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WEAR RESISTANCE OF PISTON SLEEVE MADE OF LAYERED MATERIAL STRUCTURE: MMC A356R, ANTI-ABRASION LAYER AND FGM INTERFACE

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Abstract: The aim of this paper is the numerical analysis of the one of main part of car engine – piston sleeve. The first example is for piston sleeve made of metal matrix composite (MMC) A356R. The second improved material structure is layered. Both of them are comparison to the classical structure of piston sleeve made of Cr-Ni stainless steel. The layered material structure contains the anti-abrasion layer at the inner surface of piston sleeve, where the contact and friction is highest, FGM (functionally graded material) interface and the layer of virgin material on the outer surface made of A356R. The complex thermo-elastic model with Archard's condition as a wear law is proposed. The piston sleeve is modelling as a thin walled cylindrical axisymmetric shell. The coupled between the formulation of thermo-elasticity of cylindrical axisymmetric shell and the Archard's law with functionally changes of local hardness is proposed.

Key words: Piston Sleeve, Wear, Thermo-Elasticity, Archard's Law, Metal Matrix Composite MMC, Functionally Graded Material FGM

1. INTRODUCTION

The aim of this paper is the numerical analysis of wear of the one of main part of car engine – piston sleeve. The wear process can be described as a removing a thin layer from the surface of the material (Sarkar, 1976). Wear process depends of various types of mechanisms responsible for removal of material from surfaces. At the instant of wear, the rate of volume removed per unit sliding distance must be a function of the volume of material available at the junctions. In general, this definition can be describe as:

$$\frac{\mathrm{d}V}{\mathrm{d}S} = -\beta_1 V, \tag{1}$$

where: term V denotes volume, S is the sliding distance and β_1 is a constant which depends, possibly, on the applied loads. The constant β_1 usually has negative sign describes a situation, where the original volume at the junctions diminishes with sliding distance (Sarkar, 1976).

In alternative model proposed by Archard (1953) two nominally flat surfaces contact each to other at the high asperities which flow plastically, because of the concentrated localised stresses there. As this happens, the compliance between the couple improves, that is the gap between the two surfaces diminishes resulting in further protuberances making contacts elsewhere.

For a large distance *S*, the total volume of wear per unit sliding distance can be represented as:

$$V = \beta \frac{F}{3\sigma_y} S, \tag{2}$$

where: F denotes applied load, β – wear coefficient, σ_y – yield surface limit. Denominator in (2) can be represented by local hardness H of the material, where $H = 3\sigma_y$.

Constitutive relation, contains the thermo-elasticity formulation combined with Archard's law, is described by the decrease of the thickness due to wear at the inner layer of the piston sleeve (Wajand and Wajand, 2005).

2. TEMPERATURE FIELD IN CYLINDRICAL SHELL

The axisymmetric distribution of temperature and its change through the longitudinal direction x is assumed. Moreover, a linear distribution of temperature through the thickness is considered as a loading:

$$T(x,y) = T_x(x) + \frac{\bar{T}(x)}{h} y, \tag{3}$$

where: T_x is a mean value of temperature of the shell wall, $\bar{T}$ stands for temperature gradient inner and outer layer, *x* is the axial coordinate, *h* stands for thickness of cylindrical shell.

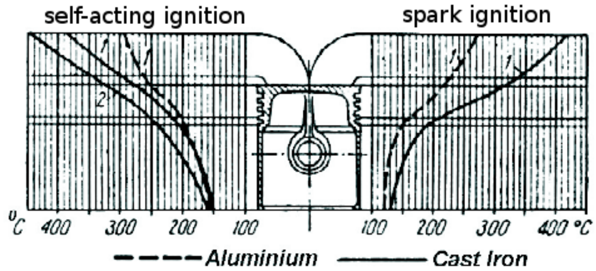

Fig. 1. Temperature distribution of piston: 1-four phase engine (Otto engine), 2-two phase engine (Diesel engine), after Leśniak (1964)

Experiments confirm that almost whole heat flux which enters the upper piston surface goes towards rings and cylindrical part

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of the piston, whereas the lower surface subjects to adiabatic conditions. On the other hand, the temperature distribution at upper piston surface is not uniform (Leśniak, 1964). Above observation is confirmed by Fig. 1, where it is shown the distribution of the temperature in Otto and Diesel engines. In this case, the temperature distribution of piston sleeve may be well approximated by square function as follows:

$$\bar{T}(x) = 150x^2 - 300x + 250, \tag{4}$$

$$T_x(x) = 75x^2 - 150x + 175, \tag{5}$$

It is assumed that the value of temperature is equal to $\mathrm{T_w} = 100\ °\mathrm{C}$ – approximately the temperature of boiling of the cooling liquid in pressure at the inner layer, whereas the value of the temperature at the outer layer is equal to $\mathrm{T_z} = 250\ °\mathrm{C}$.

