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INFLUENCE OF VARIABLE COMPRESSION RATIO REGARDING STRESS STUDY OF INTERNAL COMBUSTION ENGINES COMPONENTS

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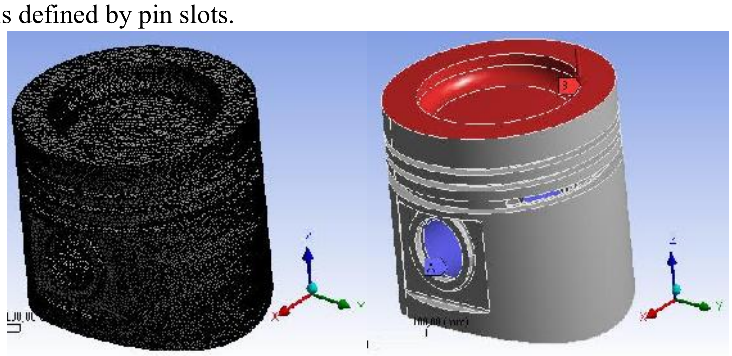
Abstract: *In case of modifying continuous geometric compression ratio while the engine is operating, the maximum variation of pressure in the cylinder and forces seeking crank mechanism is obvious. To determine the mechanism of motor claims in this case will be considered maximum values of forces that occur when the engine is running with a geometric compression ratio max (maximum value of the range set by the control system* ε_v
Keywords: *variable compression ratio, stresses, deformations.*

1. INTRODUCTION

This article focuses on the study of the main components of engine applications with distribution through windows and rotating cylinder jacket (MDR-2), when control mechanism geometric compression ratio acting on the range of values of between 8.5:1 and 10.2: 1. Changing the compression ratio is achieved by moving the cylinder head to the engine block mounted via eccentric mechanism. The study load is realized through the finite element method. Thus, after determining the magnitudes of forces for a successive series positions of motor mechanism (the angle of rotation of the crankshaft), using specialized software Ansys, was analyzed the strain on motor mechanism. In this paper are presented only the results of finite element analysis for piston engine MDR-2 (Heron combustion chamber), when it works with maximum compression ratio - ε_v = 10,2:1.

2. ANALYSIS OF HERON PISTON CHAMBER

If MDR-2 piston after 3D finite element mesh model of tetrahedron type (figure 1-a) totaling 500,971 nodes were applied constraints as can be seen in Figure 1-b. Gas pressure is applied evenly on the surface of the piston and the fixed bracket is defined by pin slots.



a) b)

Figure 1: Discretization (a) and implementations of piston constraints (b) – 3D model: A – fix support fix; B – gas pressure

The pressure applied to the 3D model is maximum (Figure 2) registered when the engine running with geometric compression ratio 10,2:1 (for $\alpha = 375^o$ RAC – $p_g = 6{,}796$MPa).
The difference between the operating temperature and the piston assembly ("cold"), it produces radial and longitudinal expansion, the latter giving form tapered piston. To prevent the cylinder block or piston seizure due to this expansion, it is designed so that the cylinder assembly is done with a game diametrically Δ ', also called "hot game". This game varies along the piston head is higher for low to prevent seizure and jacket to prevent thrashing. If the piston MDR-2, the values of this game are given by the relations: Δc '= 2 · 10-3 · D (for piston head) and Δm' = 0.4 · 10-3 · D (for mantle piston), where D is the bore in [mm] [1, 2]. The difference between the operating temperature of the piston and that of assembly ("cold"), it produces radial and longitudinal expansion, the latter giving to the piston a tapered form. To prevent the block or piston seizure in the cylinder due to this expansion, it is designed so that the cylinder assembly is made with a diametrically clearance Δ ', also called "hot clearance". This game varies along the piston being higher to the head to prevent seizure and lower to the jacket to prevent thrashing. To the MDR-2 piston, the values of this game are given by the relations: $\Delta_c^{`} = 2 \cdot 10^{-3} \cdot D$ (for piston head) and $\Delta_m^{`} = 0{,}4 \cdot 10^{-3} \cdot D$ (for jacket piston), where D is the bore in [mm] [1, 2].
According to [1, 2], at the finite element analysis, the 3D model was divided into three areas on which were applied different values of temperature: a) 523,15K for the piston head, b) 473,15K for the port-rings region, c) 373,15K for jacket piston.
Finite element analysis results which are taken into account and the temperature values above mentioned can be viewed in Figures 3, ..., 10.

