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Secția

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S U M A R

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Section

MACHINE CONSTRUCTION

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Volumul 62 (66), Numărul 3, 2016
Secţia
CONSTRUCŢII DE MAŞINI

DESIGN AND DEVELOPMENT OF A DEVICE USED IN BIAXIAL TESTING OF MATERIALS

BY

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Abstract. Biaxial testing of cruciform specimens is one of the most used experimental methods for the assessment of plane stress and strain state. The main advantage of this technique it is his versatility. The device conceived, designed and developed represents an alternative to complex testing machines. Cruciform specimen size and shape were established after conducting a series of finite element analysis in elasto-plastic domain. Experimental results obtained by biaxial tensile testing of cruciform specimens using devices attached to the universal testing machine are accurate.

Keywords: material characterization; biaxial tensile tests; plane stress states; cruciform specimens, FEA.

1. Introduction

Objectives of biaxial experiments are: to predict material failure, to evaluate materials mechanical behaviour, to calibrate constitutive models, to validate limit state theories, etc. Also this tests are used for fracture

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characterization of glass fiber composite laminate (Rashedi *et al.*, 2016), anisotropy characterization (Zhang *et al.*, 2015) etc. Loading an in-plane cruciform specimen in two orthogonal directions simultaneously, with attachable devices, is a big challenge.

Zouani *et al*. have realized a brief review of the existing multiaxial testing methods through which can be obtained a biaxial tensile stress state (Zouani *et al.*, 1999). Brieu *et al*. have proposed and validate a new biaxial tension mechanism for testing hyperelastic behavior of rubber-like materials (Brieu *et al.*, 2007).

Biaxial experiments are limited by a set of factors such as:

– cruciform specimen shape and size;

– material nature;

– cruciform specimen processing methods;

– testing system set-up;

– costs, etc.

Through biaxial tensile loadings can be tested a wide range of materials: warm rolled 316 L stainless steel (Petegem *et al.*, 2016), DC04 sheet steel (Schmaltz and Willner, 2014), dual phase steel - JSC590Y (Hanabusa *et al*., 2013), cell seeded collagen gels (Hu *et al*., 2014), rubber (Fujikawa *et al*., 2014), elastomers (Promma *et al.*, 2009) etc.

Attachable devices to universal testing machines used in biaxial tensile tests can be based on various mechanisms:

– mechanisms with sliding elements (Andrușcă *et al.*, 2014);

– mechanisms with articulated connecting rods (pantograph) (Andrușcă *et al.*, 2014);

– mechanisms with inclined planes (Andrușcă *et al.*, 2015) etc.

2. Design, Development and Testing of Attachable Device

After the analysis and design of different versions for attachable devices the alternative selected to be built was the one with sliding elements on inclined planes.

Drawing of custom apparatus is showed in Fig. 1. Main parts of the assembly are listed in Table 1.

The operating principle of the device is based on the sliding of the four main elements (driving and median). One of the two driving elements is fixed in universal testing machine. The other is driven by traction force and is controlled by the load cell of testing machine. The contact between the four elements is made over the entire length of sliding cylinders, but not on the entire circumference.

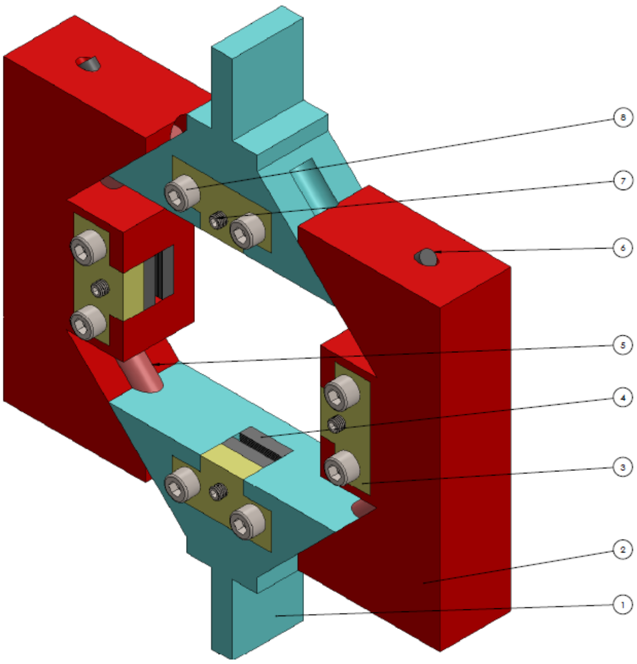

Fig. 1 – Component elements of the custom-built attachable device.

Table 1
Main Components of the Mechanism

Item No.	Description	Qty.
1	Driving element	2
2	Median element	2
3	Fixing element for grippers	4
4	Gripper	8
5	Sliding cylinder	4
6	Fixing element for sliding cylinder	4
7	Set screw	4
8	Hexagon socket head cap screw	8

For a median element (Fig. 2) can be written the condition that the resultant force projection in the horizontal direction to be null like in Eq. (1).

$$N - 2P\cos 45° + 2F\cos 45° = 0 \tag{1}$$

where: N, P, F represents forces acting on device elements.

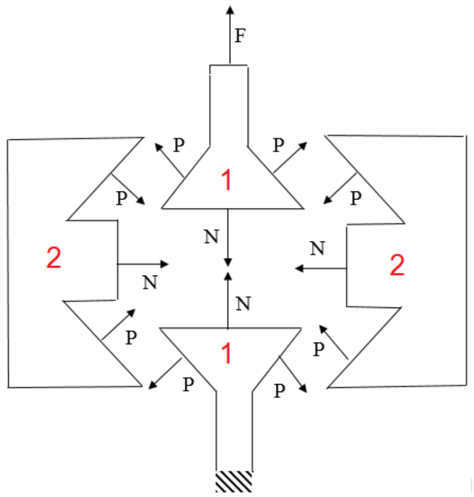

Fig. 2 – Forces acting on device elements.

Cruciform specimen model used in FEA analysis, with areas of interest, is presented in Fig. 3. Equivalent stress is calculated with von Mises limit state theory, frequently used for failure prediction of materials with predominantly ductile behavior.

$$\sigma_{VM} = \frac{1}{\sqrt{2}}\sqrt{\left(\sigma_x - \sigma_y\right)^2 + \left(\sigma_y - \sigma_z\right)^2 + \left(\sigma_z - \sigma_x\right)^2 + 6\left(\tau_{xy}^2 + \tau_{xz}^2 + \tau_{yz}^2\right)} \quad (2)$$

where: σ_i represents normal stresses and τ_{ij} represents shear stresses, [MPa].

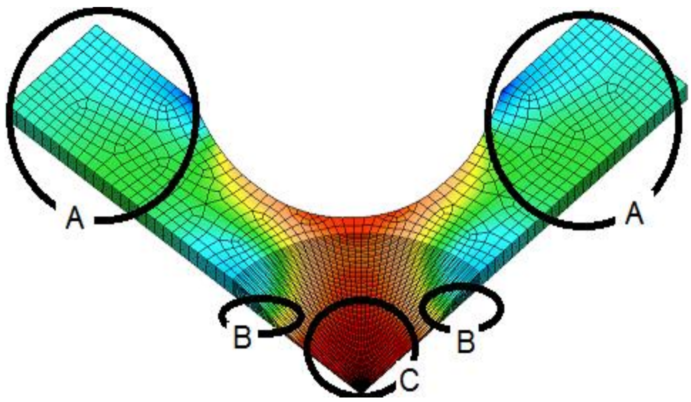

Fig. 3 – FEA analysis of 1/8 cruciform specimen (von Mises stresses)
A – arms region; B – Transition region; C – Gauge area.

After the device and cruciform specimen were designed and realized, they have been used in biaxial testing of metallic materials (Fig. 4). Cruciform specimens manufactured from Al 6082 T6 aluminum alloy were tested up to breaking.

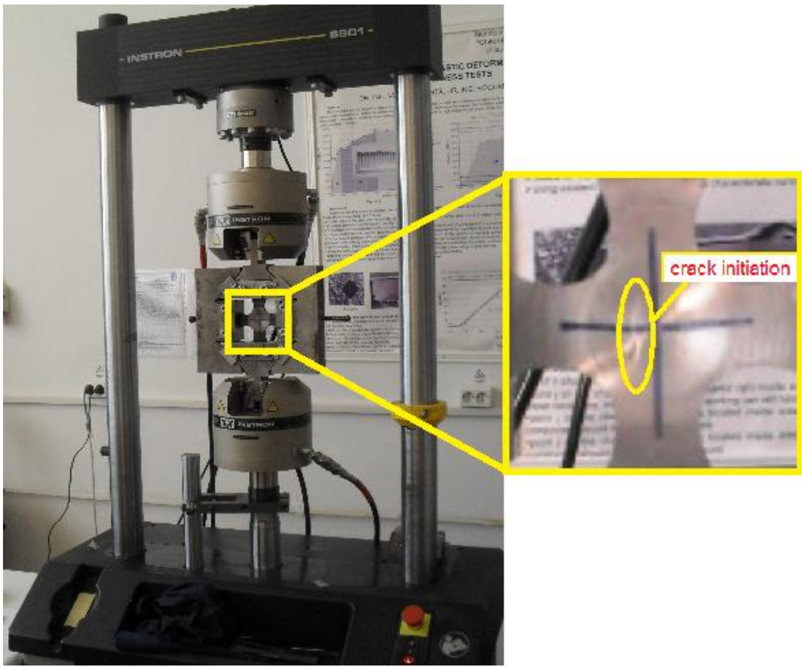

Fig. 4 – Attachable device used in biaxial tensile tests installed on Instron 8801 universal testing machine.

3. Conclusions

A new concept of an attachable device and cruciform specimen is used to asses biaxial stress state in metallic materials. Inclined planes device is the solution proposed to perform biaxial tensile tests using cruciform specimens. The experimental set-up was validated for metallic materials (S 235 JR, Al 6082 T6).

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PROIECTAREA ȘI DEZVOLTAREA UNUI DISPOZITIV FOLOSIT ÎN TESTAREA BIAXIALĂ A MATERIALELOR

(Rezumat)

Testarea la tracțiune biaxială a epruvetelor cruciforme reprezintă o tehnică experimentală foarte versatilă. Prin intermediul ei se poate obține o stare plană de tensiuni pentru o gamă variată de materiale. În contextul raționalizării și minimizării configurației experimentale este necesară găsirea unor noi soluții constructive privind mașinile, dispozitivele și epruvetele utilizate în testarea multiaxială a materialelor. Dispozitivele atașabile la mașinile de încercat universale reprezintă varianta economică la sistemele de testat complexe și scumpe. Rezultatele experimentale obținute în urma încercării la tracțiune biaxială a epruvetelor cruciforme folosind sistemele atașabile sunt precise. În această lucrare este prezentat un nou model de dispozitiv care a fost proiectat, realizat și validat pentru testarea biaxială a materialelor metalice.

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FEA STUDY ON THE INFLUENCE OF FILLET ON STRESS CONCENTRATION ON EDGES

BY

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Abstract. FILLET shape is frequently used in Computer Aided Design, CAD design of parts used in Mechanical engineering in order to reduce the stress concentration. The paper presents a Finite Element Analysis, FEA study on the influence of the FILLET applied on the edges of a through-the-thickness hole concentrator in a plate. FEA study is performed by use of open source packages: Salome-Meca and Code Aster. For the initial case, in the absence of FILLET, results are compared with the theoretical case: infinite plate subjected to biaxial stress with a through-the-thickness hole, a problem with analytical solution. FEA study determines the variation of the stress concentration coefficient versus the radius of the FILLET.

Keywords: CAD; FEA; Edges; FILLET; Stress concentration.

1. Introduction

The edges of the components used in Mechanical Engineering are sometimes stress concentration areas. CAD design frequently uses, when possible, geometrical solutions to optimise the shape in these areas in order to improve the stress distribution.

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FEA studies, (Aignătoaie, 2010; Aignătoaie, 2011; Aignătoaie, 2014), make possible to evaluate the influence of a specific shape on the stress concentration.

The theoretical problem of an infinite plate subjected to a biaxial stress field with a through-the-thickness hole concentrator, has an analytical solution, usually used in the study of the residual stresses (Bârsănescu *et al*., 2004).

The test case study is a plate with a through-the-thickness hole, Fig. 1.

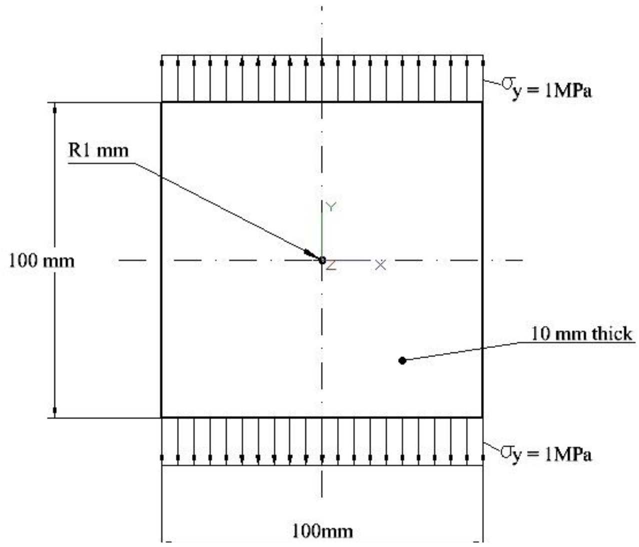

Fig. 1 − Test case study: finite plate with monoaxial load.

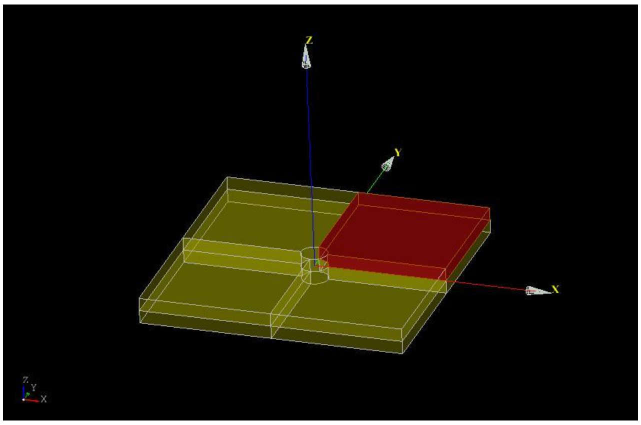

Fig. 2 − Simplified CAD model for FEA study.

2. The FEA Study

Due to symmetry properties, the model for FEA study could be reduced to 1/8 from initial shape, Fig. 2. The supports conditions implied roller conditions in any symmetry plane: the DOF (Degrees Of Freedom) are FREE in the plane and BLOCKED normal to the plane.

The plate is from steel with medium content of Carbon: E = 2.1 10^5 MPa and υ = 0.3.

The FEA study was performed by use of the Open Source packages Salome-Meca (pre/post-processor) and Code Aster (processor), as parts of CAELINUX 2013, (Caelinux).

The study has considered 4 test cases for which R, the radius of the FILLET, took the values 0 (without FILLET), 1, 2, 3 mm. Details of the meshes are included in Table 1 and Figs. 3, 6, 9, 12.

Details on the stress distribution in the vicinity of the concentrator are presented for σ_y, Figs. 4, 7, 10, 13, and $\sigma_{\text{von Mises}}$, Figs. 5, 8, 11, 14.

Table 1
Basic Parameters of the FEA Study

Study case	FILLET R, [mm]	Finite Elements [Quadratic tetrahedrons]	Nodes	DOFs
1	0	109281	173525	520575
2	1	115201	182723	548169
3	2	115627	183509	550527
4	3	117758	186506	559518

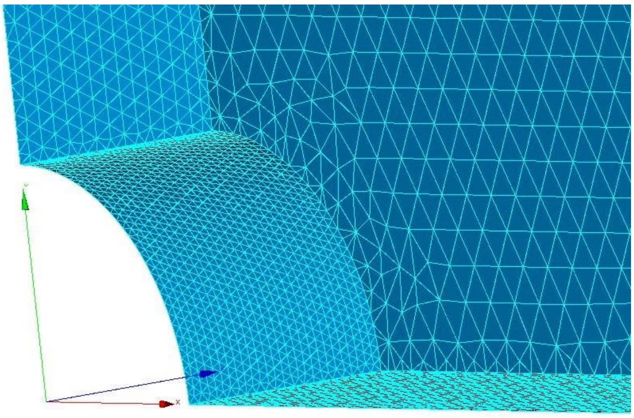

Fig. 3 − Case 1 (R = 0): Mesh details.

