The 4th International Conference
"Advanced Composite Materials Engineering"
COMAT 2012
18- 20 October 2012, Brasov, Romania

METHOD TO DETERMINE THE TOTAL DEFORMATIONS OF A CONNECTING-ROD FROM A SINGLE CYLINDER ENGINE

P. Grigore[1], I. Todor[1]
[1] *Transilvania* University of Brasov, ROMANIA, [1]paul.grigore@unitbv.ro

Abstract: *In case of an internal combustion piston engine, gas pressure force is transmitted from the piston to the crankshaft through the connecting-rod, which serves to transform the translational motion of the piston into rotary motion of the crankshaft. During engine operation, the stress of connecting-rod are those given by compression and buckling loads of gas pressure force, tensile and compression loads of inertia force of piston group. The main objective of this article is to show how can be determined the total deformations of a connecting-rod for a given single cylinder engine, using finite element analysis.*
Keywords: *connecting rod, stresses, deformations.*

1. INTRODUCTION

Deformations produced by gas pressure force on connecting rod body, tend to minimize the distance *x* (Figure 1-a) between conn-rod small-end and big-end axes. At the same time, the effect of buckling (Figure 1-b) compromising their axes parallel, resulting in bearing wear and loosening parts from motor mechanism. Under axial forces of inertia, the bearings of conn-rod small-end and big-end tend to be out of roundness (Figure 1-c). The bending stress of connecting rod is given by tangential forces of inertia (Figure 1-d). An additional cause occurrence of this stress may be eccentric of compression force F_p (Figure 1-e), determined by radial clearance between the pin and bush conn-rod small end.

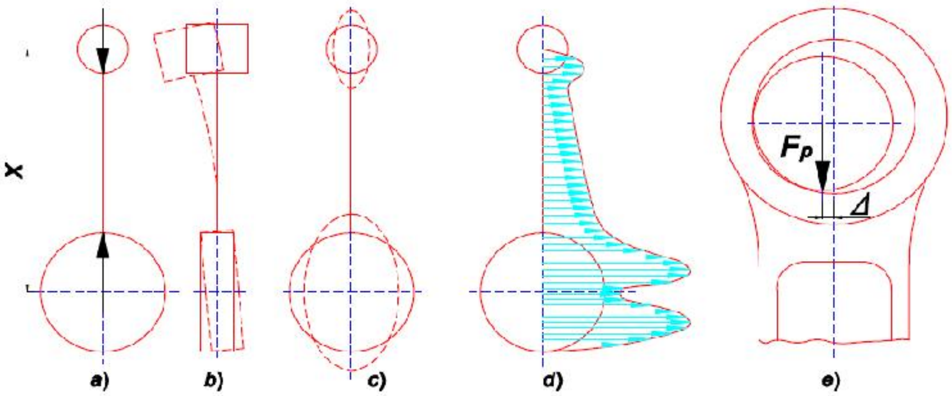


Figure 1: Deformations of the connecting rod

This article will present how can be determined the total deformations on tensile stress of a connecting-rod using finite element analysis. The conn-rod is from a single cylinder engine for wich are given:

- Rated power: P_n = 12,2 [kW];
- Nominal velocity (angular): n_n = 3000 [rot/min];
- Number of cilinders: i = 1;
- Engine type: SI;

- Number of strokes per cycle: $\tau = 4$;
- Piston stroke: $S = 85$ [mm];
- Cilinder bore: $D = 97$ [mm].

2. ANALYSIS OF CONNECTING-ROD

Tensile force F_t (Figure 2-a) has the maximum value when the piston is at TDC (the beginning of the intake stroke), so gas pressure force is negligible. Unit stress on outer and inner fiber of conn-rod small end ($\sigma^t_{\varphi e}$, $\sigma^t_{\varphi i}$), given by force F_t, varies with section (Figure 2-b), which reached maximum values in different sections (first in a vertical section, the second in a section located at 90° to the first).