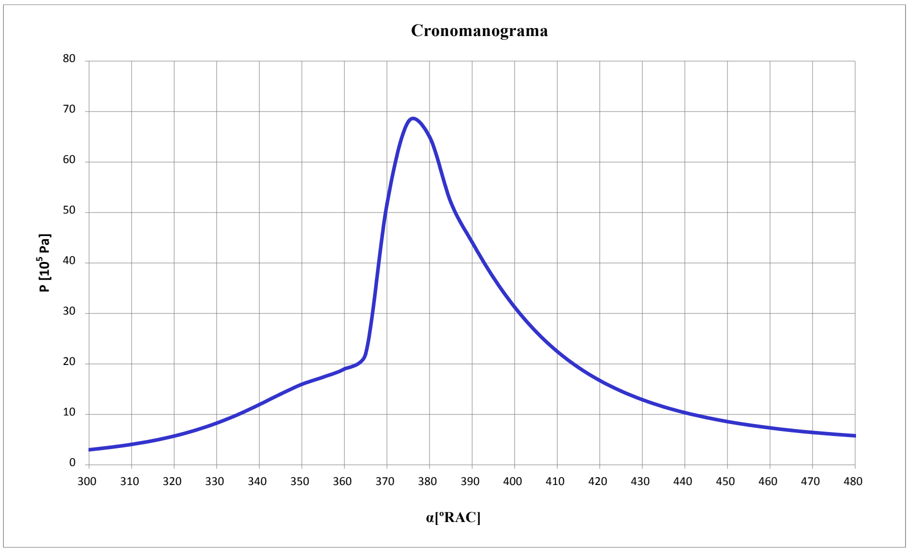
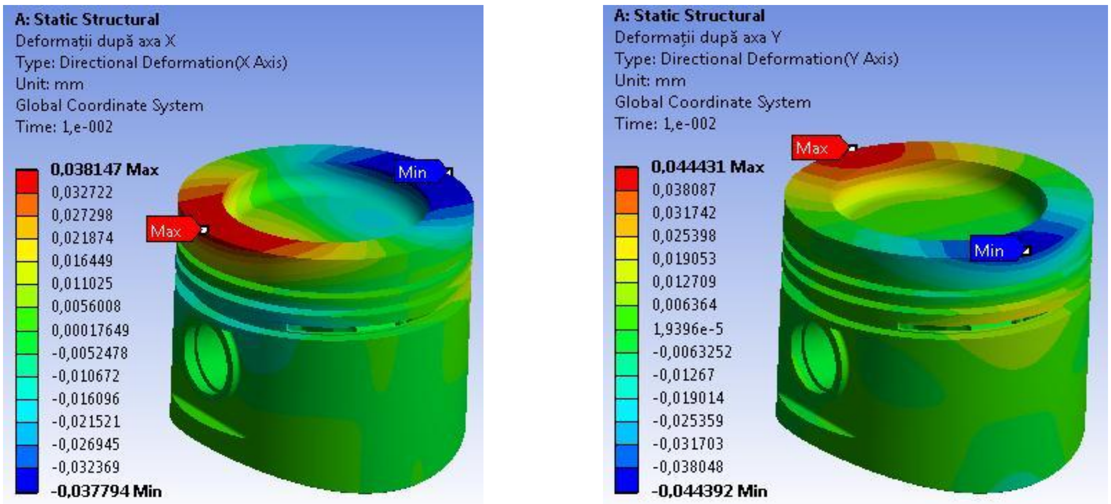
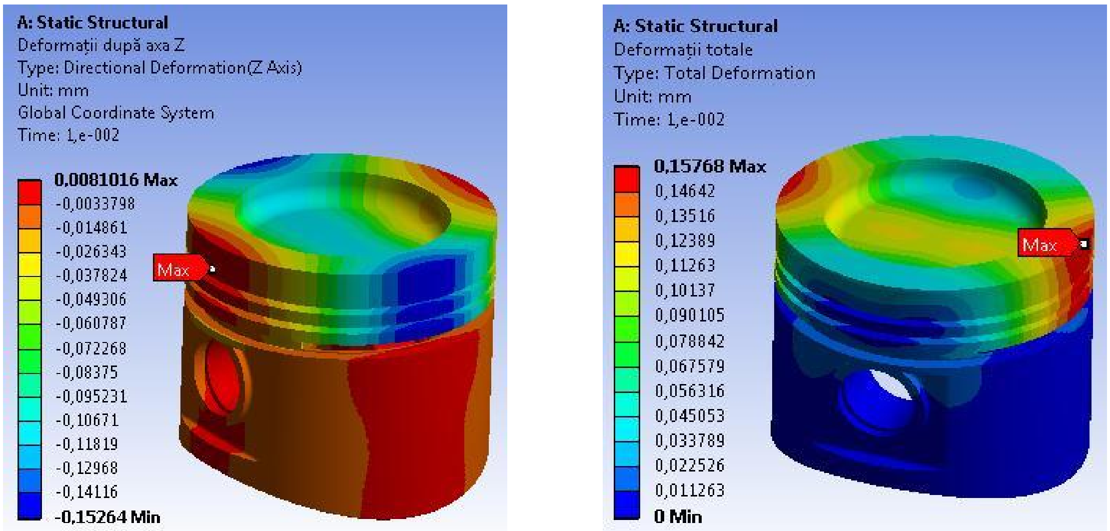