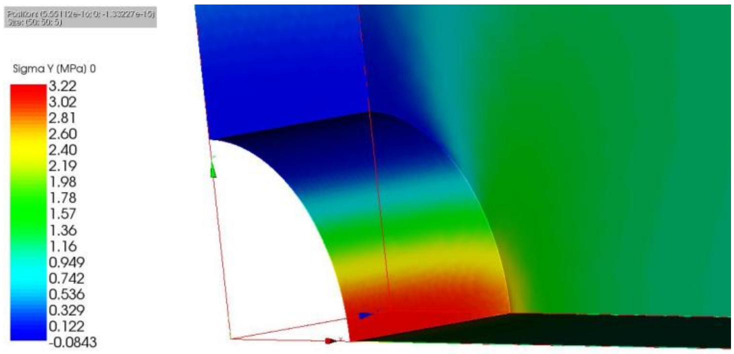

Fig. 4 − Case 1 (R = 0): σ_y, [MPa].

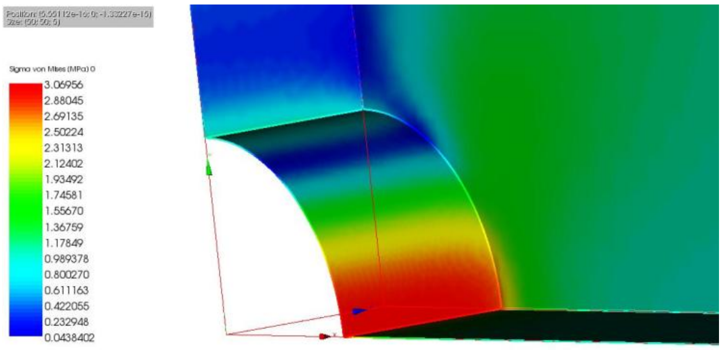

Fig. 5 − Case 1 (R = 0): $\sigma_{\text{von MISES}}$, [MPa].

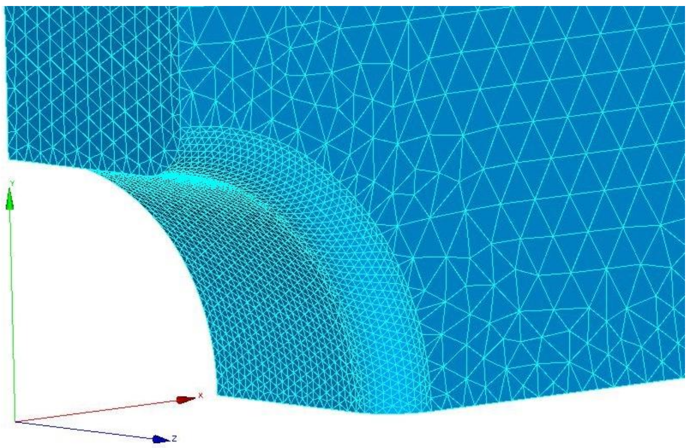

Fig. 6 − Case 2 (R = 1): Mesh details.

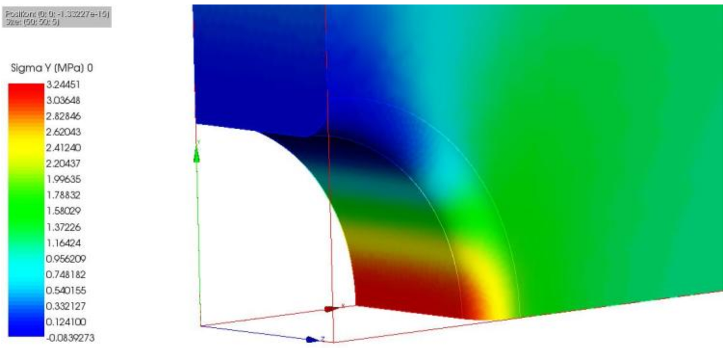

Fig. 7 − Case 2 (R = 1): σ_y, [MPa].

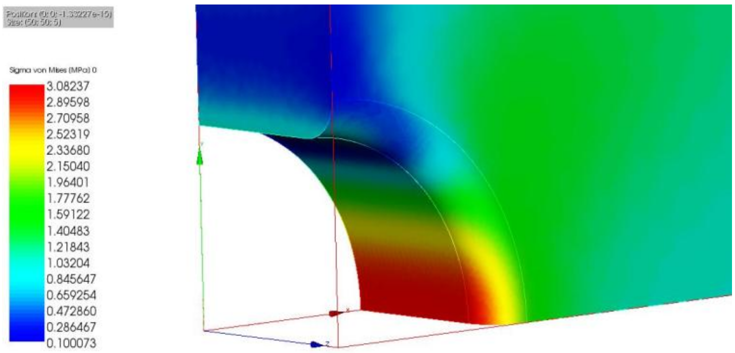

Fig. 8 − Case 2 (R = 1): $\sigma_{\text{von MISES}}$, [MPa].

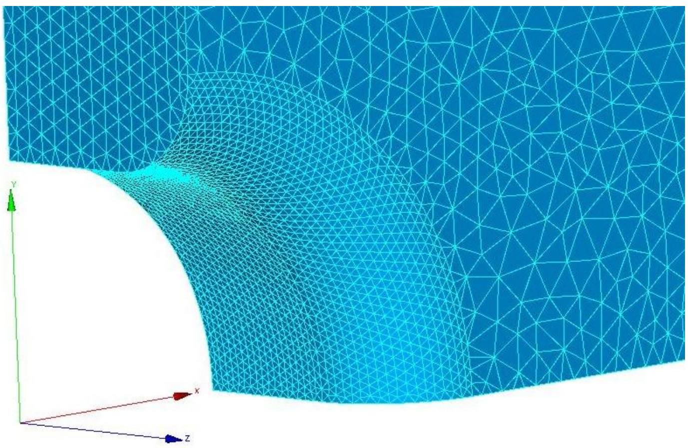

Fig. 9 − Case 3 (R = 2): Mesh details.

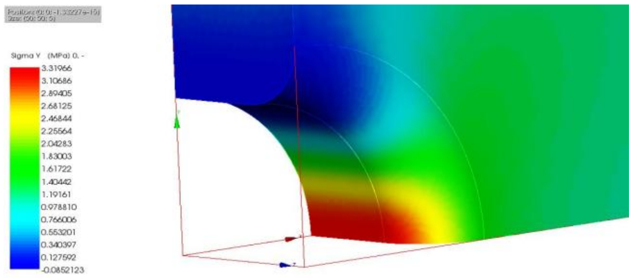

Fig. 10 − Case 3 (R = 2): σ_y, [MPa].

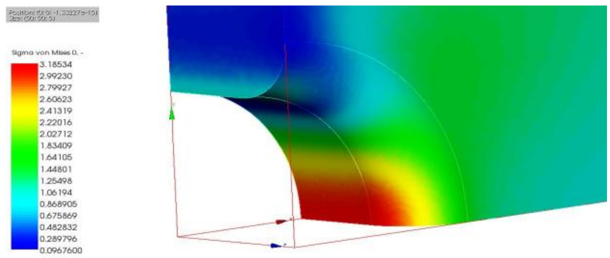

Fig. 11 − Case 3 (R = 2): $\sigma_{\text{von MISES}}$, [MPa].

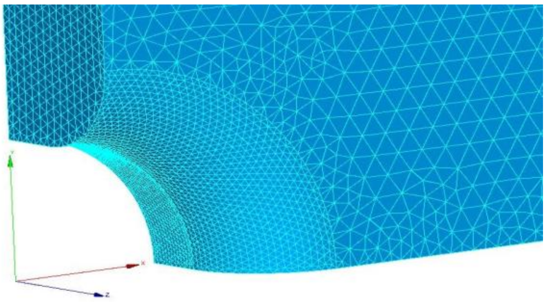

Fig. 12 − Case 4 (R = 3): Mesh details.

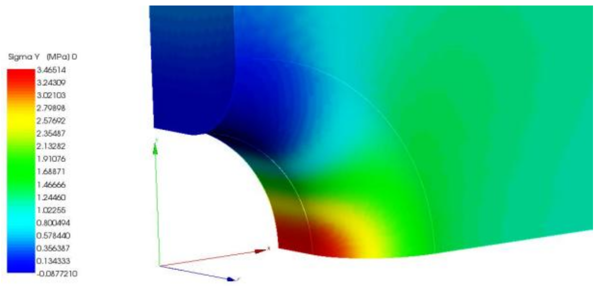

Fig. 13 − Case 4 (R = 3): σ_y, [MPa].

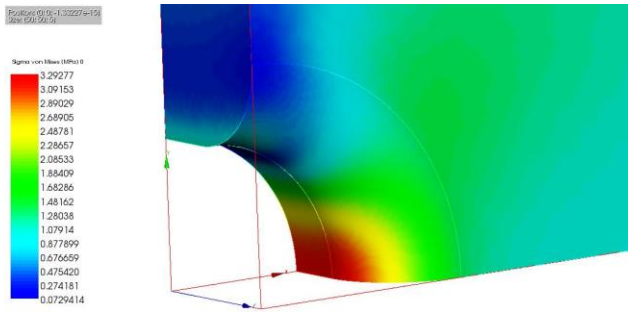

Fig. 14 − Case 4 (R = 3): $\sigma_{\text{von MISES}}$, [MPa].

The stress concentration factor, K, was calculated by use of Eq. (1):

$$K = \frac{\sigma_{\max}}{\sigma_{nom}} \tag{1}$$

where: σ_{nom} is the uniformly distributed stress in the plate: $\sigma_y = 1$ MPa; $\sigma_{\max}$ is the maximum value of the stress produced by the stress concentrator: $(\sigma_y)_{\max}$, determined analytically or by FEA.

Table 2
FEA Results

Study case	FILLET R, [mm]	σ_y [MPa]	$\sigma_{\text{von MISES}}$ [MPa]	K [Eq. (1)]
1	0	3.22	3.06	3.22
2	1	3.24	3.08	3.24
3	2	3.31	3.18	3.31
4	3	3.46	3.29	3.46

The analytical solution, (Bârsănescu *et al*., 2004), gives a theoretical concentration factor K = 3. The difference between theoretical value (K = 3) and the FEA study (K = 3.22) could be related with the two formulations: (infinite plate in 2-D space with analytical calculus) versus (finite plate in 3-D space with numerical calculus based on FEA).

3. Conclusions

– The increase of R reduces the area of the stress concentration zone, which moves towards the median plane of the plate, while stress gradient increases in value.

– The use of FILLET does not reduce the stress concentration in the stress concentrator. The variation of K versus R is not linear.

– Future researches for reducing the stress concentration could also try other geometry profiles on the edge or modifying the area in the vicinity of the hole.

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STUDIU FEA ASUPRA INFLUENȚEI UTILIZĂRII
FORMEI FILLET APLICATĂ MUCHIILOR ASUPRA FENOMENULUI
DE CONCENTRARE A TENSIUNILOR

(Rezumat)

Forma FILLET este utilizată frecvent în proiectarea CAD a unor elemente constructive utilizate în ingineria mecanică, cu scopul de a reduce fenomenul de concentrare a tensiunilor. Lucrarea prezintă un studiu AEF (Analiză cu Elemente Finite) privind utilizarea formei FILLET aplicată muchiilor unui concentrator de tensiune tip gaură străpunsă situat într-o placă. Studiul AEF este realizat cu ajutorul unor pachete Open-Source: Salome-Meca şi Code-Aster. Pentru cazul iniţial, fără FILLET, rezultatele sunt comparate cu problema teoretică cu soluţie analitică: placă infinită, cu o gaură străpunsă, solicitată biaxial. Studiul AEF determină variaţia coeficientului de concentrare a tensiunilor în funcţie de valoarea FILLET.

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EXPERIMENTAL INVESTIGATION OF FREE RESPONSE ON A CANTILEVER BEAM (FIRST FLEXURAL VIBRATION MODE)

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Abstract. This paper investigates the free response of first flexural vibration mode on a cantilever beam, mirrored in the signal delivered by a PZT piezoelectric plate transducer bonded near the fixed end, used as sensor. Despite the generally accepted theoretical model (as viscous damped free response with constant damping ratio and undamped angular frequency) it is experimentally proved that the real response is characterized by a relatively important variation of damping ratio and undamped angular frequency values, both depending by the amplitude of vibration. The investigation was done on a simple setup, by computer aided processing of the signal delivered by sensor.

Keywords: vibration; free response; cantilever beam; signal processing.

1. Introduction

A cantilever beam (as a simple example of vibratory mechanical system) is often used to introduce and illustrate different concepts in passive and active dynamics (Jassim *et al.*, 2013; Guan *et al.*, 2016). Frequently this free response on first flexural mode reveals the properties of the environment (Kramer *et al*., 2013) or materials (Paimushin *et al*., 2015).

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If the cantilever beam is placed in horizontal position, the free vibration on this mode is generally described as the movement $x(t)$ of the free end in vertical direction. In order to describe this movement x, [mm] related to time t, [s], the theoretical model of free viscous damped response of a single degree of freedom (spring-mass-damper) system (with constant values for damping ratio ξ, [] and undamped angular frequency of vibration ω_0, [rad/s]) is generally used (Kelly, 2000), according to Eq. (1).

$$x(t) = a \cdot e^{-\xi \cdot \omega_0 \cdot t} \cdot \sin(\sqrt{1-\xi^2} \cdot \omega_0 \cdot t + \varphi) \tag{1}$$

where: a, [mm] is the maximum amplitude and φ, [rad] is the phase at t = 0.

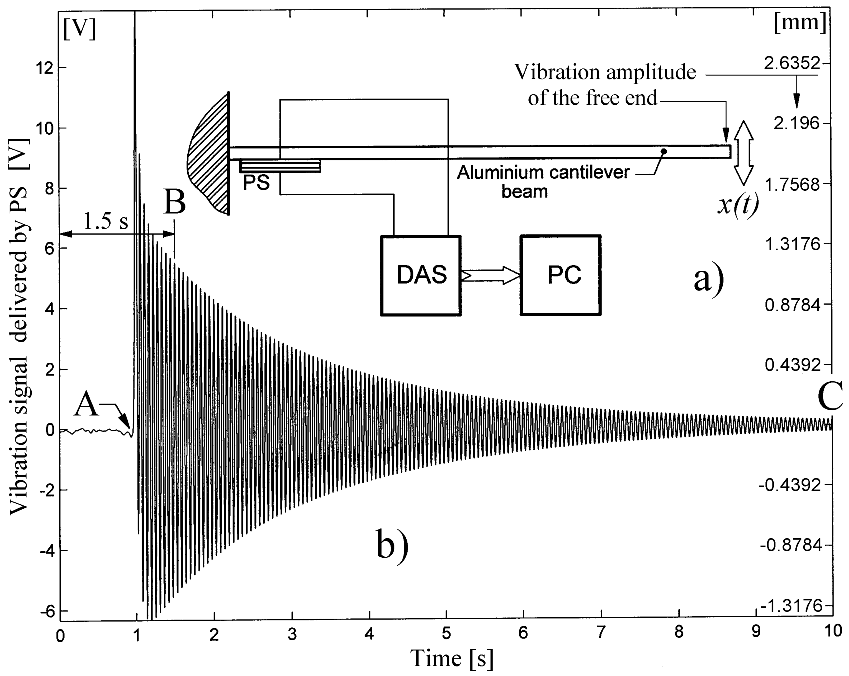

Fig. 1 – *a*) Experimental setup; *b*) Free response mirrored in the signal delivered by a PZT piezoelectric sensor (PS) with laminar rectangle design.