3. MECHANICAL MODEL

3.1. Bending of shell

The Piston sleeve is a thin structure of cylindrical shape, where the thickness is smaller in comparison with convenient its length. According to that properties, the simplest mechanical model is based on cylindrical axisymmetric thin walled shell. It is assumed:
– Body forces and pressure on circumference surface are constant, hence the assumption of axisymmetric is true. Above variables can change allow the axial direction. All forces are moved to middle layer, where the radius of cylindrical surface is equal *R* (see Fig. 2*)*.
– Radial displacement is very small in comparison to thickness *h*.
– The temperature is changed trough the thickness *h* and axial coordinate *x*.
– Influence of axial force on bending moments are small and it is omitted.
– Young modulus *E*, Poisson ratio υ and coefficient of thermal expansion α are not depend on temperature.

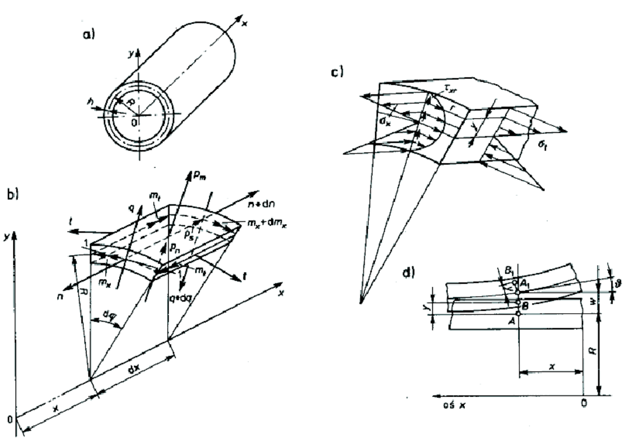

Fig. 2. Bending cylindrical shell: a) geometry, b) internal forces, c) distribution of stresses, d) deformation of middle layer; after Życzkowski (1988)

When the axial forces *n* are omitted, the internal equilibrium equation in displacement form coupled with Eq. (3) is following as:

$$\frac{\mathrm{d}^2}{\mathrm{d}x^2}\left[\mathcal{D}\left(\frac{\mathrm{d}^2 w}{\mathrm{d}x^2} - \alpha(1+\upsilon)\frac{\bar{T}}{h}\right)\right] + \frac{Eh}{R^2}w == p + \frac{Eh\alpha T_x}{R}, \tag{6}$$

where: $w(x)$ – unknown radial displacement function, x – axial coordinate, $\mathcal{D}(x,N) = \frac{Eh^3(x,N)}{12(1-\upsilon^2)}$ – bending stiffness, α – coefficient of thermal expansion, h – thickness of shell, R – radius of shell, E – Young modulus, p – pressure, υ – Poisson's ratio.

The bending stiffness $\mathcal{D}$ cannot be shifted before the differential operator, like in classical form, because the thickness of piston sleeve is not constant. There is a function of axial coordinate x and number of cycles in wear process N.

3.2. Boundary condition

Uniqueness of the solution of the boundary value problem described by equation (6) requires formulation of an appropriate set of boundary conditions. Classical formulation is not favourable because its incompatibility between thermal and mechanical boundary conditions. This effect can be solved by adding two axisymmetric elastic ribs. The first one is located at the upper part of piston sleeve ($x = 0$) and the other one in the below part ($x = l$), both modelling as disks, where l denotes the length of piston sleeve.

Differences of the temperature distribution in ribs are not taken into account because the dimension of ribs is small in comparison to dimension of the structure. A radial displacement of ribs is a function of temperature θ and transverse force q is given by formula:

$$u(\theta) = \alpha\theta r_1 - \frac{q r_1}{Eh}\frac{(1-\upsilon)\bar{r}^2 + (1+\upsilon)}{1-\bar{r}^2} = f_b(\theta, q, \mathrm{r})\ , \tag{7}$$

where: $\bar{r} = r_1/r_2$, $r_1 = R$ – outer radius of piston sleeve, $r_2 = R + l_1$, l_1 – height of rib.

The boundary conditions are as follows:

$$\begin{aligned} &w(0) = f_b(\theta, q, \mathrm{r} = r_1), \\ &w(l) = f_b(\theta, q, r = r_1), \\ &m_x(0) = 0, \\ &w'(0) = 0, \end{aligned} \tag{8}$$

where: the $\mathrm{m_x}$ is a radial moment and $\mathrm{w'}$ is an angle of deflection.

4. FORMULATION OF WEAR FOR PISTON SLEEVE

The inner surface of piston sleeve is degraded by rings and the thickness of piston sleeve is decreasing. Some experiments lead to special kind of wear in this type of structure – wedge shape wear (see Fig. 3).