Figure 2: Gas pressure from the cylinder depending of the crankshaft rotation

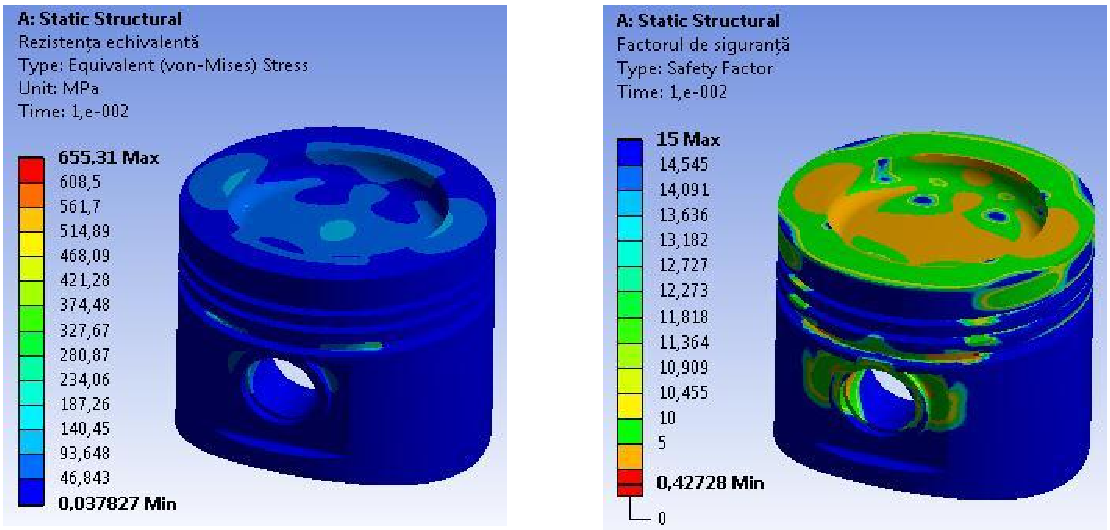


a) b)
Figure 3: Deformation by the *X* (a) and *Y* (b) axes of the piston



a) b)