The research results disclosed herein proves by computer aided signal processing that -for relatively high values of vibration amplitude- this model is not accurate, the vibration parameters invoked above (ξ and ω_0) are not constant and they heavily depend on vibration amplitude value (especially the damping ratio).

These results might be important for certain applications and it is expected to be useful for future approaches in dynamics.

2. Experimental Setup

The experimental setup is described in Fig. 1*a*. It consists of an aluminium cantilever beam (300x25x2 mm^3) with a PZT piezoelectric transducer PS (used as sensor, with laminar rectangular design, 40x25x0.5 mm^3, Sensor Tech BM 500 type, d_{31} polarization, with $d_{31} = -175\cdot10^{-12}$ m/V) bonded on the close proximity of the rigidly fixed end of the cantilever beam.

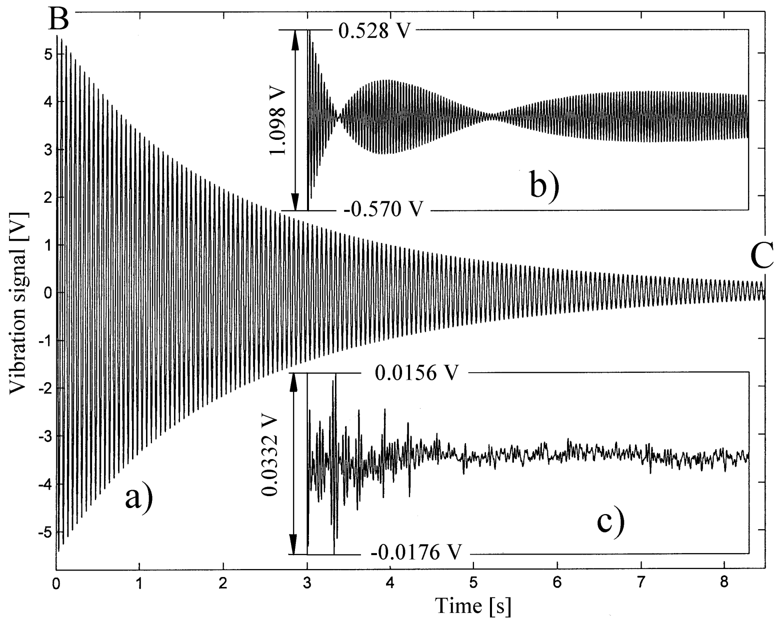

Fig. 2 – *a*) A part of signal from Fig. 1*b* used for fitting; *b*) The residual of a single fitting ($t_B < t < t_C$); *c*) The residual of fitting on intervals $(k\text{-}1)\cdot\Delta t < t_i <= k\cdot\Delta t$.

Due to th**e** flexural vibration $x(t)$ related with the free response (underdamped vibrations) on the first flexural mode of the cantilever beam, (which produces mechanical strain in the area where the sensor is placed (Horodincă, 2013) and the direct piezoelectric effect, the sensor generates a voltage $u(t) = C\cdot x(t)$, with C = 4.553 V/mm (a value supposed to be constant).

3. Signal Processing Technique (I)

This voltage is acquired in numerical format using a data acquisition system (DAS on Fig. 1*a*, with a 4424 PicoScope oscilloscope) and analyzed with a personal computer PC. Fig. 1*b* present the evolution of this voltage before and after the free response occurs (in A, manually excited). As Fig. 1*b* clearly indicates, the free response is totally mirrored in the evolution of this voltage. Assuming that the Eq. (1) describes $x(t)$ evolution between the moments depicted with B (t_B) and C (t_C) then:

$$u(t) = C \cdot x(t) = A \cdot e^{-\xi \cdot \omega_0 \cdot t} \cdot \sin(\sqrt{1-\xi^2} \cdot \omega_0 \cdot t + \varphi) \tag{2}$$

where: $t_B < t < t_C$ and $A = C{\cdot}a$, [V] is the maximum amplitude of the voltage.

The voltage between B and C -from Fig. 1*b*- is also drawn in Fig. 2*a*, considering $t_B = 0$ and $t_C = 8.5$ s.

The best way to check if the theoretical model from Eq. (2) correctly describes the experimental free response signal from Fig. 2*a* is the computer aided numerical fitting. The residual of curve fitting indicates the accuracy of the model. The best fitting means to find the appropriate values for the parameters (A, ξ, ω_0 and φ) involved in Eq. (2) in order to obtain a theoretical voltage evolution $u^t(t)$ which fits with maximum accuracy on the experimental voltage $u^e(t)$.

A computer aided fitting technique based on the minimal value of the cumulative error ε, [V] - depicted in Eq. (3) - was developed in order to find the best approximation for the parameters A, ξ, ω_0 and φ involved in $u^t(t)$ definition.

$$\varepsilon = \sum_{t_j = t_B}^{t_j = t_C} \left| u^e(t_j) - u^t(t_j) \right| = \min \tag{3}$$

With $\varepsilon = 0$ (a hypothetical situation) the theoretical and experimental voltages totally fit. An appropriate range and an increment of variation for each parameter were established. The set of four values of these parameters which produce a minimal cumulative error ε describes - according to Eq. (2) - the best fitting curve.

4. Experimental Results (I)

According to the procedure previously described, the experimental signal from Fig. 2*a* was numerically fitted. The values of parameters involved in Eq. (2) founds by fitting are written bellow in Table 1:

Table 1

Values of Fitting Parameters (a Single Fitting, for $t_B < t < t_C$)

A, [V]	ξ, []·100	ω_0, [rad/s]	φ, [rad]
5.2580	0.3792	112.9871	1.4196

Fig. 2*b* presents the evolution of the residual ($u^e(t_j)$- $u^t(t_j)$ with $t_B < t_j < t_C$). It is clear that the theoretical curve doesn't fit well the experimental evolution. Definitely this does not happen because of the fitting technique. The only available hypothesis is that the damping ratio and undamped angular frequency on experimental signal are not constant related to time. It is obvious that the beating phenomenon on Fig. 2*b* indicates certainly a variable undamped angular frequency.

With a good approximation it can be considered that the model depicted in Eq. (2) is available only for small intervals of time (*e.g.* Δt = 0.5 s). The signal fitting has been made once again on each interval (with $(k\text{-}1)\cdot\Delta t < t_i <= k\ \Delta t$), with the values found for fitting parameters written in Table 2. Before fitting, the evolution $u^e(t_j)$ on each interval was moved in origin on the abscissa.

Table 2

Values of Fitting Parameters (Fitting on Intervals $(k\text{-}1)\ \Delta t < t_i <= k\cdot\Delta t$*)*

k	A_k, [V]	ξ_k, []·100	ω_{0k}, [rad/s]	φ_k, [rad]
1	5.4757	0.4421	112.8134	1.5193
2	4.2637	0.4291	112.8692	1.4036
3	3.3522	0.3978	112.9239	1.3173
4	2.6813	0.3792	112.9726	1.2601
5	2.1656	0.3621	113.0080	1.2254
6	1.7644	0.3498	113.0403	1.2082
7	1.4467	0.3480	113.0649	1.2060
8	1.1865	0.3373	113.0864	1.2168
9	0.9814	0.3384	113.1050	1.2401
10	0.8121	0.3321	113.1154	1.2717
11	0.6720	0.3237	113.1295	1.3096
12	0.5592	0.3201	113.1396	1.3518
13	0.4655	0.3154	113.1499	1.4034
14	0.3877	0.3107	113.1583	1.4563
15	0.3237	0.3102	113.1663	1.5113
16	0.2696	0.3100	113.1703	1.5722
17	0.2253	0.3113	113.1660	1.6423

This technique of numerical fitting produces much better results as clearly is proved by the evolution of the residual given in Fig. 2*c*. Here the maximum peak to peak magnitude is 33 times less than in Fig. 2*b*. At the end of the evolution the residual is dominated by mechanical and electrical noise (vibrations in the environment and electromagnetic fields). At the beginning, the residual is possible dominated by some other excited vibration modes (poorly described by the sensor PS).

As expected, the damping ratio value is not constant, it has a relatively important variation: it decreases with the decreasing of vibration amplitude (as it is also graphically depicted in Fig. 3), while the undamped angular frequency increases slowly (as it is also graphically depicted in Fig. 4).

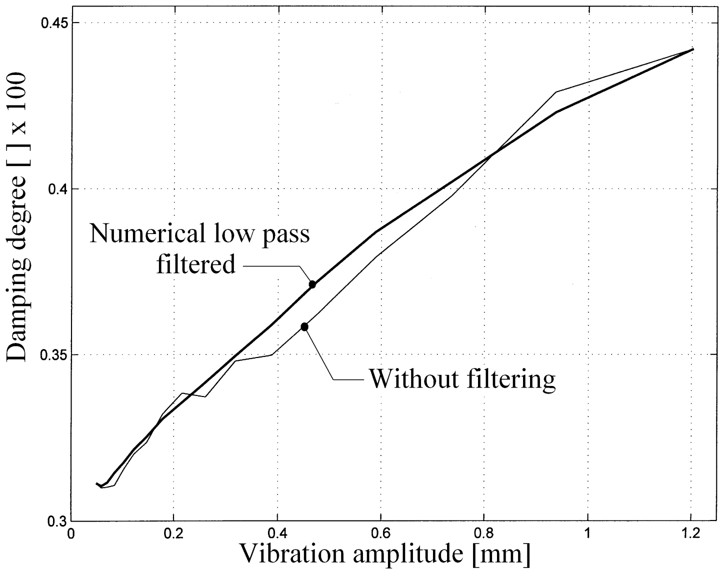

Fig. 3 – Evolution of damping ratio ξ_k (x100) related to vibration amplitude $a_k=A_k/C$.

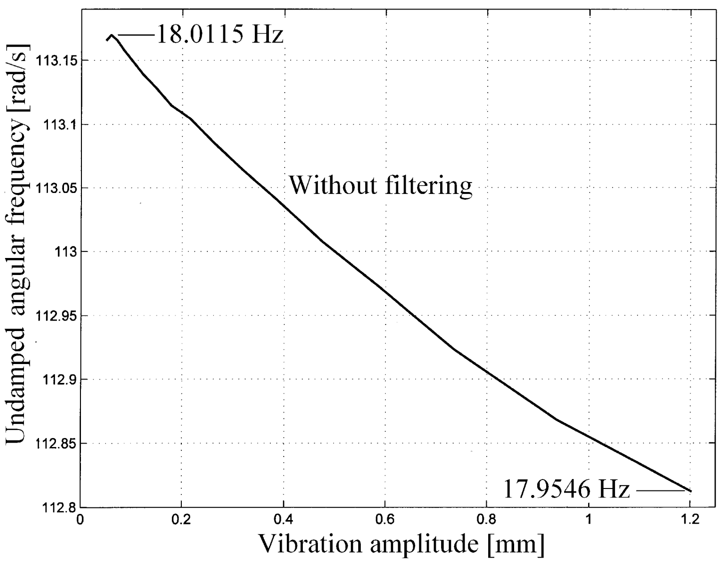

Fig. 4 – Evolution of undamped angular frequency ω_{0k} related to vibration amplitude $a_k=A_k/C$.

This means that the model proposed in Eq. (1) and Eq. (2) should be revised if it is used for large intervals of time. This dependence of damping ratio by vibration amplitude seems to be an important item useful in dynamics of cantilever beams.

For example, related with the researches exposed in (Horodincă, 2013), this result explains why when a vibratory mechanical system is actively supplied with positive mechanical modal power (which produces negative synthetic damping on first flexural mode of vibration) beyond of the stability limit, the amplitude of vibrations increases (and the absorbed modal power too) up to a certain limit depending by synthetic damping value. At this limit, the passive (positive!) damping in system (which increases with the increasing of amplitude) completely cancel the synthetic damping (negative, constant), the total damping becomes zero. The actuation input power flow becomes equal with the output (passive) power flow.

5. Signal Processing Technique (II)

Generally speaking the first signal processing technique exposed before (by curve fitting) has an important disadvantage: it takes time to apply. There is a simpler way to find out the evolution of ξ and ω_0 related by vibration amplitude.

On the evolution depicted in Fig. 2*a* let be a_i and a_{i+1} two consecutive current values of the signal amplitude (each one achieved at the instant times t_i and t_{i+1} respectively), according to Fig. 5*a*. These amplitudes are written with the considerations from Eq. (2) as follows:

$$a_i = \frac{A}{C} \cdot e^{-\xi_i \cdot \omega_{0i} \cdot t_i}, \qquad a_{i+1} = \frac{A}{C} \cdot e^{-\xi_{i+1} \cdot \omega_{0(i+1)} \cdot t_{i+1}} \tag{4}$$

where: t_{i+1}-t_i = T_j is the value of current period of the signal, and ω_{0i} is the current value of the undamped angular frequency, related by the current value of damping ration ξ_i and damped angular frequency $\omega = 2 \cdot \pi / T_j$ (Kelly, 2000) by equation:

$$\omega_{0i} = \frac{2 \cdot \pi}{T_j} \cdot \frac{1}{\sqrt{1 - \xi_i^2}} \tag{5}$$

In order to simplify this approach, let consider that $\xi_i = \xi_{i+1}$ and $\omega_{0i} = \omega_{0(i+1)}$. Let be δ_i the current value of the logarithmic decrement defined as:

$$\delta_i = \ln(\frac{a_i}{a_{i+1}}) = \ln(e^{\xi_i \cdot \omega_{0i} \cdot T_j}) = \xi_i \cdot \omega_{0i} \cdot T_j \tag{6}$$

These two last equations allow the calculus of current values for ω_{0i} and ξ_i, considering that a_i, a_{i+1} and T_j are previously determined by computer aided analysis of the signal from Fig. 2*a*, see the symbolic approach from Fig. 5*a* . If we assume that $n = 2\cdot\pi/T_j$, $m = \delta_i/T_j$ and $h = m/n$, then ξ_i is given by:

$$\xi_i = \frac{h}{\sqrt{1+h^2}} \tag{7}$$

With this result, the Eq. (5) allows the calculus for ω_{0i}.

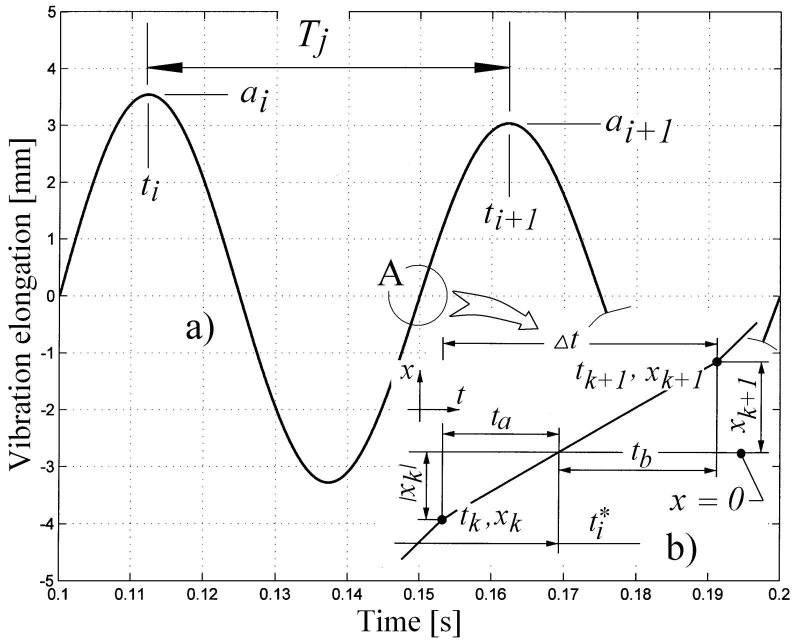

Fig. 5 – *a*) A symbolic approach for determination of amplitudes a_i, a_{i+1} and period T_j; *b*) Some considerations related to exact calculus of period T_j.

