebu, na Popovo Šapko pri Tetovu, v Sarajevu, smučarske sedežnice in vlečnice v Kranjski Gori so njegovo delo in še in še.

Ko je prof. Jože Hlebanja začel kot mlad docent leta 1960 delati na Fakulteti za strojništvo, je v projektiranje in konstruiranje žičnic pritegnil kar nekaj uspešnih študentov od drugega do četrtega letnika in tudi prve diplome pod njegovim mentorstvom so bile s tega področja. Ker v Sloveniji, še manj pa v Jugoslaviji, to področje ni bilo ne upravno ne normativno urejeno, je dosegel, da je bil ustanovljen Odbor za žičnice. Vodil je skupino raziskovalcev FS in Zavoda za raziskavo materiala, ki je sestavila normative in predvsem tudi definirala načine obveznega preizkušanja žičnic za ugotavljanje zdržljivosti in varnosti.

Pedagoški program Laboratorija za strojne elemente se je pod vodstvom prof. Jožeta Hlebanje začel dopolnjevati z raziskovalnimi programi in eksperimentalnim delom. Znanje, izkušnje in eksperimentalna oprema na fakulteti in v drugih laboratorijih ljubljanske univerze so omogočili, da je raziskovalna skupina pod njegovim vodstvom pričela konkurirati tudi pri javnih sredstvih za raziskovalno delo. Tako je že leta 1968 Sklad Borisa Kidriča financiral v Laboratoriju za strojne elemente, kot eno prvih na fakulteti, večletno raziskovalno nalogo, v kateri naj bi raziskali vklopne razmere na prižemkah žičnic. Z eksperimentalnim delom v laboratoriju je bil utemeljen povsem drugačen način vpetja prižemk, kar so kar hitro sprejeli vsi proizvajalci žičnic.

Prof. dr. Jože Hlebanja je ob obsežnem pedagoškem delu, raziskovalnem in strokovnem delu skupaj z industrijo in za njo leta 1967 končal svojo doktorsko raziskavo "Uporabnost cikličnih krivulj za boke zob zobnikov" in dobil izredno profesuro.

V šestdesetih letih je bila v gradnji nova stavba na Aškerčevi. Prof. dr. Jože Hlebanja je kot član gradbenega odbora opravil obsežno in zahtevno delo usklajevanja prostorskih potreb, opreme in komuniciranja z investitorjem. Novi prostori in kupljena nova raziskovalna oprema so omogočili še boljše in kakovostnejše raziskovalno delo, kar je temelj tudi za podiplomski študij. Prof. Jože Hlebanja je vodil vrsto mladih sodelavcev do magisterija in do doktorata. Nekaj doktorandov je takoj prevzelo odgovorna mesta v industriji, nekaj pa nas je napredovalo na učiteljska mesta v visokem šolstvu. Skupaj so se člani raziskovalne skupine udeležili veliko kongresov doma in v tujini in v osebnih stikih ustvarili dobre povezave z raziskovalnih delom v tujini. Skupaj smo člani njegove skupine prevzeli in uspešno opravili vrsto raziskovalnih in strokovnih nalog doma in v tujini.

Med leti 1973 do 1975 je bil prof. dr. Jože Hlebanja prodekan na FS. Torej prav v času, ko je morala fakulteta prenoviti svoj status in programe pedagoške in raziskovalne dejavnosti. Zahtevno in obsežno delo mu ni preprečilo, da ne bi leta 1976 pridobil redne profesure.

V strokovni javnosti je bilo posebej odmevno prvo mednarodno posvetovanje o konstrukterstvu v Jugoslaviji, ki ga je prof.dr. Jože Hlebanja organiziral leta 1976 v Ljubljani. Na njem je bil potrjen primat FS na področju konstruiranja na vsem jugoslovanskem področju. Prof. dr. Jože Hlebanja je leta 1976 dobil nagrado Sklada Borisa Kidriča, ki dokazuje njegovo vsestransko uspešno delo in pomeni najpomembnejše priznanje širše strokovne javnosti tem rezultatom.