Figure 4: Deformation by *Z* (a) axis and total deformations (b)

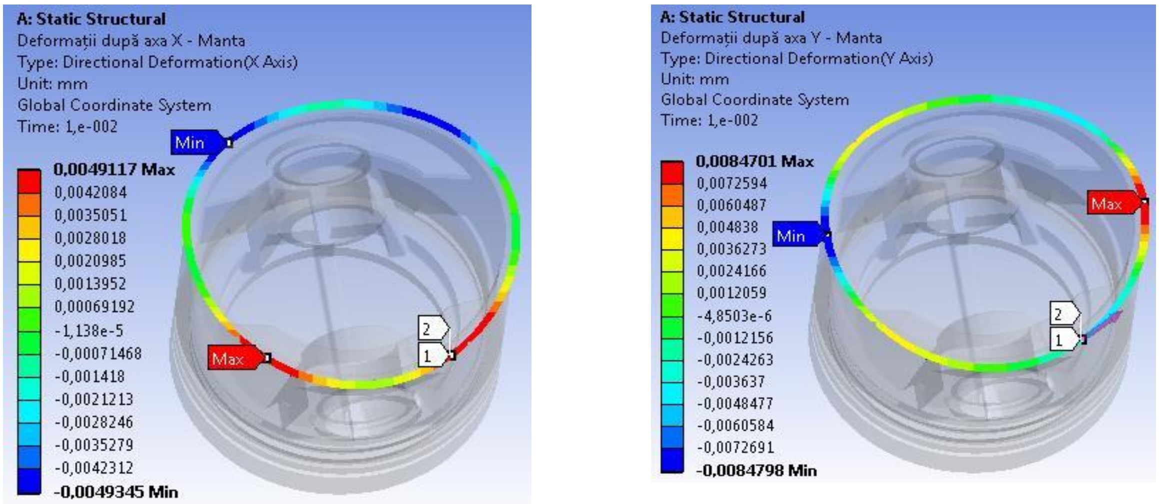


a) b)

Figure 5: Equivalent tensions (a) and safety factor (b)