It is important to highlight that (t_i, a_i) and (t_{i+1}, a_{i+1}) are two signal samples. The values of amplitudes and instant times (and consequently of current period T_j of the signal) are more accurately determined if smaller sampling period (or a bigger sampling rate) is used. Here the sampling rate is 10,000 s^{-1} (or 278 samples per signal period, approximately). The accuracy of amplitudes is not a critical item, whereas are involved mainly in the definition of damping ratio (which has an important variation as was pointed in Fig. 3). But the accuracy of current period T_j - whereas is involved mainly in the definition of undamped angular frequency (with a very small variation as was already pointed in Fig. 4) - is very important.

For this reason, the highest accuracy possible technique for determining the value of T_j was developed according to Fig. 5*b*. The numerical evolution of the vibration elongation $x(t)$ is graphically depicted as a succession of points (samples) connected by line segments. Let be (t_k, x_k) and (t_{k+1}, x_{k+1}) the coordinates of two neighbouring points placed below and above the abscissa axis $t = 0$ (generally with $x_k < 0$ and $x_{k+1} > 0$). Here t_{k+1}-$t_k = \Delta t$ is the sampling period already introduced before. Let be $t_i^* = t_k + t_a$ the abscissa value for the point of intersection of segment line with abscissa axis. There are two similar triangles on Fig. 5*b*, so t_a and t_i^* can be described as:

$$t_a = \frac{|x_k|}{|x_k| + x_{k+1}} \cdot \Delta t \qquad t_i^* = t_k + \frac{|x_k|}{|x_k| + x_{k+1}} \cdot \Delta t \tag{8}$$

A similar description can be done for next intersection t_{i+1}^* of $x(t)$ evolution with abscissa axis, so the exact value of current period is given by:

$$T_j = t_{i+1}^* - t_i^* \tag{9}$$

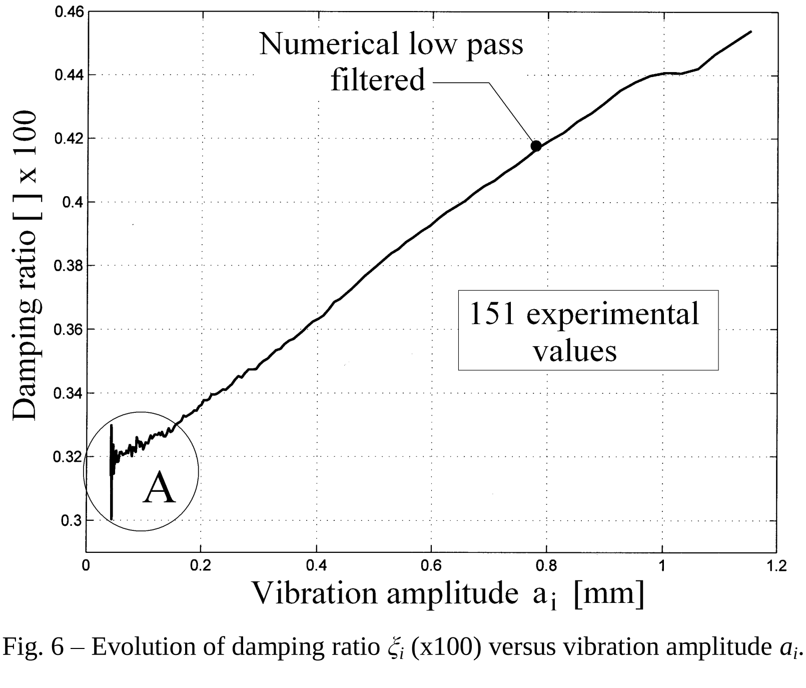

Fig. 6 – Evolution of damping ratio ξ_i (x100) versus vibration amplitude a_i.

In this approach, Eq. (8) is available also if $x_k = 0$ or $x_{k+1} = 0$. This calculation method is also available for an accurate calculus of period (and frequency as well) for all periodic experimental signals numerically described.

6. Experimental Results (II)

With these considerations, the evolution of damping ratio ξ_i versus amplitude a_i was experimentally determined - Eq. (7) - as it is graphically depicted in Fig. 6.

The evolution is quite similar with those already given in Fig. 3. But generally speaking it is more accurate and contains more experimental points (151 values). In the area marked with A (here and in Fig. 7 as well) the accuracy is bad because of the measurement noise (electrical and mechanical). This noise becomes important related with vibration amplitude value (which has a low level here).

Fig. 7 shows the evolution of undamped angular frequency ω_{0i} versus vibration amplitude a_i based on Eq. (5).

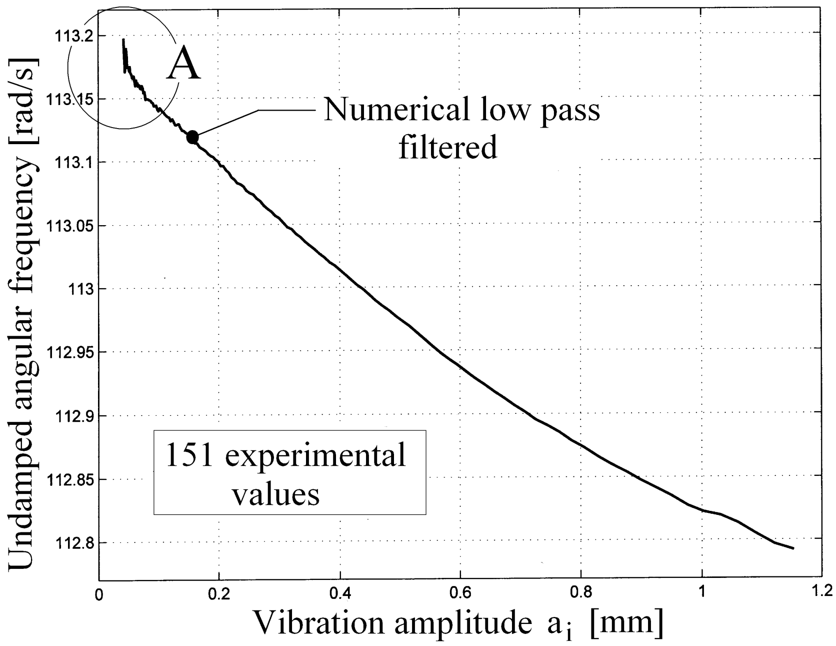

Fig. 7 – Evolution of undamped angular frequency ω_{0i} versus vibration amplitude a_i.

Here also a comparison with Fig. 4 proves that this new signal processing technique is correct.

Besides accuracy, this technique has also the advantage of a facile employment. The evolutions from Figs. 6 and 7 can be numerically fitted in order to find out the analytical form for $\xi_i = \xi_i(a_i)$ and $\omega_{0i} = \omega_{0i}(a_i)$ useful for different theoretical and experimental approaches in mechanical dynamics.

7. Conclusions

The paper proves in experimentally terms that the free response on first flexural vibration mode of a cantilever beam is characterized - despite the generally accepted theoretical model (the free response of an underdamped spring-mass-damper system) - by a relative significant variation of damping ratio ξ (according to Figs. 3 and 6) and a relatively slight variation of the undamped angular frequency ω_0 (according to Figs. 4 and 7), both related with vibration amplitude evolution.

A simple setup based on a cantilever beam with a PZT sensor, a data acquisition system (with numerical oscilloscope) and a personal computer was used, as is revealed in Fig. 1*a*). It is presumed that the signal delivered by sensor is proportional with the vibration elongation of the free end of the cantilever beam (on first vibration mode).

The evolutions of ξ and ω_0 were evaluated using two different computer processing techniques for the signal delivered by the PZT sensor placed in the proximity of the fixed end of the cantilever beam during the free response of the cantilever beam.

The first technique (relatively accurate) is based on numerically fitting of the experimental signal divided in sequences with short durations of time (0.5 s), while the second technique (more accurate and faster) is based on numerical processing of the evolution of amplitude and period of the signal. Both techniques were conceived and developed by author.

These results are interesting for scientific research in dynamics (https://www.mathworks.com/help/pde/). For example, an ambiguous item from a previous research (Horodincă, 2013) is clarified here: the increasing of passive damping ratio when the amplitude of vibration increases explain why if a negative synthetic modal damping is generated (by positive active modal mechanical power flow in a system with positive velocity-force feedback) the amplitude of vibration increases until the passive (positive) damping completely cancel the synthetic (negative) damping. If the total modal damping (synthetic and passive) is negative, the system becomes unstable, it starts vibrating.

In the near future the theoretical and experimental approach presented here (related by second processing technique of the signal) will be applied on the same setup, but using as strain sensor for flexural vibration a Wheatstone bridge with strain gauges placed also by bonding near the fixed end of the cantilever, collocated with the PZT sensor. A previous result of research (Horodincă, 2013) indicates that between the strain (as it is described by PZT sensor with an electrical voltage evolution) and vibration $x(t)$ at free end in cantilever beam (due to the flexural vibration on the first mode) exists a shift of phase (with a presumed average value of 14^{o}). A future research should establish a measurement and compensation technique in order to eliminate this phase shifting when the PZT transducers are used as sensor and actuator in a close loop feedback control system. Otherwise a feedback with a pure proportional control law acts as a proportional-derivative feedback (with a small derivative gain because this shift of phase).

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CERCETAREA EXPERIMENTALĂ A RĂSPUNSULUI LIBER PE PRIMUL MOD DE VIBRAŢIE (ÎNCOVOIERE) PENTRU O GRINDĂ ÎNCASTRATĂ LA UN CAPĂT

(Rezumat)

Lucrarea cercetează răspunsul liber pe primul mod de vibraţie flexională (încovoiere) pentru o grindă încastrată la un capăt, reflectat în semnalul furnizat de un sensor piezoelectric de tip plachetă, plasat prin lipire în proximitatea capătului încastrat al grinzii. Modelul teoretic general acceptat al acestui răspuns liber (amortizat vâscos, cu valori constante pentru gradul de amortizare şi pulsaţia proprie neamortizată) este contrazis, variaţii importante ale acestor parametri, ambii depinzând de amplitudinea vibraţiei. Cercetările au fost efectuate pe un stand experimental simplu (grindă încastrată la un capăt, senzor, osciloscop numeric şi calculator), cu aplicarea unor tehnici proprii de prelucrare asistată de calculator a semnalului furnizat de senzor.

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PROFILE EVOLUTION IN CYLINDRICAL ROLLER BEARINGS

II. RATING LIVES EVALUATION

BY

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Abstract. The roller profile appears to be the key element to attain a longer rating life for both cylindrical and tapered roller bearings. The class I discontinuities that exist along cylindrical-crowned roller profile generate high pressure peaks in pressure distributions that diminish considerably the modified rating life as is defined by ISO 16281:2008. After a certain number of cycles the material will shakedown elastically to a slightly modified roller profile and a stable state of compressive residual stresses. If were taken place, these changes have to be considered in the life evaluation.

In a previous paper, an analysis model has been developed to simulate the nonlinear strain rate dependent deformation of rolling bearing steel stressed in the elastic-plastic domain. An experimental validation of the developed elastic-plastic model is pointed out in the first part of the present paper.

In the second part of the paper, the basic reference rating lives have been evaluated using the methodology given in ISO 16281:2008. In this respect, pure elastic conditions as well as elastic-plastic material, able to undertake plastic modification of the roller profile, have been considered.

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The favourable effect of roller profile changes as result of the local plastic deformations has revealed.

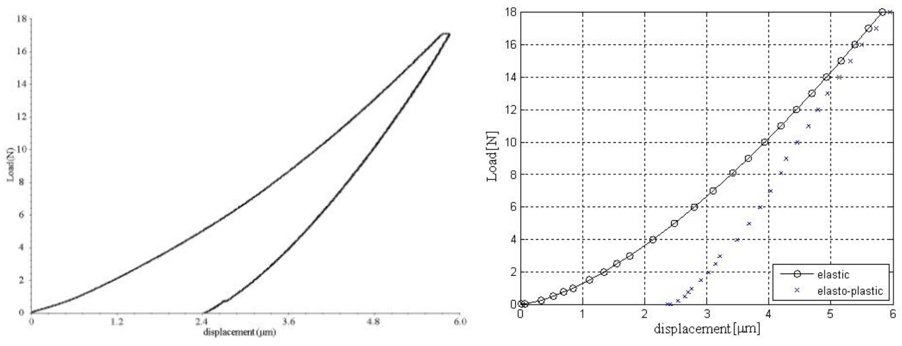
Keywords: elastic-plastic contact; residual stresses; cylindrical roller bearings; reference rating life.

1. Introduction

The cross-section roller's profile controls the pressure distribution in the contact area and radically affects the bearing dynamic load rating and life (Harris and Kotzalas, 2007; Ioannides *et al.*, 1999; ISO 16281:2008). Consequently the primary target has been to reduce the stresses in the roller-raceway contact by optimizing the roller profile. The diversity of crowning profiles includes: the single straight line with chamfer ends, single circular arc or a combination of multiple circular arcs, cylindrical-crowned known also as ZB profile. When Lundberg's logarithmic profile is used, (Lundberg, 1939), the distribution of contact pressure might results axially uniform. Still this theoretical profile has an infinite drop at the end of the effective contact length. In an elastic analysis, as is admitted in (ISO 16281:2008), these end increases in pressure distribution cause a severe diminishing of both the reference dynamic load rating and modified rating life.

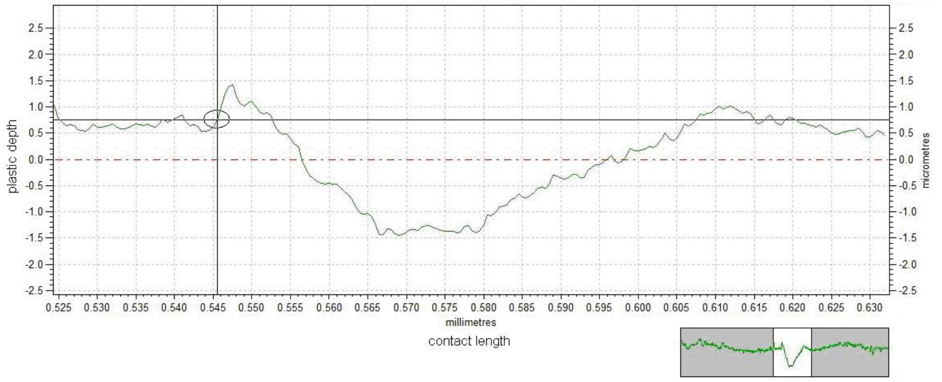
2. Experimental Validation of Improved Incremental Algorithm

The CETR UMT-2 tribometer and test specimen of 51206 axial bearing ring were used to perform the indentation tests and Taylor-Hobson Form Talysurf I50 profilometer was used to perform the micro-indentation measurements (Benchea and Crețu, 2010). For a normal load of 18 N the experimental and numerical values are comparatively depicted in Fig. 1.

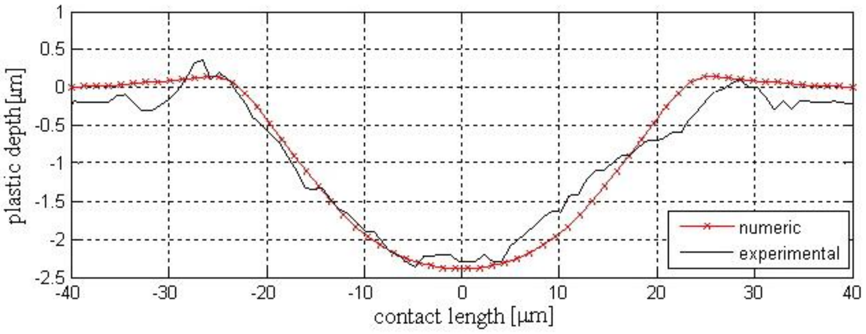


a *b*

Fig. 1 – Displacements vs. loading force:
a – experimental; *b* – numerical.



a



b

Fig. 2 – Indenter print depth:
a – experimental; *b* – experimental and numerical.