Prof. dr. Jožetu Hlebanju je bilo ob koncu sedemdesetih let poverjeno vodenje raziskovalnega programa Transportne naprave, strojni elementi in konstruiranje, ki ga je financirala Raziskovalna skupnost Slovenije iz javnih sredstev. V okviru tega programa so bili organizirani kongresi, posvetovanja in strokovna srečanja, na katerih so bile definirane osnovne panoge področja in z njimi usklajeni novi dodiplomski in podiplomski programi ter raziskovalna področja. Prof. dr. Jože Hlebanja je za oba pedagoška programa odprl vrsto novih predmetov: Tribologija, Metodika konstruiranja, Osnove konstruiranja, Transportne naprave, Lesnoobdelovalni stroji in Kmetijski stroji.

Pod njegovim vodstvom se je raziskovalni program preoblikoval v raziskovalno polje Konstruiranje, v okviru katerega se raziskovalno in strokovno delo odvija še dandanes in v katerem v pretežni meri vodimo posamezne raziskovalne programe njegovi učenci.

Konec 70. let in vsa 80. leta je prof. dr. Jože Hlebanja vodil obsežno pedagoško-raziskovalno skupino. Značilna zanj je bila velika uspešnost članov tako pri podiplomskem študiju kakor pri raziskovalno strokovnem napredku. Pri prof. dr. Jožetu Hlebanji nas je doktoriralo deset, še veliko večje število podiplomcev pa je magistriralo. Na njegovo pobudo smo se dejavno vključevali v mednarodna združenja IFToMM, VDI, SAE, ASMME in v jugoslovanska, ki jih je soustanovil prof. dr. Jože Hlebanja: JuDEKO, JUMV, JDM itn. Kot izvedenec je sodeloval v tehniških komitejih pri VDI oz. DIN za zobnike in za drsne ležaje, vodil je domač TC itn.

Pod njegovim vodstvom se je Katedra za delovne stroje in transport opremila z moderno raziskovalno opremo, bili smo uspešni na razpisih za javna sredstva, potekal je pospešen podiplomski program, predvsem pa smo lahko z ugledom in znanjem opravili zahtevna strokovna in raziskovalna dela za partnerje iz vse Jugoslavije.

Ko je prof. dr. Jože Hlebanja leta 1991 odšel v pokoj, je na Fakulteti za strojništvo in v Sloveniji pustil urejeno strokovno področje konstruiranja. Imeli smo dogovorjen program razvoja, dogovorjeno področja dela na pedagoškem in raziskovalnem področju, v laboratorijih kakovostno opremo, vpeljano sodelovanje s tujino in z industrijo, pomembne rezultate in velik ugled kot raziskovalci, ki so z aplikacijami vključeni v industrijo.

Univerza v Ljubljani se je na predlog Fakultete za strojništvo zato prof. dr. Jožetu Hlebanji kar malo pozno oddolžila za ves prispevek k razvoju visokega šolstva in znanosti, ko mu je leta 1996 podelila naziv zaslužni profesor Univerze v Ljubljani.

Na strokovnem in pedagoškem področju je sled, ki jo je prof.dr. Jože Hlebanja pustil doma in v tujini globoka in bo dolgo vidna. Vsi, ki smo ga na tej poti srečevali in z njim sodelovali pa zadnja leta, pogrešamo vsakodnevna srečanja, polna prijaznosti, topline in resnosti pri strokovnih zadevah, pa tudi vseh pripetljajev in veselih dogodkov, ki smo jih kot prijatelji doživljali dolgo skupaj in o katerih govori kar nekaj anekdot.

Sklenimo ta zapis z voščilom: “Prof. dr. Jože Hlebanja - dosti zdravja in sreče, še na mnoga leta,” in z željo, da se bomo še velikokrat srečali na fakulteti in zunaj nje.

Prof. dr. Matija Fajdiga, jeseni 2006

Bunshahova nagrada - Bunshah Award

Doc.dr. Bojan Podgornik je kot raziskovalec zaposlen na Fakulteti za strojništvo v Ljubljani, Centru za tribologijo in tehnično diagnostiko, kjer pod vodstvom prof. dr. Jožeta Vižintina deluje na področju inženiringa površin in trdih prevlek. V zadnjih 10 letih ima s področja inženiringa površin prek 60 objav v mednarodnih revijah in zbornikih mednarodnih konferenc.