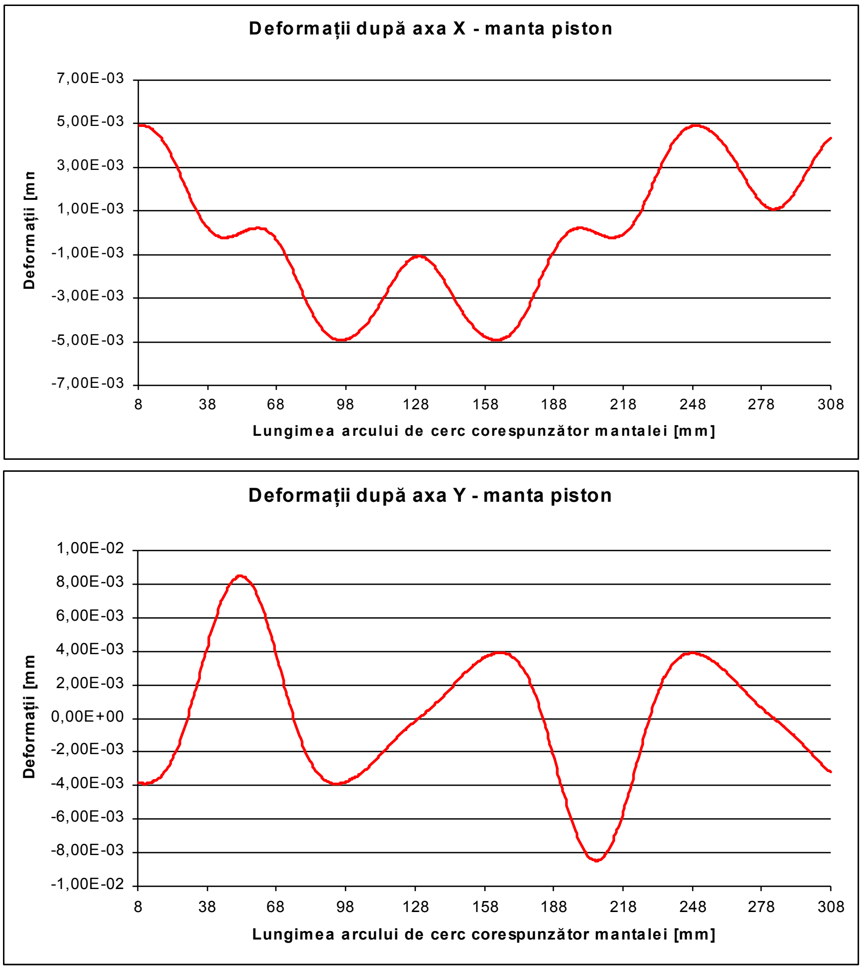
Having regarded to diametrically game imposed on physical model, within the finite element analysis, deformations by the X and Y axes are of high interest. As can be view in Figure 3-b, the deformation of maximum value by the Y axis (≈ 0,044mm) is higher than that recorded by X axis (≈ 0,038mm). Obviously it can be observed that is not is exceeded the value of clearance Δ_c realized to the assembly of the piston with the cylinder (0,097mm radius). Maintaining its low values is essential having regard of noise level during the engine operation. Transverse motion which apply alternative the piston with shock on cylinder and his dumping, produce the cylinder vibration, and characteristic noise called knocking of the piston. Reducing diametrically clearance results the limiting of the noise emitted by the engine and oil consumption, but in this case, should be taken into account and the prevention of seizing.

Regarding the safety factor (Figure 5-b), the minimum value of ≈0,427 is found at the channel lubrication ring. Maximum value of the total deformation of the piston (Figure 4-b) is found in the head of the piston. By the action of gas pressure and implicitly the deformation of the piston, pushing the jacket on the cylinder tends to focus only on the bottom of it. Deformations occurring in the jacket level tend to provide a form of ellipsis, in which case there is a danger of piston block is higher diameter thereof exceeds the cylinder bore. In this case, the jacket deformations (Figures 6 and 7) by the axes X and Y must not exceed diametrically clearance Δ_m (0,0194mm radius) imposed to physical model.



a) b)