The Fig. 2 presents comparatively the measured profile of the indentation versus the profile obtained numerically with the elastic-plastic model.

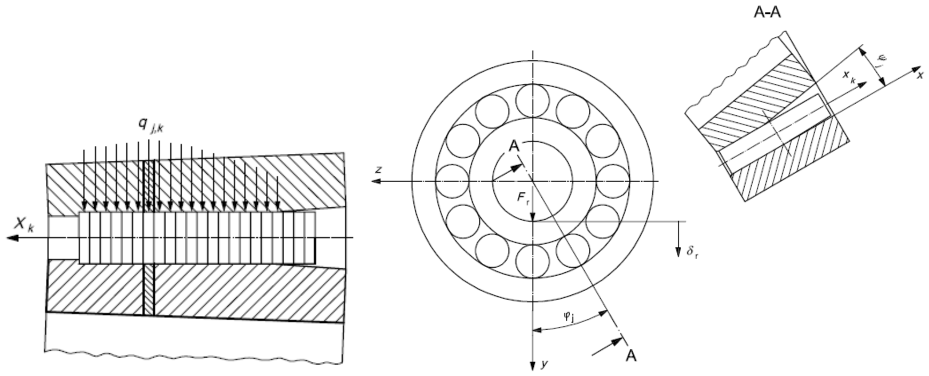
The experimental profile has diameter of 46.65 µm and a depth of 2.40 µm that are very close to those obtained numerically, 46.24 µm for the diameter and 2.38 µm for the depth.

3. Method for Calculating the Reference Rating Lives

3.1. Lamina Model

For the case where the raceways of rolling bearings are cylindrical, the elastic deflection of a misaligned rolling element can be described by a lamina model.

To calculate the elastic deflection, the roller is divided into n_s identical laminas, Fig. 3*a*. According to (ISO 16281:2008) the number of laminas shall be at least $n_s = 30$.



a *b*

Fig. 3 – Roller bearing internal load distribution (ISO 16281:2008):
a – lamina model; *b* – roller bearing misalignment.

The load $q_{j,k}$ on lamina *k* of the roller *j* is:

$$q_{j,k} = c_s \cdot \delta_{j,k}^{10/9} \tag{1}$$

where: $c_s = \dfrac{35948 \cdot L_{we}^{8/9}}{n_s}$, L_{we} – roller length.

For a radial load of the inner ring, the elastic deflection of the rolling element *j* is:

$$\delta_j = \delta_r \cdot \cos\varphi_j - \frac{s}{2} \tag{2}$$

where: *s* – radial clearance of bearing, φ_j – angular position of rolling element, δ_r – radial displacement given by load.

3.2. Rating Lives for Roller Bearings

For a normal load distribution, the difference between the dynamic equivalent rolling element loads for a rotating and a stationary inner ring is less than 2% (ISO 16281:2008). Generally, the inner ring is considered to be rotating and the outer ring to be stationary.

Basic reference rating life, L_{10r}, is given by:

$$L_{10r}=\left(\sum_{k=1}^{n_s}\left(\left(\frac{q_{kci}}{q_{kei}}\right)^{-4.5}+\left(\frac{q_{kce}}{q_{kee}}\right)^{-4.5}\right)\right)^{-\frac{8}{9}} \tag{3}$$

where: q_{kci} – basic dynamic load rating of a bearing lamina of the inner ring, q_{kce} – basic dynamic load rating of a bearing lamina of the outer ring, q_{kei} – dynamic equivalent load on a lamina k of a rotating inner ring, q_{kee} – dynamic equivalent load on a lamina k of a stationary outer ring.

According to (ISO 16281:2008), the dynamic equivalent loads on each lamina of the rotating inner ring and of each lamina of the stationary outer ring are computed as a function of the stress riser coefficients evaluated for each lamina of each roller. The stress risers are calculated considering the actual pressure distribution on the corresponding lamina versus and ideal hertzian pressure.

4. Basic Reference Rating Lives Evaluation

4.1. Effect of a Transient Overload on Pressure Distributions

Transitory overloads may induce permanent profile changes that modify the elastic pressure distributions for the subsequent normal running load. Fig. 4*a* exemplifies the elastic pressure distributions along profile of the most loaded rollers of a cylindrical roller bearing that supports 100 kN radial load.

If the cylindrical roller bearing was subjected to a transient overload of 450 kN the elastic shakedown took place and roller profile changes due to plastic deformations.

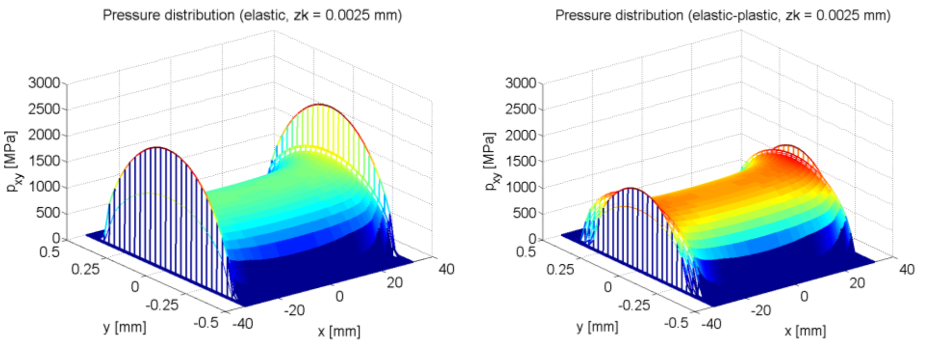


Fig. 4 – 3D Elastic pressure distributions,
(F_R = 100 kN, profile: $R1$ = 8100 mm, zk = 0.0025 mm):
a – before a transient overload of 450 kN; *b* – after a transient overload of 450 kN.

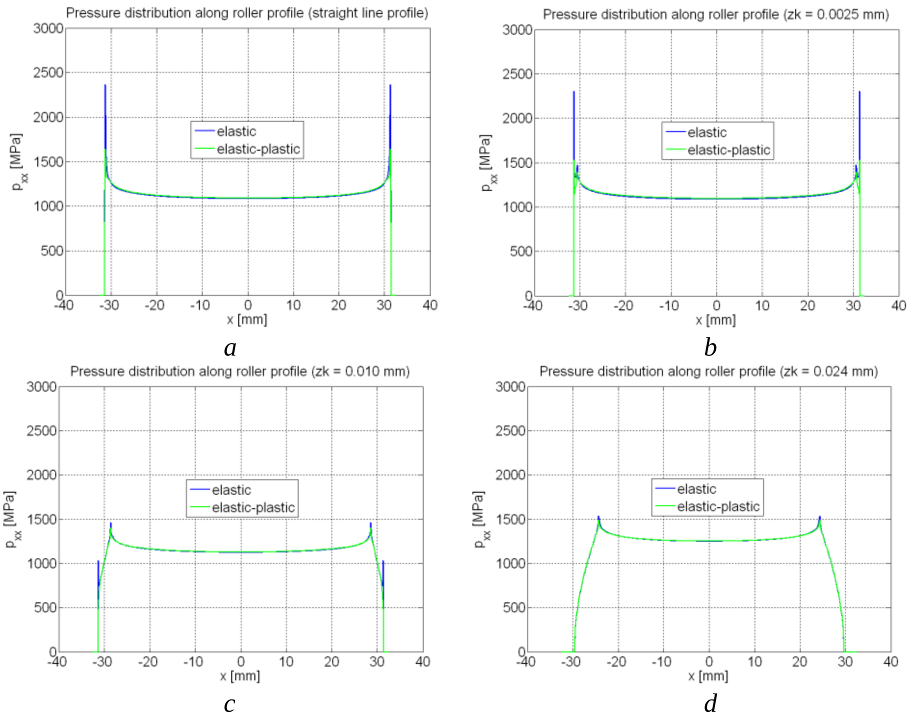


Fig. 5 – 2D Elastic pressure distributions for F_R = 100 kN,
before and after a transient overload of 450 kN:
a – straight line profile; *b* – crowned profile: *R*1 = 8100 mm, *zk* = 0.0025 mm;
c – crowned profile: *zk* = 0.010 mm, *d* – crowned profile: *zk* = 0.024 mm.

The induced plastic deformations alter the elastic pressure distributions, as in Fig. 4*b* and Fig. 5.

4.2. Normal Loading Conditions with Transient Overloading

Experimental investigations, (Crețu and Popinceanu, 1985; Muro *et al.*, 1973), as well as theoretical and computer simulations, (Chen *et al.*, 1988; Crețu *et al.*, 2007; Ko and Ioannides, 1988), have shown that the presence of compressive residual stresses can have a positive effect on fatigue lives of rolling bearings.

The slicing technique recommended in (ISO 16281:2008) make possible to consider the profile modifications in the evaluation of the basic reference rating life of a roller bearing. It does not include the role played by residual stresses. The basic reference rating lives, evaluated for a radial load of 100 kN and basic design roller profile are presented in Table 1. The value of basic rating life provided by standard (ISO 281:2007) is added.

Table 1

Basic Reference Rating Lives for Normal Loading Conditions (F_R = 100 kN)

Roller profile	Profile drop *zk*, μm	Basic rating lives, L_{10} hours, (ISO 281)	Basic reference rating lives, L_{10r}, hours, (ISO 16281)	
			Design profile	Elastic shake-down profile
Straight	–	$4.43 \cdot 10^4$	$3.80 \cdot 10^4$	$1.20 \cdot 10^5$
ZB	2.5	$4.43 \cdot 10^4$	$5.28 \cdot 10^4$	$1.26 \cdot 10^5$
ZB	10	$4.43 \cdot 10^4$	$1.09 \cdot 10^5$	$1.10 \cdot 10^5$
ZB	24	$4.43 \cdot 10^4$	$5.19 \cdot 10^4$	$5.28 \cdot 10^4$

The results from Fig. 4*b*, Fig. 5 and Table 1 reveal that a very short overload, able to induce small alterations of roller's profile, attenuates the pressure peaks with a beneficial effect on fatigue life.

4.1. Very Heavy Loading Conditions

Bearings incorporating rollers with unmodified profiles as well as bearings with modified roller profiles by elastic-shakedown process have been considered. The basic reference rating lives, L_{10r}, (ISO 16281:2008), evaluated for a running load of 450 kN are exemplified in Table 2.

Table 2

Basic Reference Rating Lives for Very Heavy Loading Conditions (F_R = 450 kN)

Roller profile	Profile drop *zk*, μm	Basic rating lives, L_{10} hours, (ISO 281)	Basic reference rating lives, L_{10r}, hours, (ISO 16281)	
			Design profile	Elastic shake-down profile
Straight	–	$2.94 \cdot 10^2$	$1.15 \cdot 10^1$	$3.37 \cdot 10^1$
ZB	2.5	$2.94 \cdot 10^2$	$1.29 \cdot 10^1$	$5.15 \cdot 10^1$
ZB	10	$2.94 \cdot 10^2$	$3.17 \cdot 10^2$	$3.98 \cdot 10^2$
ZB	24	$2.94 \cdot 10^2$	$2.49 \cdot 10^2$	$3.36 \cdot 10^2$

The profile modifications caused by plastic deformations evolved at discontinuities zones significantly diminish the peak pressures and finally led to longer fatigue lives.

5. Conclusions

The methodology presented in ISO 16281:2008 has been involved to evaluate the basic reference rating lives. The favourable effect of roller profile changes as result of the local plastic deformations has revealed.

The elastic-plastic model, presented in first part of the paper, is useful for a more realistic evaluation of the basic reference rating life of roller bearings for normal loading condition, after a high transient overload.

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EVOLUŢIA PROFILULUI ROLEI LA RULMENŢII CU ROLE

II. Evaluarea durabilităţii

(Rezumat)

Pentru condiţii severe de funcţionare (C/P > 0.3) a rulmenţilor cu role, vârfurile de presiune, prezente în zonele de discontinuitate ale profilului rolei, sunt capabile să depăşească limita de elasticitate şi să inducă local modificări ale stării de tensiuni remanente şi de deformaţii plastice. În prima parte lucrarea prezintă validarea experimentală a algoritmului utilizat pentru obţinerea profilului modificat al rolei. În partea a doua a lucrării este prezentată metodologia de determinare a durabilităţii rulmenţilor conform standardului ISO 16281:2008. S-a evidenţiat faptul că profilul rolei modificat prin considerarea deformaţiilor plastice atenuează vârfurile de presiuni apărute la zonele de discontinuitate ale profilului determinând creşteri semnificative ale durabilităţii rulmentului.

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EXEMPLE OF A EXPERT SYSTEM USED IN ENGINEERING DESIGN

BY

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Abstract. This paper presents the structure of a expert system. The economic activities need tools for integrate different aspect of life cycle of products. Shorter production time, shorter design time and also a shorter life time of products, bring designer in front of a problem. How they integrate all the knowledge in a system that can help them. The approach that we present, allow to different trades to bring to the main platform all the personal knowledge that any actor are able to pass to the design process.

Keywords: globalisation; information; data; knowledge; expert system.

1. Introduction

The increasing technical complexity, reducing execution time and diminishing budgets have led to fierce competition in the industry. The situation in the industry can be understood only by re-analyzing the developments in various fields, developments that have changed the company's operations.

The history of the last three decades and the developments of means to transmit the information demonstrate that the companies are often dependent on

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the economic environment. This environment dictates market expectations, customer needs, tracing the path ahead for businesses.

The markets are very versatile; with windows that lasts 12-18 months. This evolution compels companies to produce timely, but also to cope with market instability. The competitive advantage remains of those companies that can capitalize costs related to research.

2. Job Transformation from a Centralized Place into a Delocalized Environment

Increases economic activities taking place in our society, make us dependent from the material resources and energy. Thus, to increase the level of life, it has imposed a growing influence of information and communication technologies that translate into remarkable growth of information in electronic form.

Closed economies were unable to adapt to new technologies and are totally outdated. Economic environment brings forward the concept of producing well at a price as low as possible.

The world is changing. It became smaller and more competitive. Market globalisation forced to increas the interdependence of the world economies, thanks in particular loss border, and liberalization, which led to increased movement of capital and products.

Today, Internet has become an extended public network linking the world. The phone has not produced so many changes that has produced by Internet that allowed the exchange of information between various individuals in time and space. Internet produced the job transformation from a central space to a distributed environment.

In each area regarding a project, there are many programs and methods to increase productivity. The problem is: how do we integrate these modules in specific environments of designers with different skills, so that they can communicate and exchange information on the product or process in order to be done.

3. From Information to Expertise

The design consists of a body of suggested elements that allow to describe a product (shape, size, means of obtaining, etc.) and can give a global answer to a specification (functions that must be provided, operating conditions, the life product you want, environment, etc.) (Tichkiewitch *et al*., 1993).

Knowledge only occurs in an environment that is our environment that is specific to us (our profession skill) (Charlet, 2001). Even if a primary resource, it is closely related concepts of “knowing” and “competence”.

Concretely the information brings various data together to define a fact. The information therefore involves an understanding of the relationship between data. But information does not really pinpoint why those data are and how they have evolved over time. Information is so by nature static and context dependent.

So, the information comes from putting into relation existing data between multiple databases. Behind these relationships are hiding models which can create a dynamic vision of the situation. Understanding these patterns lead to the birth of knowledge.

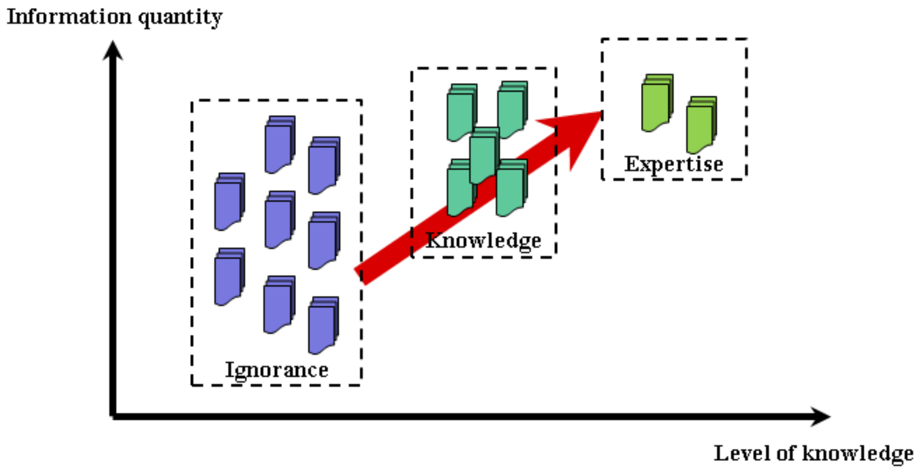

Fig. 1 – Evolution from information to knowledge.