Maja 2006 je dr. Podgornik na mednarodni konferenci "International Conference on Metallurgical Coatings and Thin films - ICMCTF" v San Diegu, ZDA prejel Bunshahovo nagrado, ki jo Ameriško združenje za vakuumsko tehniko, oddelek za napredni inženiring površin (Advanced Surface Engineering Divission) podeljuje za najboljši prispevek, predstavljen na konferenci.

Ameriško zdrženje za vakuumsko tehniko vsako leto v San Diegu, ZDA organizira Mednarodno konferenco o trdih prevlekah in tankih površinskih plasteh - ICMCTF, ki je najpomembnejši svetovni dogodek na področju trdih prevlek in površinskih plasti (www2.avs.org/conferences/icmctf). Konference, na kateri je predstavljenih prek 700 prispevkov s področja trdih prevlek in površinskih plasti, v obliki predavanj ali posterjev, se redno udeležuje prek 800 raziskovalcev in proizvajalcev opreme z vsega sveta. V spomin na prof. R.F. Bunshaha, njegovo dediščino in inovativne dosežke na področju nanosa trdih prevlek se od leta 1983 dalje na konferenci podeljuje istoimenska nagrada - Bunshah Award. Prof. R.F. Bunshah je bil eden od pionirjev na področju fizikalnega nanosa trdih prevlek iz parne faze (PVD), ki ni bil le izjemen znanstvenik in raziskovalec, temveč tudi človek predan prenašanju svojih spoznanj čim širši javnosti. Ta predanost je v letu 1974 privedla do organizacije prve konference s področja tankih plasti in trdih prevlek, ki je nato prerasla v vsakoletni mednarodni dogodek. V njegov spomin se na konferenci podeljuje Bunshahova nagrada, katere namen je izpostaviti najboljše objavljene prispevke. Izbira temelji na izvirnosti prispevka, znanstvenega vpliva na področje trdih prevlek in kakovosti objavljenega prispevka.

Dr. Podgornik je nagrado prejel za prispevek, predstavljen na konferenci ICMCTF v letu 2005, z naslovom "Tribological Reactions Between Oil Additives and DLC Coatings for Automotive Applications". V prispevku so bili predstavljeni rezultati večletnega raziskovalnega dela na področju prevlek na temelju trdega ogljika (Diamon Like Carbon - DLC), predvsem njihove združljivosti z znanimi mazivi in možnosti uporabe v avtomobilski industriji. Tribološke raziskave, ki so potekale v sodelovanju z enim od vodilnih podjetij na področju trdih prevlek (Balzers, Liechtenstein), so bile opravljene v Centru za tribologijo in tehnično diagnostiko na Fakulteti za strojništvo v Ljubljani, katerega vodja je prof. dr. Jože Vižintin.

Doktorat, magisterija in diplome - Doctor's, Master's and Diploma Degrees

DOKTORAT

Na Fakulteti za strojništvo Univerze v Mariboru je z uspehom zagovarjal svojo doktorsko disertacijo:

dne *14. julija 2006*: **Petja Pižmoht**, z naslovom: "Vrednotenje opcij razvojno-tehnoloških projektov".

S tem je navedeni kandidat dosegel akademsko stopnjo doktorja znanosti.

MAGISTERIJA

Na Fakulteti za strojništvo Univerze v Ljubljani je z uspehom zagovarjal svoje magistrsko delo:

dne *26. septembra 2006*: **Miha Janežič**, z naslovom: "Vpliv eksperimentalne podpore na uspešnost napovedovanja združljivosti izdelka".

Na Fakulteti za strojništvo Univerze v Mariboru je z uspehom zagovarjal svoje magistrsko delo:

dne *7. julija 2006*: **Emil Škrinjar**, z naslovom: "Gospodarjenje z vodo in čiščenje odpadnih vod v industriji sladkorja".

S tem sta navedena kandidata dosegla akademsko stopnjo magistra znanosti.