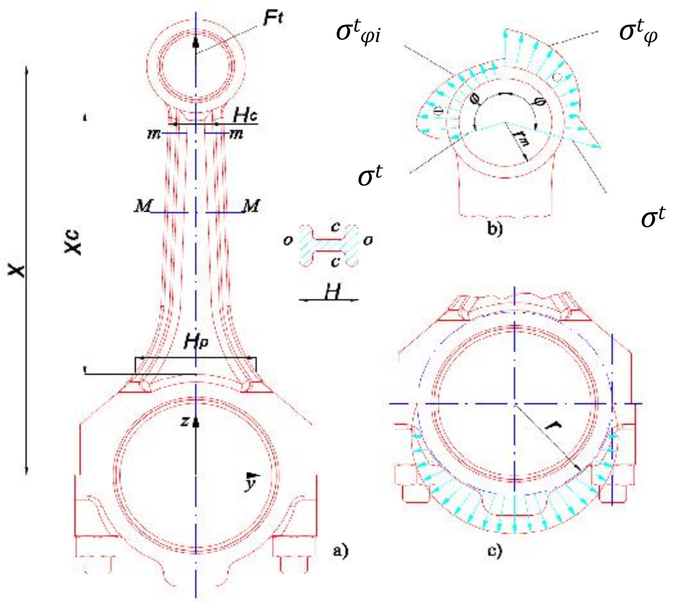


Figure 2: Calculation scheme for tensile stress:

a) characteristic dimensions of conn-rod body;
b) conn-rod small end;
c) conn-rod big end.

For FEM analysis with ANSYS, tensile force is evenly distributed on the upper half of conn-rod small end (Figure 3). Maximum introduced corresponds to the force of inertia calculated with the equation [2, 3, 4]:

$$F_t = -m_p \cdot r_m \cdot \omega^2 \cdot (1+\lambda_b) = -1648{,}332[N] \tag{1}$$

where: $m_p = 0{,}813$[kg] – piston group mass;
$r_m = 0{,}0155$[m] – average radius of the connecting rod small end (figure 2-b);
$\omega = 314{,}159$[rad/s] – angular velocity of the crankshaft (for $n = 3000$[rot/min]);
$\lambda_b = 0{,}3012$ – crank radius /rod length.

Regarding the conn-rod big end, tensile force is distributed on its lower half (rod cap) and unit stress are determined only for the outer fiber (Figure 2-c). In this case, tensile force is greater, because in (1) is introduced the body mass rod. Thus, the force F_t will be [2, 3, 4]:

$$F_t = -r \cdot \omega^2 \left[(m_p + m_{1b}) \cdot (1+\lambda_b) + (m_{2b} - m_{cb}) \right] = -6250[N] \tag{2}$$

where: $m_{1b} = 0{,}223$[kg] – concentrated mass in the conn-rod small end axis;
$m_{2b} = 0{,}583$[kg] – concentrated mass in the conn-rod big end axis;
$m_{cb} = 0{,}178$[kg] – rod cap mass.

To determine deformations in this case, a separate analysis was performed during which constraints were applied as shown in Figure 4.

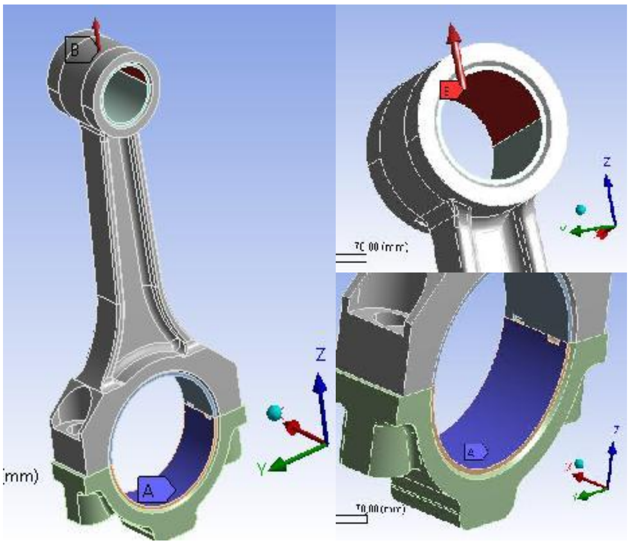


Figure 3: Applying constraints (case of conn-rod small end):
A – fixed suport;
B – tensile force F_t.