Davenport and Prusak (1998) defined knowledge as a result of thinking: “knowledge is an amalgam of experiences, values, contextual information and insight that form a framework for evaluating and incorporating new experiences and information. They are the result of thinking. In these companies often become embedded in documents, but also in daily routine, in processes, practices and norms in specific enterprise”.

Knowledge is manifested in the form of representations of spirit which it can manipulate and build to reach the whole of cognitive tasks required to achieve the desired purpose. This notion of representation is fundamental. After Richard (1990), cognitive activities (the specific human spirit) “respond to a representation of situation”.

We can consider the information included, assimilated to knowledge, which allows to achieve the projected purpose. The information provides a new perspective for interpretation of an event or object. It acts on knowledge by adding new parameters or the existing structure (Fig. 1).

For Skyrme (1994), the difference between information and knowledge is dependent on the human perception. Even if the knowledge is simplified and submitted as information, they cannot be exploited unless they are filtered through the mind.

4. Extracting Knowledge

Increased economic and scientific activities can be translated into remarkable growth of available information in electronic form, which led to the creation of new tools necessary for analyzing and structuring of documents that allows users to browse them and/or evaluate.

Extracting knowledge consists in extracting the necessary knowledge of a document and represents the structured form. This form can then allow storing information into a database or to be used as a basis for the automatic generation of summaries.

The process of extracting knowledge regardless of the field, involves a number of steps: selection, analysis, processing, data mining and interpretation.

This process is incremental, and the central role is offered for the concepts of information and data.

5. Formulating the Knowledge

The machines are computers that treat information, and they cannot reason alone but only on the basis of knowledge that has already been made.

Different formulations that can be made starting from the same knowledge are not identical. The linking between explicit knowledge and their formulation shows important interest:

– Keep following the provenance information modeling components that enabled a database;

– To maintain the coherence of global resources heterogeneous;

– Exploiting knowledge to better manage an enterprise documents.

Thus formulating knowledge and knowledge are two different things.

Formulating knowledge consists of collections of symbols used to express some knowledge in natural language. The documents are the type of items used to formulate knowledge.

Knowledge modeling expressed in a document in electronic form, requires working with their formulation.

6. Structuring Knowledge

Structuring knowledge consists to identify, document and preserve memory in all activities and all explicit knowledge relative to these activities.

The objective is not to improve, to bring up to date or to enrich, but to identify and preserve knowledge (to identify, locate, model, formula and archive).

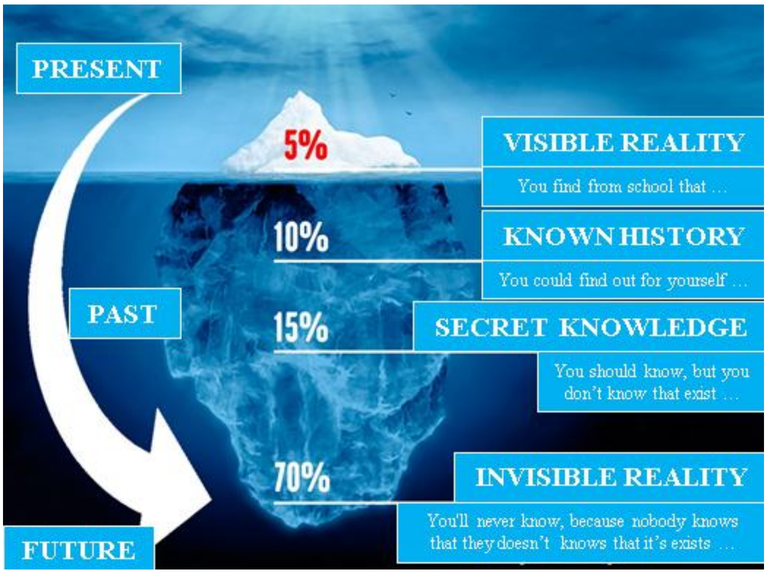

Fig. 2 – Evolution from information to knowledge.

To limit the loss of knowledge, there are companies such as Dassault wishing to formalize gestures transmitted orally until now so they can show new employees (Helderle, 1996).

A temporary rupture is a potential loss factor of knowledge (Duizabo and Guillaume, 1996).

From our point of view, (Tichkiewitch *et al*., 2004), structuring knowledge is now considered a framework in which all its processes as knowledge processes. In this idea, all processes involve: creating, disseminating, and applying knowledge to renew the company's survival.

Two Japanese experts in (Nonaka and Takeuchi, 1997), distinguishes two kinds of knowledge:

– Tacit knowledge;

– Explicit knowledge.

Tacit knowledge is possesses by individual. They are difficult to convey and not formalized. They are competencies and personal experiences, intuition, trade secrets, etc. After Nonaka this type of knowledge are important to initiate a process of creating new knowledge. Explicit knowledge are formalized and submitted as reusable documents like information regarding processes, projects, customers, suppliers, etc. In other words are documents that can be add and/or scanned using a shared information system.

Vinck in (Vinck, 1997) proposes a grouping of knowledge and nonknowledge into two groups: implicit knowledge, explicit knowledge using the iceberg model (Fig. 2).

7. Dissemination of Knowledge

Disseminating knowledge is the most advanced form of management. It is not only a matter of creating initial conditions, encouraging the emergence and local exchange of knowledge, nor of formalizing this knowledge in such a way as to preserve them in a certain activity, but to allow their dissemination and application to different contexts.

The cost of dissemination is also visible in the sense that the corresponding actions and their related costs are identified. On the other hand, the potential gain appears much less identified. The potential benefits and gains that can be expected from knowledge dissemination:

– reduction of errors;
– reduction of redundancies;
– faster problem resolution;
– improved decision-making;
– reduced research and development costs;
– increasing the autonomy of workers;
– improved relationships with clients.

8. Expert System Used in Design

Expert systems are one of the applications of artificial intelligence that have left research labs for use in the corporate world. Many expert systems have been successfully implemented to solve practical problems.

An expert system is proposed in (Fig. 3) and reproduces the behavior of a human expert performing an intellectual task in a specific field. This software capable of performing an expert task (classification, diagnosis, design, planning, etc.) with performances equal to those of the best specialists has some fundamental characteristics:

– expert systems are generally designed to solve classification or decision-making problems;

– expert systems are tools of artificial intelligence, they are used only when no exact algorithmic method is available or practicable;

– an expert system is only conceivable for areas where human experts exist. An expert is someone who knows an area and is more or less capable of transmitting what he knows: for example, a child is not related to his mother tongue.

The structure of an expert system is organized around three main elements: the knowledge base, the fact base, the inference engine.

The knowledge base contains all the information that a human expert would need to carry out his work in a given field. It is the only component of the system that contains knowledge specific to the domain that the system is supposed to cover.

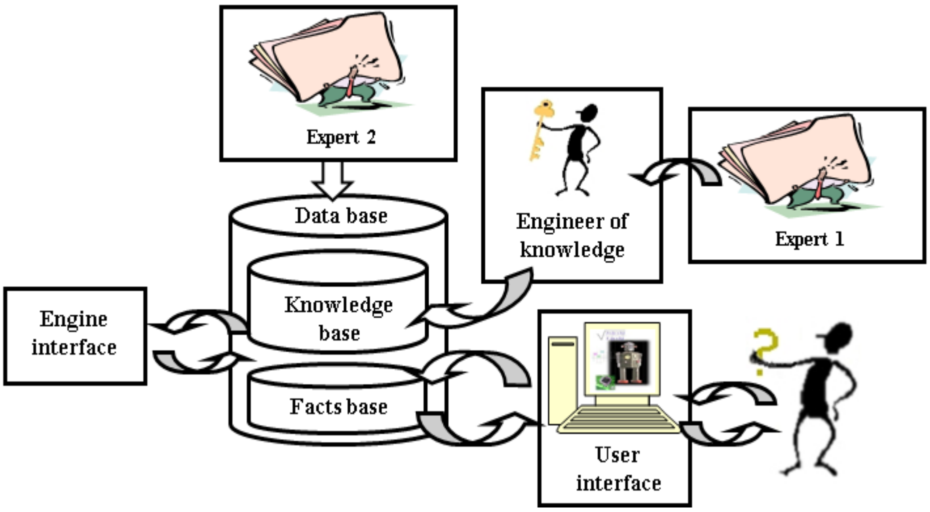

Fig. 3 – Expert system structure.

Secondly, the basis of facts, that is, the state of the current problem, may be more or less complex. It contains mainly the initial data, a particular context, the different processes already executed on these data, the data acquired by the processes. It will therefore be represented in multiple ways: by files, property lists, objects.

The third component is used to exploit this knowledge. An inference engine is required to relate the description of a problem to the analytical capabilities of a given situation.

9. Conclusions

In general, and thanks to the structuring of the knowledge base, the inference engine will be able to answer questions, reason and draw the consequences implied by the knowledge included in the system.

A major peculiarity in the architecture of expert systems lies in the clear separation between the knowledge base and the inference engine, the mechanism by which this knowledge is exploited.

The tools developed for the help of engineers are tools that allow to the actors, through computer models, to manipulate the knowledge available on the project. The objective is not to realize an automatic design, but to allow external actors (designers) to reflect on their main problems.

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EXEMPLU DE SISTEM EXPERT UTILIZAT ÎN PROIECTARE

(Rezumat)

Instrumentele dezvoltate pentru ajutorul inginerilor sunt instrumente care permit actorilor, prin modele computerizate, de a manipula cunoștințele disponibile cu privire la proiect. Obiectivul nu este de a realiza o proiectare automată, ci pentru a permite actorilor externi (proiectanți), de a reflecta asupra principalelor probleme.

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THE SMALL-MEDIUM ENTERPRISES ROLE IN THE ROMANIAN ECONOMY

BY

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Abstract. This paper presents the role that it plays in the economy of Romanian SMEs. Their role is vital both for the development and maintenance of Romania's economic performance. SMEs represent and lead player of supply on the labor market.

Keywords: SME; economic environment; management; development; human resources.

1. Introduction

Globalization is the process of denationalization of markets, policies and legal systems, as well as the growth of the global economy. The consequences of these political and economic restructurings on people, local economies and the environment are subjects of open discussion at the level of international organizations, governments or the academic world.

The international interdependence of markets is reflected in the fact that an economic event located in one country can affect other economies, mainly through trade and financial flows. International integration of economies has been strengthened, channels of transmission of shocks have been diversified,

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imposing on States to adapt their economic action and increasingly raising the problem of international cooperation. The era of mass production is at the end of his life. Companies that are not interested in customer needs are starting to decline. It is that decline is the engine of the changes that will be made.

The response to this situation sees the emergence of new concepts in the company with the introduction of project management and total quality control. Training employees in these new techniques is a key motivator for employers. The concentration of functions in departments leads to a separation between projects, waterproof structures where there is no communication. The new technological solutions implemented help to increase the flow of information. Local networks, general relational database systems, increasingly powerful workstations, computers using operating systems lead to a real liberalization of IT tools. They find their place perfectly in the organization of the company, but the management of the equipment becomes more difficult to control.

2. The Role of SMEs in the Romanian Society

Small and medium enterprises represent a very important factor in development in every nation's economy. Their benefits consist of labor flexibility in their close relationship with the local environment. In a post-industrial global economy, consumers are oriented towards standard products, but at the same time, we can observe a marked inclination towards consumption of customized products and services and the growing requirement for appearance and product quality. This trend, albeit still limited, appears in poorer countries and in traditional activities sectors. It can be said that there is opportunity for innovative activities and the international specialization of SMEs in Romania can be identified more in the traditional sectors than in cutting-edge industries.

With their small size, SMEs atomicity enhances the character of the market, reducing power of influence of the large enterprises. In addition, their large numbers stimulates competition by limiting the monopoly positions of large enterprises and thus decreasing their ability to raise prices. Their ability to respond promptly to local needs, thanks to the flexibility exhibited by, SMEs often end up acting in local markets more effectively than larger operators (Archibugi and Lundvall, 2001).

The importance and topicality of the research, results from the position of SMEs in the private sector of developed countries and those in developing countries.

Improving access to finance the SMEs play a crucial role in promoting entrepreneurship and enhance competition in the Member States of the European Union (Parrott *et al.*, 2010). Access to sufficient and adequate capital to grow and develop is one of the main obstacles facing most European SMEs. This situation is compounded by the difficulties faced by SMEs in relation to financial intermediaries who consider their funding as an activity with a high risk and low profitability.

3. The Overall Evolution of the SME Sector in 2004-2009

In Table 1 we summarized the evolution of main macroeconomic indicators, in order to outline the economic and social sector in which SMEs operate.

The main factors influencing this growth continued to increase domestic consumption of goods and services and increasing investment flows to Romania, especially in the first half of 2007, as shown in the Annual Report BNR mentioned as the main source of macroeconomic data.

Table 1

The Evolution of Main Macroeconomic Indicators in Romania, in 2004-2009 ***

Indicators	2004	2005	2006	2007	2008	2009
Total GDP (mil lei)	246469	288176	344536	404709	503958	467673
GDP annual growth rate (%)	8.5	4.4	7.9	6	7.1	-7.2
GDP/capita (lei/inhabitant)	11018	13327	15962	18791	23440	21752
Inflation rate (%)	11.9	9	6.6	4.9	6.3	4.74
Employed persons (thousands)	9158	9147	9313	9353	-	-
Average number of employees (thousands)	4469	4559	4667	4720	4806	4594
Employees in the private sector (thousand)	2259	2575	2726	-	-	-
Total unemployed (thousand)	558	523	460	368	403	709
Unemployment rate (%)	6.3	5.9	5.2	4.1	4.4	7.8
Foreign Trade balance (millions Euro)	-5323	-7806	-11759	-17586	-18372	-6754
External Debt (million)	18298	24641	28628	36728	51761	64207
Surplus/Deficit Budget (million)	-3693	-2268	-5099	-9448	-24654	-36400

Macroeconomic indicators whose evolution I caught her in Table 1 outlines two major economic phenomena in the period under review: on the one hand economic and social context were favorable, timely creation of SMEs, on the other hand the momentum and development of these types of businesses have generated significant economic benefits for the entire economy.

Despite the difficulties, Romania has experienced a significant process of economic growth, as reflected in GDP growth in the period under review. Thus, in absolute GDP increased from 246.469 million lei in 2004 to about 503.959 million lei at the end of 2008. The real GDP growth was permanently located above the EU average, but had fluctuated, with a peak growth in 2004 (8.4%) and a minimum one year later (4, 4%) in 2005. For 2008, the growth rate of GDP was 7.1%.

4. Conclusions

Interaction with others, active participation in resource development and recognition of individual human performance by others, may be grounds to study and acquire professional development. Improve individual performances are by extension, organizational results favorable. The ability to generate and use knowledge and innovation are the main sources of growth and competitive advantage.

The increased flexibility of SMEs, the entrepreneur permanent contact with the organization the ability to produce goods and services to satisfy different needs and demands, organizational environment favorable to change and innovation are elements that explain the increased performance of the SME sector.

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ROLUL IMM-URILOR ÎN SOCIETATEA ROMÂNEASCĂ

(Rezumat)

Această lucrare prezintă rolul pe care îl joacă în economia românească IMM-urile. Rolul lor este esențial atât pentru dezvoltarea cât și pentru menținerea performanței economice din România. IMM-urile reprezintă și jucatorul principal al ofertei pe piața forței de muncă.