DIPLOMIRALI SO

Na Fakulteti za strojništvo Univerze v Ljubljani so pridobili naziv univerzitetni diplomirani inženir strojništva:

dne *5. septembra 2006*: Jan ČERNETIČ, Zmago JEREB, Miha KOŠENINA, Klemen OBLAK, Aljaž OSTERMAN, Jure POČKAR, Goran VIŠNJIĆ, Rok ZUPANČIČ;

dne *6. septembra 2006*: Uroš LESKOVŠEK, Tadej MUHIČ, Matej PLETERSKI;

dne *28. septembra 2006*: Boštjan HUDOKLIN, Vili PEPEL, Domen ŠERUGA, Henrik ZALETELJ.

Na Fakulteti za strojništvo Univerze v Mariboru so pridobili naziv univerzitetni diplomirani inženir strojništva:

dne *14. septembra 2006*: Niko ROZMAN, Daniel VIPAVC;

dne *28. septembra 2006*: Gregor CVETKO.

*

Na Fakulteti za strojništvo Univerze v Ljubljani so pridobili naziv diplomirani inženir strojništva:

dne *31. avgusta 2006*: Ana DOLENC, Barbara PIRC;

dne *13. septembra 2006*: Alan BESEDNJAK, Andrej HOSTNIK, Klemen KLANČAR, Niko KOKELJ, Anže LIKOZAR, Luka MAROVT, Roman PETEK;

dne *19. septembra 2006*: Dejan GRABNAR, Gorazd JELENC, Enej SAKSIDA.

Na Fakulteti za strojništvo Univerze v Mariboru so pridobili naziv diplomirani inženir strojništva:

dne *14. septembra 2006*: Janez KÜZMIČ, Franci PACEK;

dne *21. septembra 2006*: Dušan IKOVIC, Janez RAMŠAK, Mihael VODENIK;

dne *28. septembra 2006*: Mirko GORŠEK, Igor KABLAREVIĆ, Marko KASTELIC, Primož KITEK, Marko KRAJNC, Zdravko KUKOVEC, Franci REČNIK, Miha TOMAŠEK, Tomo VINCEKOVIĆ.

Navodila avtorjem - Instructions for Authors

Članki morajo vsebovati:
- naslov, povzetek, besedilo članka in podnaslove slik v slovenskem in angleškem jeziku,
- dvojezične preglednice in slike (diagrami, risbe ali fotografije),
- seznam literature in
- podatke o avtorjih.

Strojniški vestnik izhaja od leta 1992 v dveh jezikih, tj. v slovenščini in angleščini, zato je obvezen prevod v angleščino. Obe besedili morata biti strokovno in jezikovno med seboj usklajeni. Članki naj bodo kratki in naj obsegajo približno 8 strani. Izjemoma so strokovni članki, na željo avtorja, lahko tudi samo v slovenščini, vsebovati pa morajo angleški povzetek.

Za članke iz tujine (v primeru, da so vsi avtorji tujci) morajo prevod v slovenščino priskrbeti avtorji. Prevajanje lahko proti plačilu organizira uredništvo. Če je članek ocenjen kot znanstveni, je lahko objavljen tudi samo v angleščini s slovenskim povzetkom, ki ga pripravi uredništvo.

Papers submitted for publication should comprise:
- Title, Abstract, Main Body of Text and Figure Captions in Slovene and English,
- Bilingual Tables and Figures (graphs, drawings or photographs),
- List of references and
- Information about the authors.

Since 1992, the Journal of Mechanical Engineering has been published bilingually, in Slovenian and English. The two texts must be compatible both in terms of technical content and language. Papers should be as short as possible and should on average comprise 8 pages. In exceptional cases, at the request of the authors, speciality papers may be written only in Slovene, but must include an English abstract.

For papers from abroad (in case that none of authors is Slovene) authors should provide Slovenian translation. Translation could be organised by editorial, but the authors have to pay for it. If the paper is reviewed as scientific, it can be published only in English language with Slovenian abstract, that is prepared by the editorial board.