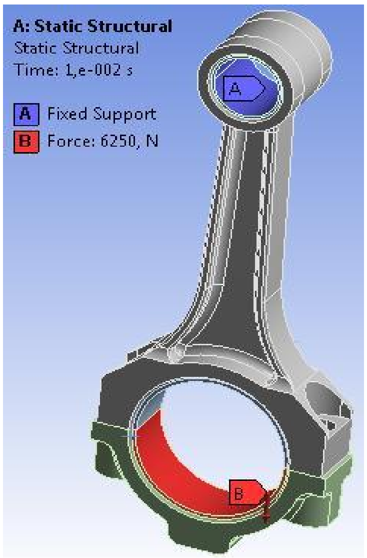
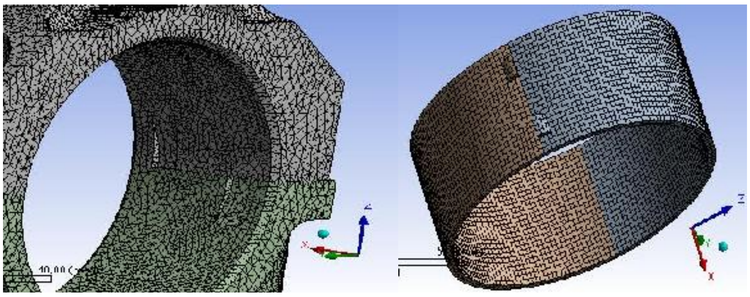


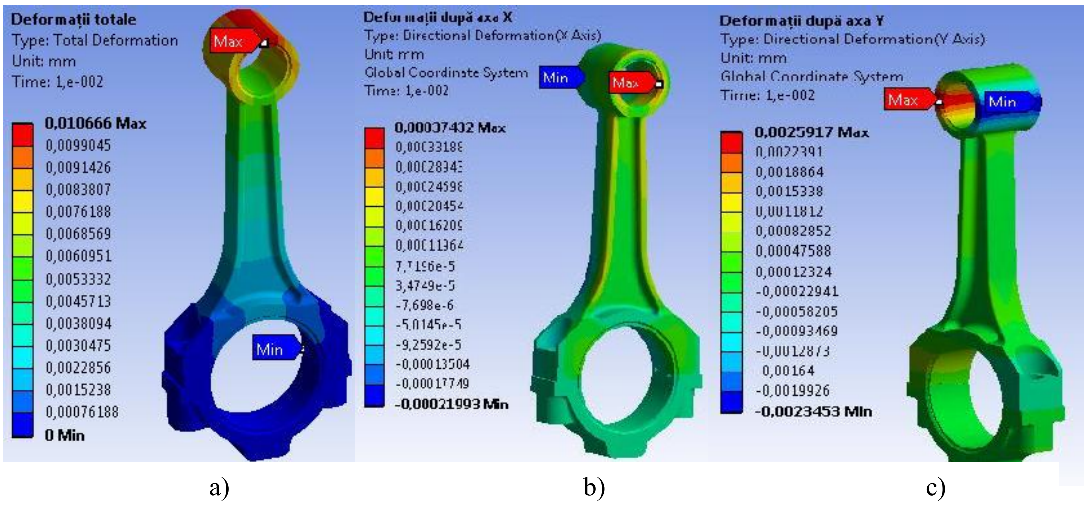
Figure 4: Applying constraints (case of conn-rod big end):
A – fixed suport;
B – tensile force F_t.



a) b)