Figure 6: Deformation by the *X* (a) and *Y* (b) axes of the jacket piston



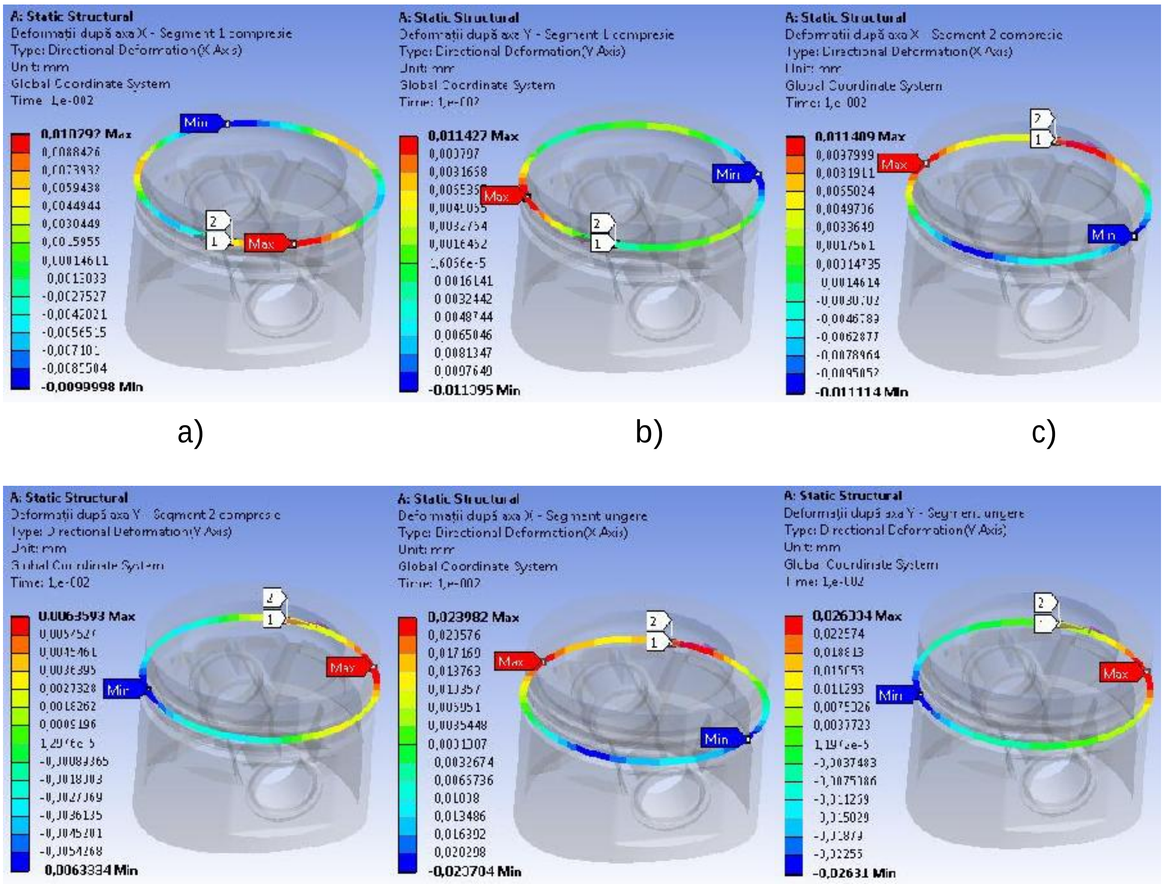
a

b

Figure 7: The jacket deformation graphics by the *X* (a) and *Y* (b) axes

Once with the piston deformation the channels for the rings lean towards the transverse plane, preventing their normal application on the cylinder mirror. Even if the rings are designed and built properly, sealing their task cannot be fully accomplished if the piston operates with larger diametrically clearances.

In the figure 8 can be view the graphics of the rings channels deformations by the axes X and Y, where, also, the maximum values must not exceed the diametrically clearances imposed to the physical model (0,097mm radius).



a) b) c)

d) e) f)

Figure 8: Deformation by the X and Y axes of the rings channel
First ring of compression: a) deformation by the X axis; b) deformation by the Y axis;
Second ring of compression: c) deformation by the X axis; d) deformation by the Y axis;
Lubrification ring: e) deformation by the X axis; f) deformation by the Y axis;