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BOND-GRAPH METHOD IN THE PLASTIC ANALYSIS OF STATICALLY INDETERMINATE SYSTEMS

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Abstract. The work presents the possibility of finding the failure mechanisms with their corresponding ultimate forces for a continuous beam, by using the bond-graph method. The results accuracy is demonstrated by applying a classical method, based on the Virtual Work Principle. The bond-graph method is validated once again for the analysis of different kind of systems subjected to static and dynamic actions.

Keywords: bond-graph; failure mechanism; ultimate load; fully plastic moment; continuous beam.

1. Introduction

In case of statically indeterminate beams made of ductile materials, like steel, their plastic analysis presumes to identify the ultimate load, that is the minimum load which transforms the system into a mechanism with one degree of freedom. For this reason, all possible failure mechanisms are constructed and for each of them, the corresponding load is assessed (Ibănescu and Toma, 2013).

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In case of statically indeterminate beams to the "n[th]" degree, the occurrence of "n+1" plastic hinges lead to a mechanism with one degree of freedom. There are several possible locations of these hinges, each case being characterized by a corresponding value of the ultimate load. The real collapse mechanism is that one for which the minimum ultimate load is obtained.

2. Ultimate Load Assessment by Using the Virtual Work Principle

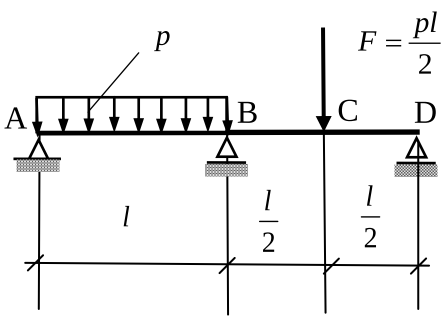

Fig. 1 – Statically indeterminate beam.

Let us consider a continuous beam (Fig. 1), acted by a distributed load, *p* in intensity, and a concentrate load *F*, whose magnitude depends on the same parameter *p*.

The beam is statically indeterminate to the first degree, which means that two plastic hinges transforms the beam into a mechanism.

There are three distinct possible failure mechanisms.

A first possible collapse mechanism is produced by two plastic hinges

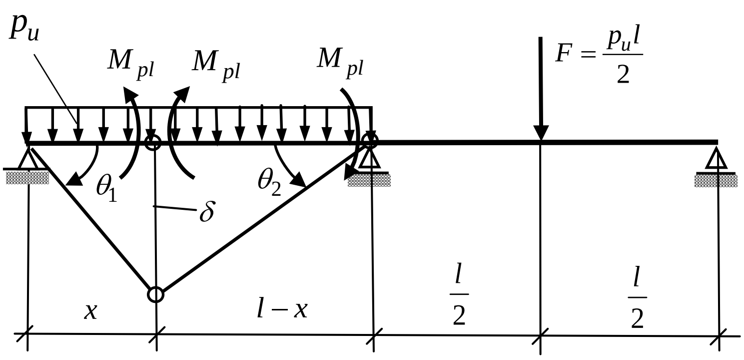

Fig. 2 – Failure mechanism no. 1.

(sections entirely plasticised) located at section B and somewhere between A and B, along the portion acted by the distributed load (Fig. 2).

For finding the ultimate load corresponding to this mechanism, we apply the Virtual Work Principle, that is, the virtual work done by real forces on virtual displacements, consistent with the system constraints, equals zero. It leads on the following equation:

$$-M_{pl}\theta_1 - 2M_{pl}\theta_2 + p_u x\frac{\delta}{2} + p_u(l-x)\frac{\delta}{2} = 0 \quad (1)$$

where M_{pl} is the fully plastic moment - the bending moment which develops on a section which is entirely plasticised.

The virtual displacements are expressed in terms of a unique displacement, δ.

$$\theta_1 = \frac{\delta}{x} \qquad \theta_2 = \frac{\delta}{l-x} \quad (2)$$

The ultimate load p_u depend on x, the abscissa of the plastic hinge located along the part of the beam acted by the distributed load.

$$p_u = M_{pl}\frac{2(l+x)}{lx(l-x)} \quad (3)$$

As the ultimate load is the minimum load for which the beam becomes a mechanism with one degree of freedom, x will be obtained from the following equation:

$$\frac{dp_u}{dx} = 0 \quad (4)$$

The final shape of this equation is:

$$x^2 + 2lx - l^2 = 0 \quad (5)$$

whose solutions are:

$$x_1 = -l(1-\sqrt{2}) \quad \text{and} \quad x_2 = -l(1+\sqrt{2}) \quad (6)$$

The single valid solution for our case is:

$$x_1 = -l(1-\sqrt{2}) = 0.414l \quad (7)$$

The ultimate load is determined by substituting x_1 in relation (3).

The final result has the form:

$$p_u^{(1)} = 11.65\frac{M_{pl}}{l^2} \quad (8)$$

Another possibility for the location of plastic hinges is at section B and C. The corresponding failure mechanism is pictured in Fig. 3.

The same procedure is followed and the equation provided by the application of the Virtual Work Principle is now:

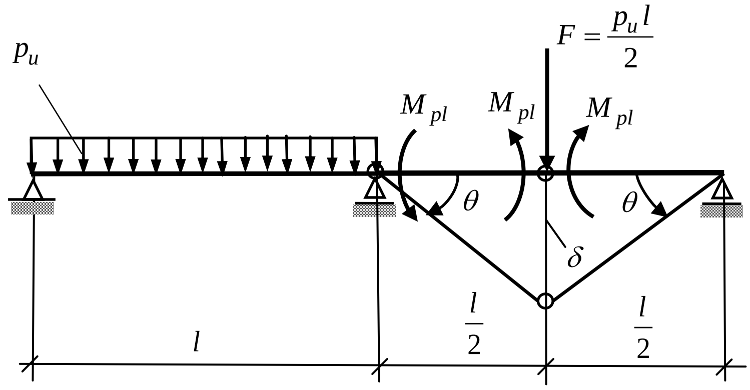

Fig. 3 – Failure mechanism no. 2.

$$3M_{pl}\theta + p_u\frac{l}{2}\delta = 0 \tag{9}$$

The linear virtual displacement δ is expressed in terms of angular displacement θ:

$$\delta = \theta\frac{l}{2} \tag{10}$$

In these circumstances, the ultimate load for this second case becomes:

$$p_u^{(2)} = 12\frac{M_{pl}}{l^2} \tag{11}$$

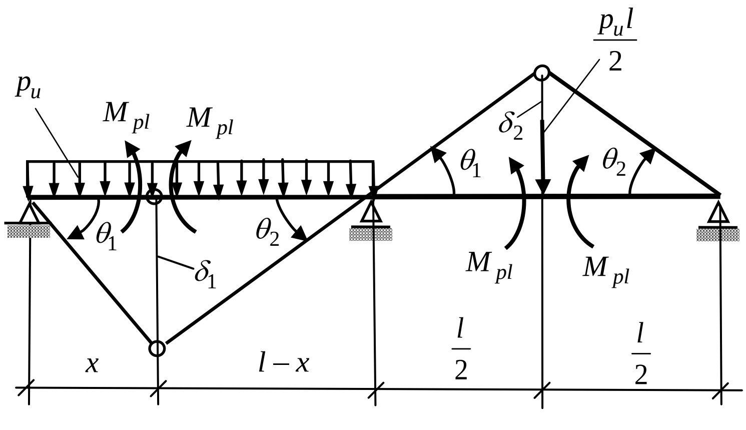

Fig. 4 – Failure mechanism no. 3.

The third and last possible collapse mechanism is that one with a plastic hinge somewhere between section A and B and the other one at section C (Fig. 4).

The virtual work depends once again on *x*, the abscissa of the section between A and B where one plastic hinge occurs:

$$-M_{pl}\theta_1 - M_{pl}\theta_2 + p_u x\frac{\delta_1}{2} + p_u(l-x)\frac{\delta_1}{2} + 2M_{pl}\theta_2 - p_u\frac{l}{2}\delta_2 = 0 \tag{12}$$

Taking into account that

$$\delta_1 = \theta_1 x, \quad \delta_2 = \theta_2(l-x) \text{ and } \theta_1 = \theta_2\frac{l-x}{x} \tag{13}$$

it results:

$$p_u = \frac{4M_{pl}}{xl} \tag{14}$$

The minimum value of the load is obtained for x tending to infinity, which is not a possible solution.

The conclusion of the analysis is that the real collapse mechanism is the first one, because the ultimate load has the lowest obtained value, presented in relation (8).

3. Ultimate Load Assessment by Using the Bond-Graph Method

An alternative manner for finding the ultimate loads corresponding to each collapse mechanism is the bond-graph method. This method can be applied not only to dynamic problems (Borutzky, 2010; Ibănescu and Ungureanu, 2015; Ibănescu and Ibănescu 2016; Ibănescu, 2017), but also to static problems (Ibănescu and Ibănescu, 2004; Ibănescu and Ibănescu, 2014), as further presented.

The procedure consists in considering the deflected beam as a mechanism with dynamic characteristics, for which the bond-graph model is constructed. This model leads to mechanism dynamic equation. The equilibrium equation is obtained by considering the inertial terms equal to zero.

The three possible failure mechanisms previously presented have been studied.

The associated mechanism for the first possible failure mechanism presented in Fig. 2, is pictured in Fig. 5 and its bond-graph model in Fig. 6. ω_1 and ω_2 are the angular velocities of the bars and J_1 and J_2 are the moments of inertia of bars with respect to their fixed ends. The fully plastic moments and the loads applied upon the beam are modelled as sources of effort. The connection between ω_1 and ω_2 is done by a transformer, having the parameter $\frac{x}{l-x}$. This parameter results from the two possibilities of expressing the mobile hinge velocity, *v*, that is $v = \omega_1 x = \omega_2(x-l)$, which is valid only in case of small displacements.

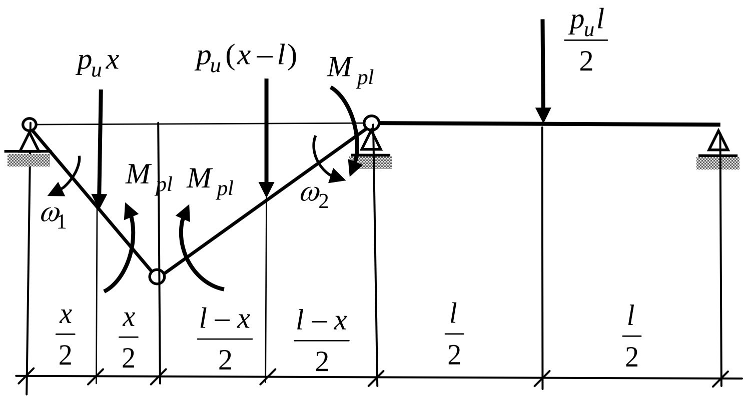

Fig. 5 – Associated mechanism for case no. 1.

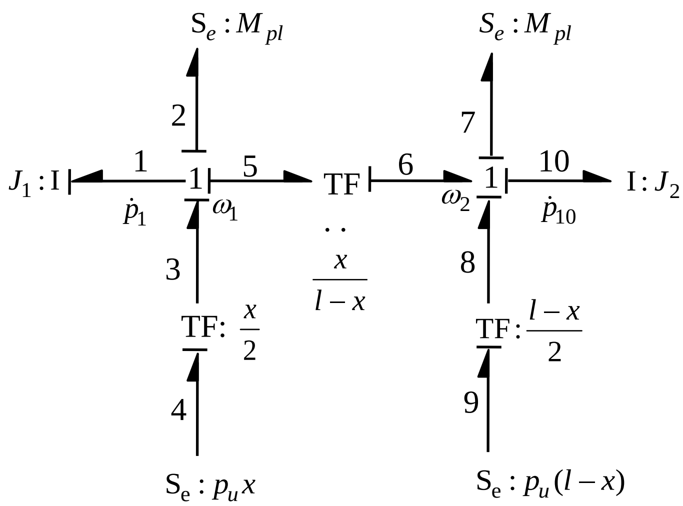

Fig. 6 – Bond-graph model for case no. 1.

Based on bond-graph methodology, the following motion equation results:

$$\dot{p}_1 = -M_{pl} + \frac{p_u x^2}{2} - \frac{2x}{l-x} M_{pl} - \frac{x}{l-x} \dot{p}_{11} + p_u \frac{x(l-x)}{2} \tag{15}$$

where $\dot{p}_1$ and $\dot{p}_{11}$ are the derivatives with respect to time of the bars angular moments about their fixed ends. In case of static equilibrium, these derivatives are naught. The inertial forces are not considered, because they have no function in the equilibrium equation, like the previously mentioned derivatives.

In these circumstances, the equilibrium equation has the form:

$$0 = -M_{pl} + \frac{p_u x^2}{2} - \frac{2x}{l-x} M_{pl} + p_u \frac{x(l-x)}{2} \tag{16}$$

which leads to the same solution for the ultimate load, given in Eq. (3), obtained by using the Virtual Work Principle.

For the second case the mechanism and the bond-graph model are presented in Fig. 7 and Fig. 8, respectively.

The motion equation of the mechanism, obtained from the bond-graph

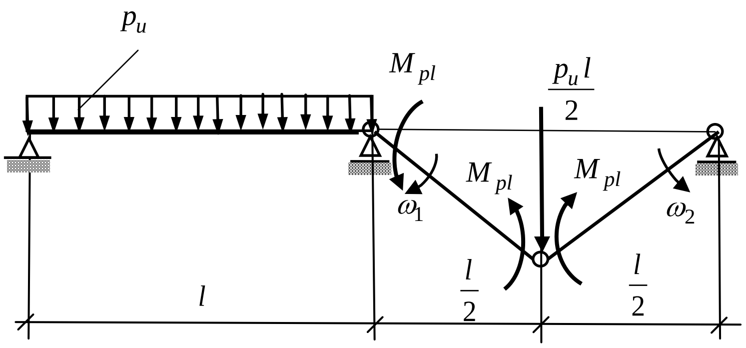

Fig. 7 – Associated mechanism for case no. 2.

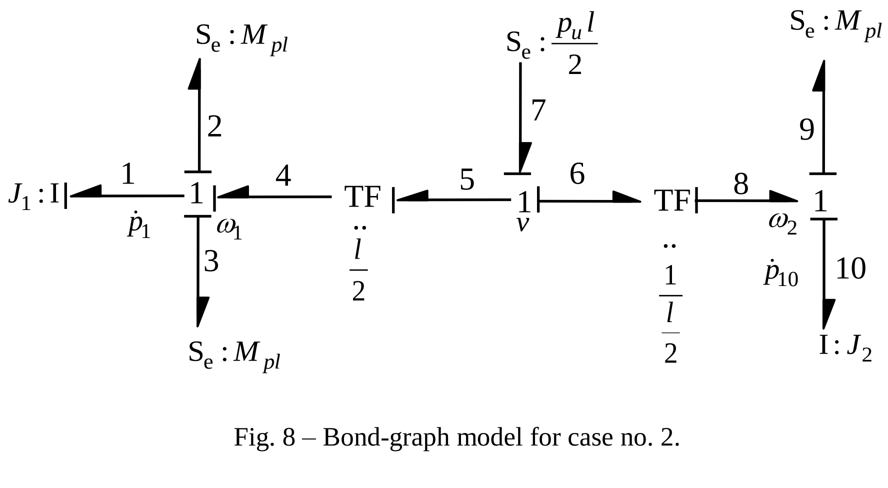

Fig. 8 – Bond-graph model for case no. 2.

model is:

$$\dot{p}_1 = -2M_{pl} + \frac{p_u l^2}{4} - M_{pl} + \dot{p}_{10} \tag{17}$$

After making the derivatives $\dot{p}_1$ and $\dot{p}_{10}$ equal to zero, the following equilibrium equation results:

$$0 = -2M_{pl} + \frac{p_u l^2}{4} - M_{pl} \tag{18}$$

The ultimate load obtained from Eq. (18) has exactly the same shape as presented in Eq. (11).