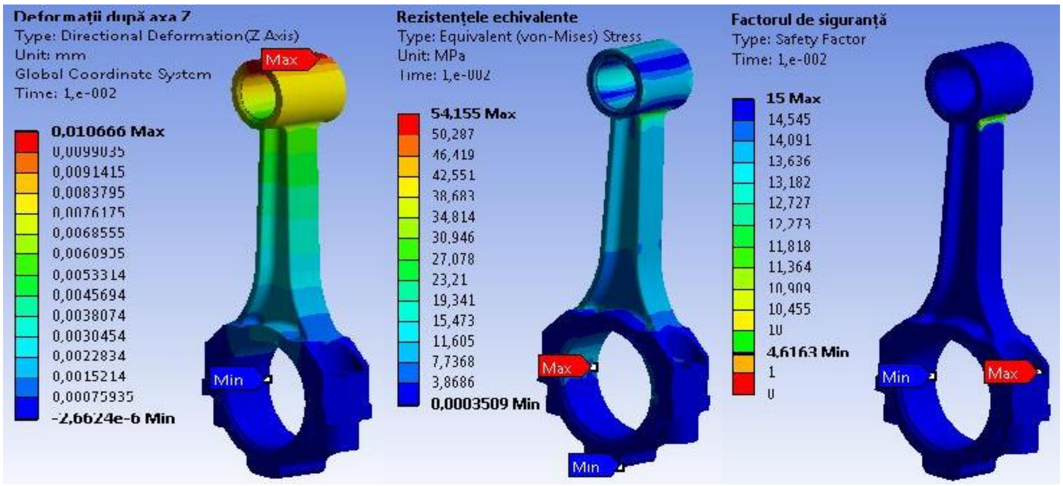
Figure 5: Mesh rod – 3D model: a) conn-rod body and big end; b) bearings liner of conn-rod big end

Meshing 3D model of connecting-rod was made for each component, as follows: a) for conn-rod body and big end were used tetrahedral elements totaling 254 688 nodes (Figure 5-a), b) for bush of conn-rod small end and bearings liner were used hexahedron elements totaling 146 203 nodes (Figure 5-b).
Each component was assigned the corresponding material so [2, 3, 4]: 41MoC11 for rod, CuAl 10Fe3 for bush of conn-rod small end; CuSn12Ni for bearings liner.
In Figures 6, ..., 15 are shown the results of two finite element analysis.



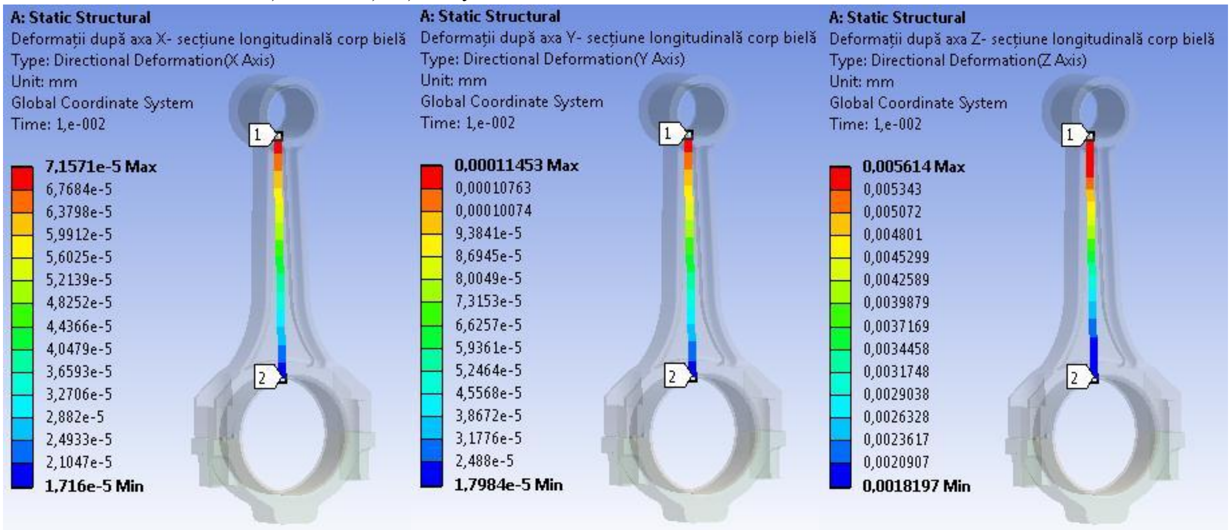
a) b) c)

Figure 6: Case of conn-rod small end: a) total deformation; b), c) directional deformation for X and Y axis



a) b) c)