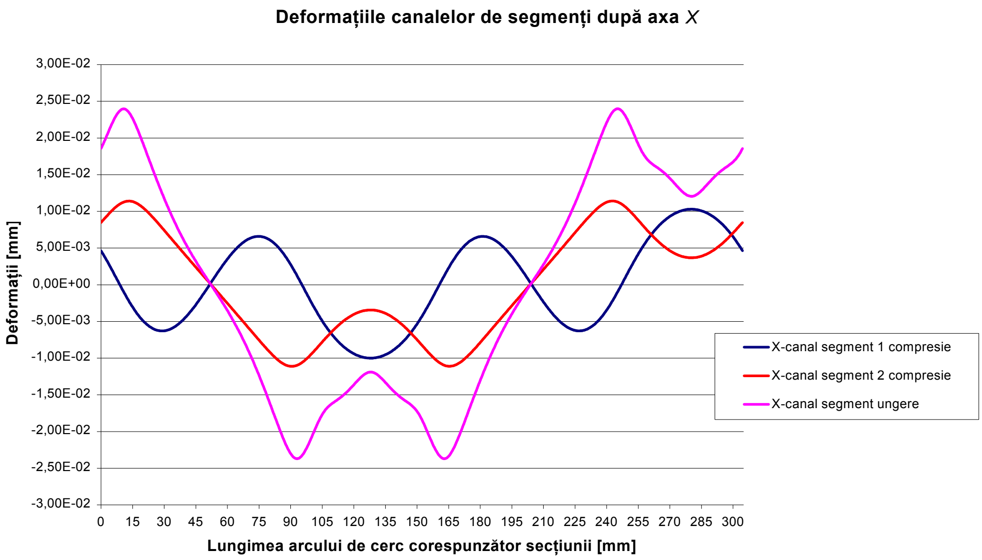


Figure 9: Chanel segments deformations by the axis X

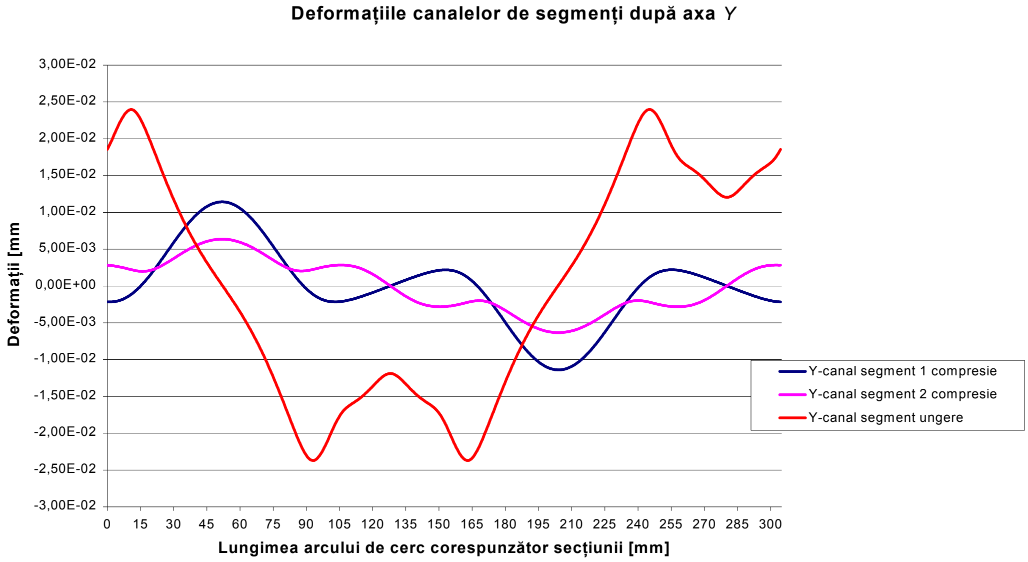


Figure 10: Chanel segments deformations by the axis *Y*

As can view in Figures 8, 9 and 10, the maximum deformations values by X and Y axes at the port region rings (≈0,026 mm by Y axis) is recorded in the channel lubrication ring because the corresponding cross section is reduced due to oil discharge holes scrapings ring. However, the clearence value $\Delta_c^{`}$ is not exceeded.

3. CONCLUSIONS

During engine function, on the piston head acts the force generated by the gas pressure F_g gas, which is passed on to the shoulder bolt through the mantle. Deform pressure Force piston so inclined channel ring of the transverse, preventing uniform application of cylinder mirror rings. If MDR-2 engine, the obtained results with the finite element analysis of the Heron piston combustion chamber, show how it behaves when operating with a compression ratio of 10,2:1. As presented, the maximum deformations and equivalent stress fall within acceptable limits. Therefore is not jendangered the proper functioning of the engine, blocking the piston being avoided.

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