VSEBINA ČLANKA

Članek naj bo napisan v naslednji obliki:
- Naslov, ki primerno opisuje vsebino članka.
- Povzetek, ki naj bo skrajšana oblika članka in naj ne presega 250 besed. Povzetek mora vsebovati osnove, jedro in cilje raziskave, uporabljeno metodologijo dela,povzetek rezulatov in osnovne sklepe.
- Uvod, v katerem naj bo pregled novejšega stanja in zadostne informacije za razumevanje ter pregled rezultatov dela, predstavljenih v članku.
- Teorija.
- Eksperimentalni del, ki naj vsebuje podatke o postavitvi preskusa in metode, uporabljene pri pridobitvi rezultatov.
- Rezultati, ki naj bodo jasno prikazani, po potrebi v obliki slik in preglednic.
- Razprava, v kateri naj bodo prikazane povezave in posplošitve, uporabljene za pridobitev rezultatov. Prikazana naj bo tudi pomembnost rezultatov in primerjava s poprej objavljenimi deli. (Zaradi narave posameznih raziskav so lahko rezultati in razprava, za jasnost in preprostejše bralčevo razumevanje, združeni v eno poglavje.)
- Sklepi, v katerih naj bo prikazan en ali več sklepov, ki izhajajo iz rezultatov in razprave.
- Literatura, ki mora biti v besedilu oštevilčena zaporedno in označena z oglatimi oklepaji [1] ter na koncu članka zbrana v seznamu literature. Vse opombe naj bodo označene z uporabo dvignjene številke[1].

THE FORMAT OF THE PAPER

The paper should be written in the following format:
- A Title, which adequately describes the content of the paper.
- An Abstract, which should be viewed as a mini version of the paper and should not exceed 250 words. The Abstract should state the principal objectives and the scope of the investigation, the methodology employed, summarize the results and state the principal conclusions.
- An Introduction, which should provide a review of recent literature and sufficient background information to allow the results of the paper to be understood and evaluated.
- A Theory
- An Experimental section, which should provide details of the experimental set-up and the methods used for obtaining the results.
- A Results section, which should clearly and concisely present the data using figures and tables where appropriate.
- A Discussion section, which should describe the relationships and generalisations shown by the results and discuss the significance of the results making comparisons with previously published work. (Because of the nature of some studies it may be appropriate to combine the Results and Discussion sections into a single section to improve the clarity and make it easier for the reader.)
- Conclusions, which should present one or more conclusions that have been drawn from the results and subsequent discussion.
- References, which must be numbered consecutively in the text using square brackets [1] and collected together in a reference list at the end of the paper. Any footnotes should be indicated by the use of a superscript[1].

OBLIKA ČLANKA

Besedilo članka naj bo pripravljeno v urejevalnilku Microsoft Word. Članek nam dostavite v elektronski obliki.

Ne uporabljajte urejevalnika LaTeX, saj program, s katerim pripravljamo Strojniški vestnik, ne uporablja njegovega formata.

Enačbe naj bodo v besedilu postavljene v ločene vrstice in na desnem robu označene s tekočo številko v okroglih oklepajih

THE LAYOUT OF THE TEXT

Texts should be written in Microsoft Word format. Paper must be submitted in electronic version.

Do not use a LaTeX text editor, since this is not compatible with the publishing procedure of the Journal of Mechanical Engineering.

Equations should be on a separate line in the main body of the text and marked on the right-hand side of the page with numbers in round brackets.

Enote in okrajšave

V besedilu, preglednicah in slikah uporabljajte le standardne označbe in okrajšave SI. Simbole fizikalnih veličin v besedilu pišite poševno (kurzivno), (npr. *v, T, n* itn.). Simbole enot, ki sestojijo iz črk, pa pokončno (npr. ms^{-1}, K, min, mm itn.).

Vse okrajšave naj bodo, ko se prvič pojavijo, napisane v celoti v **slovenskem jeziku**, npr. časovno spremenljiva geometrija (ČSG).

Slike

Slike morajo biti zaporedno oštevilčene in označene, v besedilu in podnaslovu, kot sl. 1, sl. 2 itn. Posnete naj bodo v ločljivosti, primerni za tisk, v kateremkoli od razširjenih formatov, npr. BMP, JPG, GIF. Diagrami in risbe morajo biti pripravljeni v vektorskem formatu.

Pri označevanju osi v diagramih, kadar je le mogoče, uporabite označbe veličin (npr. *t, v, m* itn.), da ni potrebno dvojezično označevanje. V diagramih z več krivuljami, mora biti vsaka krivulja označena. Pomen oznake mora biti pojasnjen v podnapisu slike.