Figure 7: Case of conn-rod small end: a) directional deformation – *Z* axis; b) equivalent (von-Mises) stress; c), d) safty factor



a) b) c)

Figure 8: Case of conn-rod small end – longitudinal section of rod body: a), b), c) directional deformation for *X*, *Y* and *Z* axis

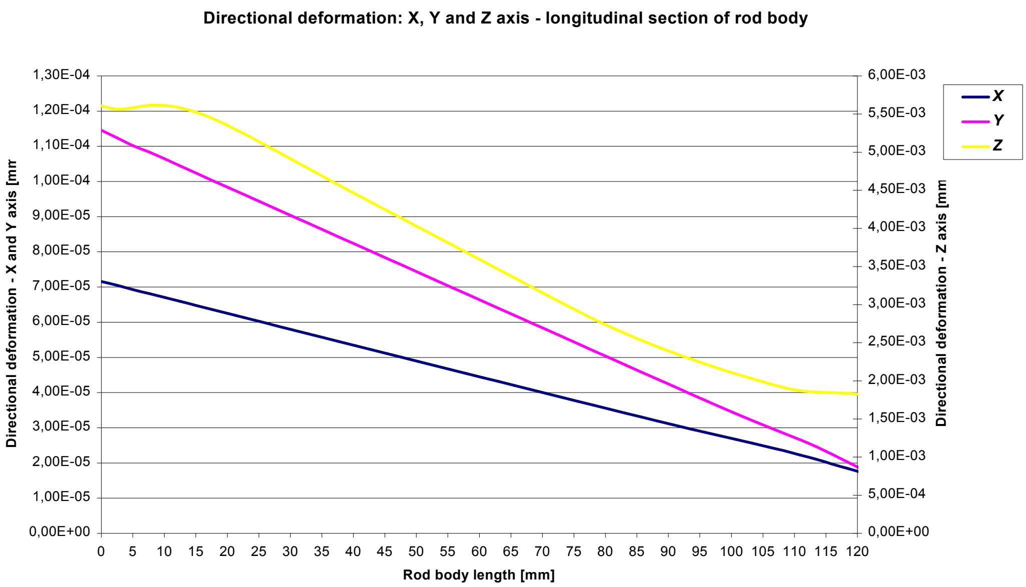
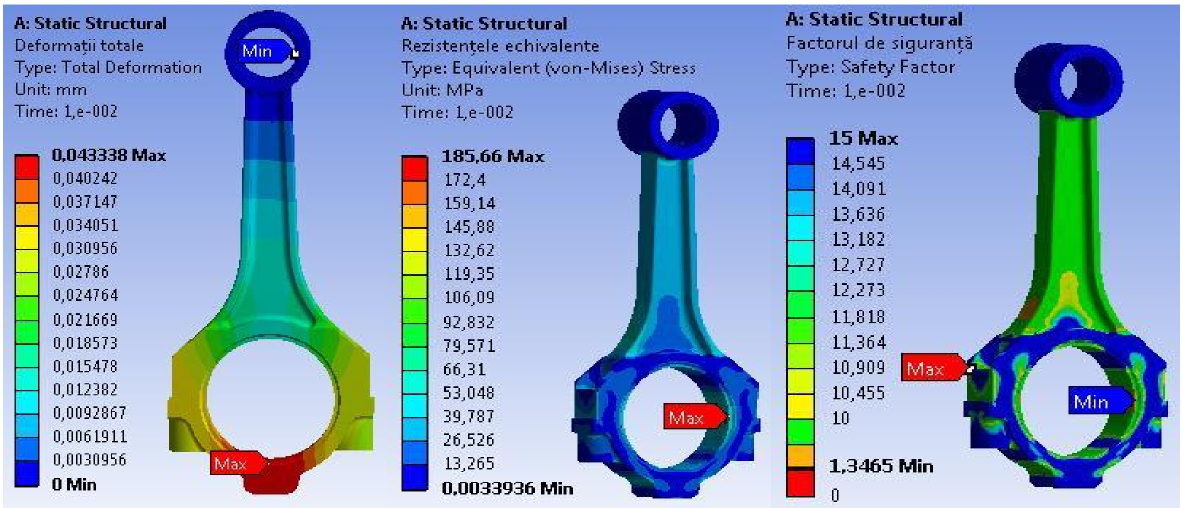
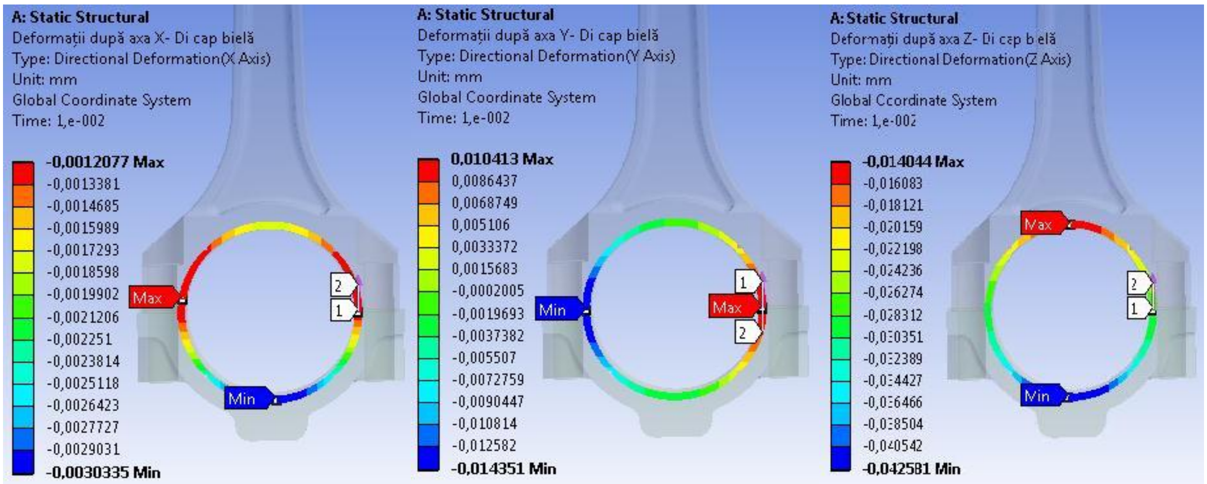