The same procedure is followed for the last possible failure mechanism, presented in Fig. 4. The corresponding mechanism and the bond-graph model are pictured in Fig. 9 and Fig. 10.

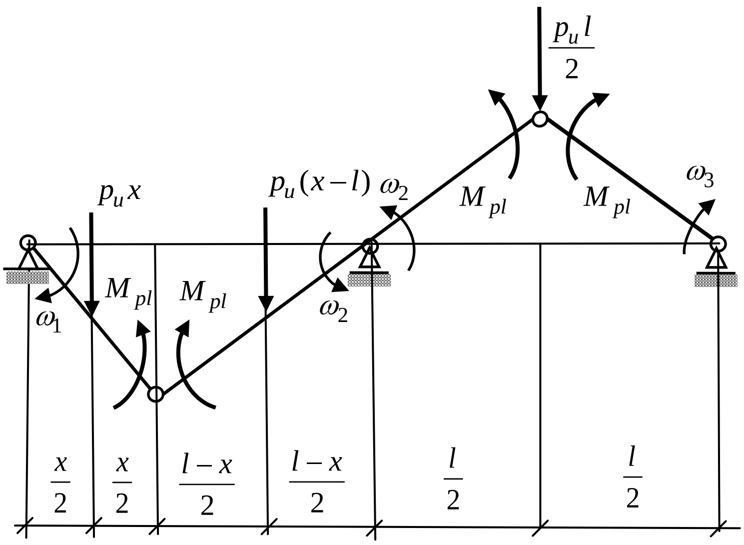

Fig. 9 – Associated mechanism for case no. 3.

In this last case, the motion equation and the equilibrium equation are:

$$\begin{aligned}\dot{p}_1 = -M_{pl} + \frac{p_u x^2}{2} - \frac{p_u l^2}{4}\frac{x}{l-x} + M_{pl}\frac{x}{l-x} + \dot{p}_{18}\frac{x}{l-x} + \\ + \dot{p}_{11}\frac{x}{l-x} + p_u\frac{x(l-x)}{2}\end{aligned} \tag{19}$$

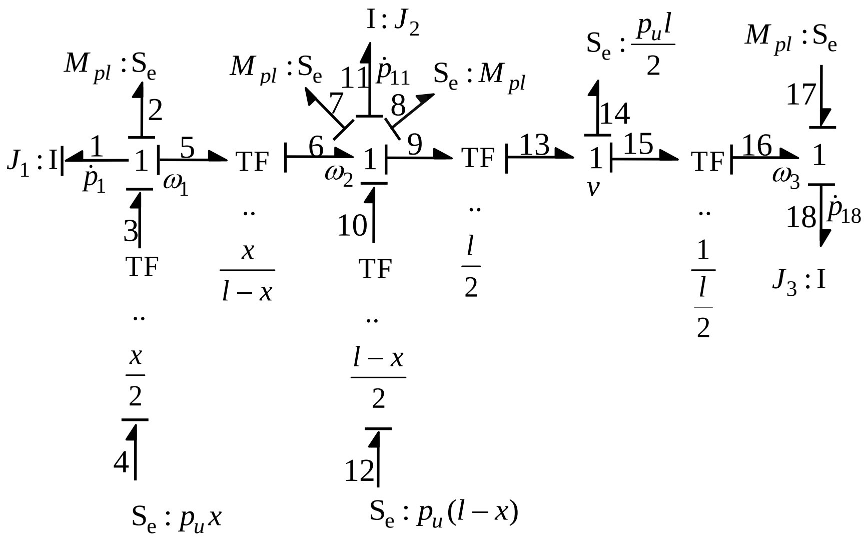

Fig. 10 – Bond-graph model for case no. 3.

$$0=-M_{pl}+\frac{p_u x^2}{2}-\frac{p_u l^2}{4}\frac{x}{l-x}+M_{pl}\frac{x}{l-x}+p_u\frac{x(l-x)}{2} \tag{20}$$

The solution of the Eq. (20) coincides with that one derived by applying the Virtual Work Principle (see Eq. (14)).

4. Conclusions

1. The bond-graph method is a trustful procedure for finding the ultimate loads in case of different statically indeterminate systems. It has been proved that it leads to the same results as those obtained by using the classical Strength of Materials methods.

2. The bond graph method represents a simple method for solving large categories of plastic analysis problems, offering the possibility of checking the results provided by other methods.

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METODA BOND-GRAPH ÎN ANALIZA PLASTICĂ A SISTEMELOR STATIC NEDETERMINATE

(Rezumat)

Lucrarea prezintă modul posibil de evaluare a mecanismelor de cedare și a forțelor ultime corespunzătoare pentru o grindă continuă, folosind metoda bond-graph. Corectitudinea rezultatelor este demonstrată prin compararea lor cu cele obținute prin metode clasice, cum ar fi aplicarea principiului lucrului mecanic virtual. Astfel, se obține încă o dată validarea acestei metode de calcul pentru analiza diferitelor sisteme, supuse acțiunilor statice sau dinamice.

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THE EXPERIMENTAL PLANNING FOR SINGLE POINT INCREMENTAL FORMING USING *LATIN* SQUARE METHOD

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Abstract. Incremental sheet forming is a new technical forming method which can provide a high accuracy of products, at short time and low production costs. Single point incremental sheet forming is a series innovative processes, flexible, which it can be obtaine various geometries without the special tool geometry. Incremental forming term is used for a variety processes, characterized by the fact that all local deformation area moves over the entire product surface. The present work is going to be focuses on the experimental planning in order to obtain a product through single point incremental sheet forming using *latin* square method.

Keywords: experimental planning; single point incremental forming; *latin* square.

1. Introduction

Single point incremental sheet forming method (Fig. 1) is an innovative forming approach method for sheet materials which it is using CNC milling machines tools, which can be applicable for prototypes production.

At single point incremental sheet forming process the blank is clamped on clamping device and the tool performs a rotational movement and also the

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approach movement (Tisza *et al*., 2013). The most important criterion to express the material plasticity limit at single point incremental sheet forming process is the maximum forming angle (Radu, 2011).

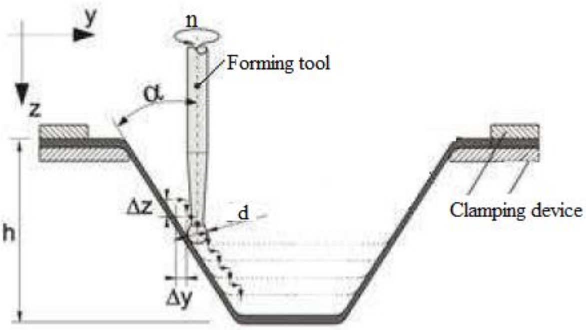

Fig. 1 – The basic principle for single point incremental sheet forming process (Tisza *et al*., 2013).

The significances of notations from Fig. 1 are the following: d – tool diameter; α – wall angle; h – workpiece height; n – rotational tool movement Δz – tool movement one Z axis; Δy – tool movement one Y axis.

The main characteristics of single point incremental sheet forming are the following:

– at single point incremental sheet forming process, we don't need a die, but we need a blank clamping device system, various tools dimensions, forming rollers, active plate etc;

– single point incremental sheet forming process requires a lot of time in comparation with conventional forming, but doesn't require high costs with equipments;

– single point incremental sheet forming process has a high flexibility, the same equipment can be used for perform various parts geometries;

– forming angle achieved is bigger than conventional forming, this thing make suitable to work with different materials which are hard for forming;

– the springback effect decreases the accuracy for conventional forming, but using single point incremental sheet forming process this disadvantage will be removed (Tisza *et al*., 2013).

Is very important to know that this process works with a biggest rapport weight/strength (Naga *et al*., 2012). To get desired final shape, single point incremental sheet forming will use a tool with hemispherical shape, where it has a trajectory predefined for deformation locally.

The blank material to get the parts through single point incremental sheet forming process is determined by technological factors and conditions for exploiting of parts. Technological factors characterized by stretching material properties, the maximum permissible deformations, or future machining operation and finishing (turning, milling, galvanization, polishing, etc.) or possibility to be assembled with another parts (soldering, welding, riveting). Ferrous and non-ferrous materials used for manufacturing the parts by single point incremental sheet forming process are standardized and delivered like thin sheets, thick plates, cover plate and strips of various sizes (Teodorescu, 1987; Iliescu, 1987).

Single point incremental sheet forming is a new process recently appeared, from this reason is not very well presented in specialty literature, regarding the deformation conditions for some types of materials, concerning the advantages which it has and potential industrial applications. More than that publications issued, presents some limited results (Jackson *et al*., 2008), which makes it necessary to continue research in this area. The studies regarding this process focus on three main directions: measuring the deformations and displacements produced on sheet, estimation the deformations using the finite element method and measure the deformation forces (Jackson *et al*., 2008).

2. *Latin* Square Method

In experimental research are using various methods like Taguchi method (Nedelcu *et al*., 2009; Dave *et al*., 2012), least squares method (Gramescu *et al*., 2014) and others. Above method takes its name from the *latin* alphabet, used for the first time when was described the plan, (http://designtheory.org/; http://www.isogenic.info/; https://onlinecourses) (Table 1). This kind of plan is recommended when test in parallel three independent variables, each with several degrees of variation. The advantage is that reducing the number of experimental conditions analyzed from K_n conditions at Kxn conditions where, n represent the number of variables considered for the study and k the modalities number of variables (http://www.statsdirect.com/; https://www.ilri.org/; http://personal).

Table 1
Presentation of Latin Square Method

	A_1	A_2	A_3
B_1	C_1	C_2	C_3
B_2	C_1	C_2	C_3
B_3	C_1	C_2	C_3

Latin square allows experiments planning to evaluate k factors effects at a number of levels variation p > 2, (http://www.statsdirect.com/; https://www.ilri.org/; http://personal). *Latin* square method provides information when the interactions effects between factors are smaller than the effects made by main factors, and experimental errors follows a normal distribution. Latin square method by order n is a matrix composed type n x n where n is a distinct symbol, each cell belonging to *latin* letters, once on line or on column, (http://www.statsdirect.com/; https://www.ilri.org/; http://personal). For example: for an factorial experiment involving three influencing (A, B, C) at three variation levels, through 3x3 *latin* square type plan, the experiences number which can be obtained the informations is only 9 (Table 2), instead of 27 according to a factorial plan complete (3^3), (http://designtheory.org/; http://www.isogenic.info/; https://onlinecourses).

Table 2
Example for Experimental Plan According 3x3 Latin Square Plan

	A_1	A_2	A_3
B_1	C_1 (1)	C_2 (2)	C_3 (3)
B_2	C_2 (4)	C_3 (5)	C_1 (6)
B_3	C_3 (7)	C_1 (8)	C_2 (9)

The terms significance from Table 2 are the following: the number from brackets is each experiment number; first matrix line is A factor for all 3 levels; the first matrix column is B factor for all 3 levels and C factor is written like *latin* square. In industrial experiments, a variable often rely on time units. The other variable may be represented by machines or operators (http://www.statsdirect.com/; https://www.ilri.org/; http://personal).

3. The Experiment Planning

For experiments tests will be used CNC machine Akira Seiki SR3 XP (Fig. 2) from Fine Mechanic and Nanotechnologies laboratory, Department of Machine Manufacturing Technology, “Gheorghe Asachi” Technical University of Iași. Some characteristics of machining center are presented in Table 3.

Table 3
Akira Seiki SR3 XP Parameters (*** *Equipment Guide*)

Parameters	U.M.	Values
X/Y/Z Travel	mm	762/ 410/ 460
Table dimensions	mm	910x380
Spindle	rpm	9000
Power	HP	12
Cutting speed	m/min	36/36/30

The tool used at experimental tests is a tool with spherical peak shape (Fig. 2) from X163CrMoV12 material and the hardness is 57 HRC. The peak tool diameter is 9 mm and the length is 100 mm.

Fig. 2 – The tool which will be used at single point incremental sheet forming process.

For fixing the tool on CNC milling machine tool will choose a tool holder with collet and for fixing the sheet blank will be use the clamping device from Fig. 3, which is composed from the following elements: motherboard which is clamping on CNC milling machine tool table; support plate; 4 supports; sheet blank clamping plate and 16 fixing screws, 4 nuts. For fixing the clamping device on CNC milling machine tool will be achieved by means of clamps.

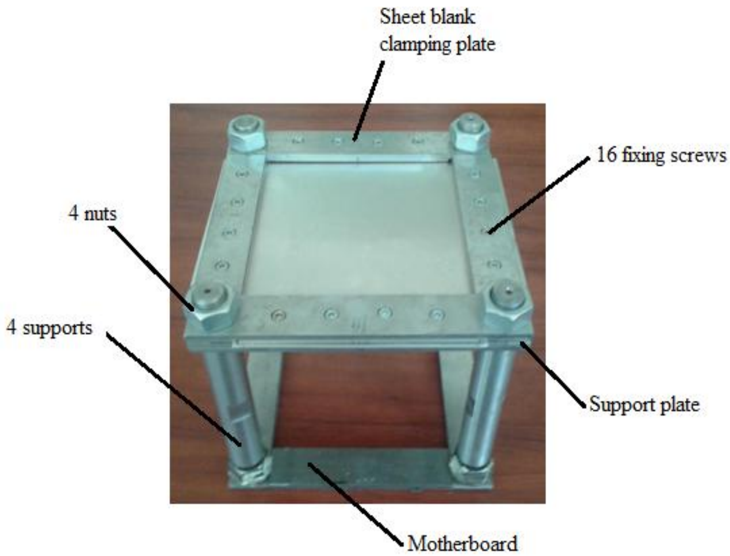

Fig. 3 – Sheet blank clamping device.

The sheet blank material is AlMg3 EN 5754 H111 with thickness 1 mm. During the experimental tests will follow 3 influence factors (A, B, C), using *latin* square method 3x3 type. This influencing factors are presented in Table 4.

Table 4
The Experimental Plan Using the Latin Square Method

		Wall angle 5º	Wall angle 10º	Wall angle 15º
Experiment 1	Speed 1500 rpm	Forming depth 0.05 mm	Forming depth 0.055 mm	Forming depth 0.06 mm
Experiment 2	Speed 3000 rpm	Forming depth 0.065 mm	Forming depth 0.07 mm	Forming depth 0.075 mm
Experiment 3	Speed 6000 rpm	Forming depth 0.08 mm	Forming depth 0.085 mm	Forming depth 0.09 mm

4. Conclusions

For single point incremental sheet forming process isn't used a die, but it is need the clamping device for fixing the sheet blank, a variety tools, forming rollers, active plates, etc. In comparison with classic forming, single point incremental sheet forming need a lot of time but doesn't require high costs with equipments.

Latin square method has the following advantages, as follows: significant reduction of experiments number to be performed is easily method to be analyzed.

Also, the method requires that the number of experiments, rows and columns to be the same and each experiment to be approximate the same in each row and column.

The blank material chosen for experimental research is very little used in specialty literature to achieve various parts and technological parameters chosen are easily to be varied on machine tool, also in software package to simulate the process.

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PLANIFICAREA EXPERIMENTULUI PENTRU AMBUTISAREA INCREMENTALĂ ÎNTR-UN SINGUR PUNCT UTILIZÂND METODA PĂTRATULUI *LATIN*

(Rezumat)

Ambutisarea incrementală este o nouă tehnică de ambutisare care poate oferi o precizie ridicată a produselor, într-un timp foarte scurt şi costuri de producţie reduse. Ambutisarea incrementală reprezintă o serie de procese inovative, flexibile, în care se pot obține diferite geometrii fară a fi necesară o geometrie specială a sculei (Tisza *et al.*, 2013). Termenul de ambutisare incrementală este utilizat pentru o varietate de procese, toate caracterizate prin faptul că zona de deformare locală se deplasează pe toată suprafața produsului. Lucrarea prezintă planificarea experimentelor în vederea obţinerii unui produs prin ambutisare incrementală într-un singur punct utilizând metoda pătratului *latin.*