Vse označbe na slikah morajo biti dvojezične.

Preglednice

Preglednice morajo biti zaporedno oštevilčene in označene, v besedilu in podnaslovu, kot preglednica 1, preglednica 2 itn. V preglednicah ne uporabljajte izpisanih imen veličin, ampak samo ustrezne simbole, da se izognemo dvojezični podvojitvi imen. K fizikalnim veličinam, npr. *t* (pisano poševno), pripišite enote (pisano pokončno) v novo vrsto brez oklepajev.

Vsi podnaslovi preglednic morajo biti dvojezični.

Seznam literature

Vsa literatura mora biti navedena v seznamu na koncu članka v prikazani obliki po vrsti za revije, zbornike in knjige:

[1] A. Wagner, I. Bajsić, M. Fajdiga (2004) Measurement of the surface-temperature field in a fog lamp using resistance-based temperature detectors, *Stroj. vestn.* 2(2004), pp. 72-79.

[2] Vesenjak, M., Ren Z. (2003) Dinamična simulacija deformiranja cestne varnostne ograje pri naletu vozila. *Kuhljevi dnevi '03*, Zreče, 25.-26. september 2003.

[3] Muhs, D. et al. (2003) Roloff/Matek Maschinenelemente – Tabellen, 16. Auflage. *Vieweg Verlag*, Wiesbaden.

Podatki o avtorjih

Članku priložite tudi podatke o avtorjih: imena, nazive, popolne poštne naslove in naslove elektronske pošte.

SPREJEM ČLANKOV IN AVTORSKE PRAVICE

Uredništvo Strojniškega vestnika si pridržuje pravico do odločanja o sprejemu članka za objavo, strokovno oceno recenzentov in morebitnem predlogu za krajšanje ali izpopolnitev ter terminološke in jezikovne korekture.

Avtor mora predložiti pisno izjavo, da je besedilo njegovo izvirno delo in ni bilo v dani obliki še nikjer objavljeno. Z objavo preidejo avtorske pravice na Strojniški vestnik. Pri morebitnih kasnejših objavah mora biti SV naveden kot vir.

Units and abbreviations

Only standard SI symbols and abbreviations should be used in the text, tables and figures. Symbols for physical quantities in the text should be written in italics (e.g. *v, T, n,* etc.). Symbols for units that consist of letters should be in plain text (e.g. ms^{-1}, K, min, mm, etc.).

All abbreviations should be spelt out in full on first appearance, e.g., variable time geometry (VTG).

Figures

Figures must be cited in consecutive numerical order in the text and referred to in both the text and the caption as Fig. 1, Fig. 2, etc. Pictures may be saved in resolution good enough for printing in any common format, e.g. BMP, GIF, JPG. However, graphs and line drawings sholud be prepared as vector images.

When labelling axes, physical quantities, e.g. *t, v, m,* etc. should be used whenever possible to minimise the need to label the axes in two languages. Multi-curve graphs should have individual curves marked with a symbol, the meaning of the symbol should be explained in the figure caption.

All figure captions must be bilingual.

Tables

Tables must be cited in consecutive numerical order in the text and referred to in both the text and the caption as Table 1, Table 2, etc. The use of names for quantities in tables should be avoided if possible: corresponding symbols are preferred to minimise the need to use both Slovenian and English names. In addition to the physical quantity, e.g. *t* (in italics), units (normal text), should be added in new line without brackets.

All table captions must be bilingual.

The list of references

References should be collected at the end of the paper in the following styles for journals, proceedings and books, respectively:

[1] A. Wagner, I. Bajsić, M. Fajdiga (2004) Measurement of the surface-temperature field in a fog lamp using resistance-based temperature detectors, *Stroj. vestn.* 2(2004), pp. 72-79.

[2] Vesenjak, M., Ren Z. (2003) Dinamična simulacija deformiranja cestne varnostne ograje pri naletu vozila. *Kuhljevi dnevi '03*, Zreče, 25.-26. september 2003.

[3] Muhs, D. et al. (2003) Roloff/Matek Maschinenelemente – Tabellen, 16. Auflage. *Vieweg Verlag*, Wiesbaden.

Author information

The information about the authors should be enclosed with the paper: names, complete postal and e-mail addresses.

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