Figure 9: The graph of directional deformations for conn-rod body (longitudinal section)



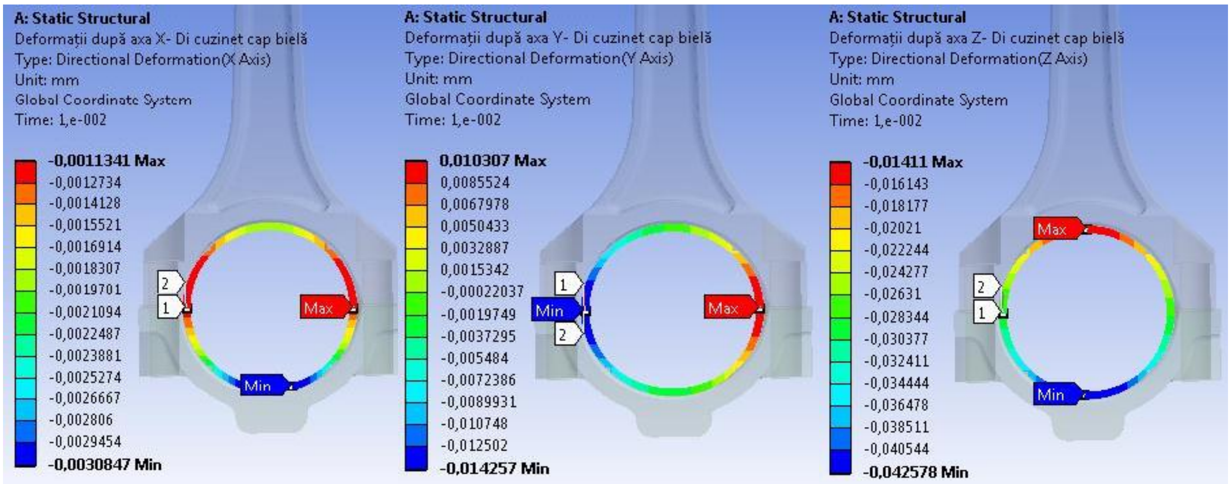
a) b) c)

Figure 10: Case of conn-rod big end: a) total deformation; b) equivalent (von-Mises) stress; c) safety factor



a) b) c)

Figure 11: Case of conn-rod big end: a), b), c) bore diameter of conn-rod big end – directional deformation for *X*, *Y* and *Z* axis



a) b) c)

Figure 12: Case of conn-rod big end: a), b), c) bore diameter of bearings liner – directional deformation for *X*, *Y* and *Z* axis

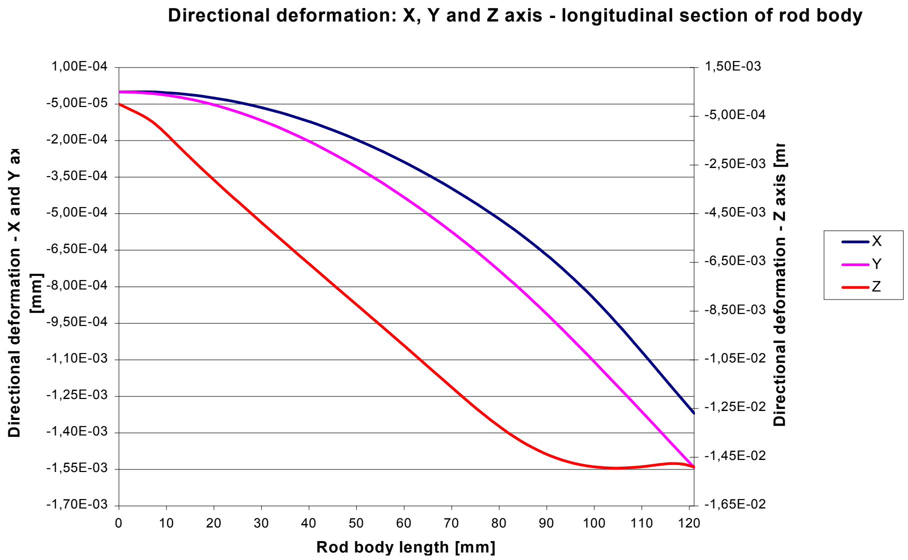


Figure 13: The graph of directional deformations for conn-rod body (longitudinal section)

3. CONCLUSIONS

Maximum deformation is achieved on the conn-rod small end, for Z axis (≈ 0.0106mm). This should not exceed 1/2 of the bearing clearance to prevent pin gripping [3]. For this engine, the floating pin have a clearance of $\Delta = 0{,}0015 \cdot d_{eb} = 0{,}0375$mm, where $d_{eb} = 25$mm represents the pin diameter. In this case, according to the coordinate system used (Figures 2 and 3), deformations of 3D model for Y and Z axis requiring special attention.
Maximum deformations for conn-rod big end (Figures 11 and 12) must not exceed 1/2 of the clearance made on mounting rod with crankpin. For this engine, $d_M = \Delta = 0{,}003 \cdot 0{,}171$mm [3] (where $d_M = 57$mm is crankpin diameter).

ACKNOWLEDGEMENT

This paper is supported by the Sectoral Operational Programme Human Resources Development (SOPHRD), financed from the European Social Fund and by the Romanian Government under the contract number POSDRU/88/1.5/S/59321.

REFERENCES

[1] Ansys Help.
[2] Abăităncei, D., Bobescu, Gh. „*Motoare pentru automobile*”, Editura Didactică şi Pedagocică, Bucureşti, 1975.
[3] Grünwald, B. „*Teoria, calculul şi construcţia motoarelor pentru autovehicule rutiere*”, Editura Didactică şi Pedagocică, Bucureş ti, 1980.
[4] Mărdărescu, R., „*Motoare de automobile şi tractoare: Teoria, calculul şi construcţia*”, Editura Învăţământului, Braş ov, 1959.