All facts above considered the formulation of the thickness function can be described as follows:

$$\begin{cases} h(x,N) = h_0 & \text{for } x < x_0 \\ h(x,N) = h_0 + \delta h(x,N) & \text{for } x \ge x_0, \end{cases} \tag{9}$$

where: $N-$ number of cycles, $\delta h(x, N)$ is an infinitesimal change of the thickness at one cycle depends on variable x, x_0 is the point where the wedge's shape wear starts. Application of Eq. (1) in the problem of thin shell, where the sliding distance is collinear to coordinate x and total volume is reduced to the problem of change of thickness, previous equation can be rewritten as follows:

$$\frac{\mathrm{d}\delta h}{\mathrm{d}x} = -a\,\delta h, \tag{10}$$

where: constant a stands for a certain material parameter [1/m] having essential role in modelling of wedge's shape wear, see Eqs. (10)-(15). Other words, Eq. (10) can be read as: the rate of thickness removed per unit sliding distance is a function of the thickness of material available at the junction, see Sarkar (1976). After solve very simple differential equation (10), we can introduced:

$$\delta h(x, N) = C(N)e^{-ax}, \tag{11}$$

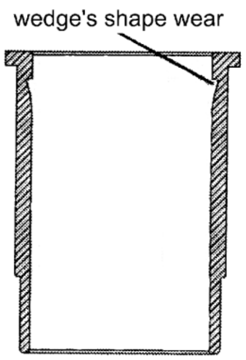

Fig. 3. Wedge's shape wear of piston sleeve

On the other hand the total volume of removed material can be described as:

$$\Delta V = \int_{x_0}^{x_1} 2\pi R \delta h(x, N)\,\mathrm{d}x, \tag{12}$$

where: the $x_1 - x_0$ is a sliding distance in one cycle. Comparison Archard's law and result of integration from Eq. (12), we can describe the following equation:

$$\beta \frac{F}{H} S = 2\pi R\, C(N) \frac{e^{-ax_0} - e^{-ax_1}}{a}. \tag{13}$$

Next it is assumed $\mathrm{e}^{-\mathrm{a}x_1} \approx 0$, $\mathrm{e}^{-\mathrm{a}x_0} \approx 1$ and summarise sliding distance $\mathrm{S} = \mathrm{N}(\mathrm{x}_1 - \mathrm{x}_0)$, we can easily find that:

$$C(N) = -\frac{a}{2\pi R}\,\beta \frac{F}{H}\,N(x_1 - x_0). \tag{14}$$

Substituting (14) to (11) and after that introduced both of them into equation (9) we have finally relation describing change of thickness in the following form:

$$\begin{cases} h(x, N) = h_0 & \text{for } x < x_0 \\ h(x, N) = h_0 - \dfrac{a}{2\pi R}\,\beta \dfrac{FS}{H}\,e^{-ax} & \text{for } x \geq x_0. \end{cases} \tag{15}$$

The constant wear parameter $\beta = 2 \cdot 10^{-6}$ is experimentally determined and substituting to all numerical examples after Natarajan et al. (2006). The load F is calculated as follow:

$$F = 2\pi R h_r p_g\,, \tag{16}$$

where: $\mathrm{p_g}$ denotes the constant gas pressure in the cylinder in the one cycle of sliding and $\mathrm{h_r}$ is a medium height of the cylinder rings (Leśniak, 1964). Obviously the original problem is not axisymmetric since wedge's shape wear occurs in the plane perpendicular to the axis of piston pin. However, for simplicity the axially symmetry of the problem is assumed.

5. RESULTS

Numerical integration of the problem takes advantage of step-by-step procedure and the shooting method presented in Press et al., 1983 and Skrzypek et al., 2008.

a) b)

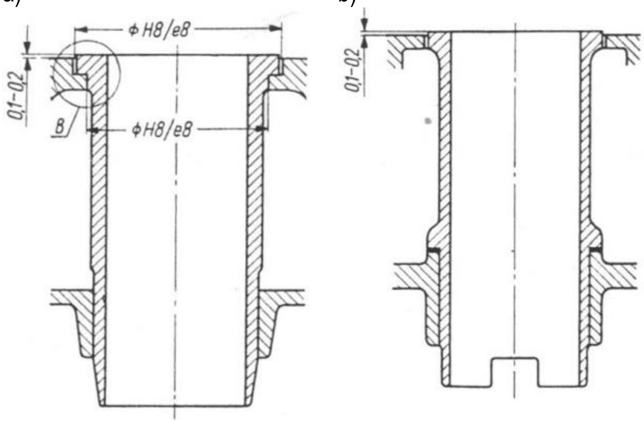

Fig. 4. Wet piston sleeves: a) piston sleeve with upper flange, b) piston sleeve with bottom flange (Wajand and Wajand, 2005)

From the technological point of view there are two types of wet piston sleeve. One of them has got flange on the upper surface (see Fig. 4a). The bottom part of the piston can extend after heating up. In another type of piston a flange is in the bottom part (see Fig. 4b), where the flange rest on ring-shaped sprue (Wajand and Wajand, 2005). Aforementioned it is assumed mechanical model presented with all geometrical parameters in Fig. 5 and it is used in all numerical examples in this paper.

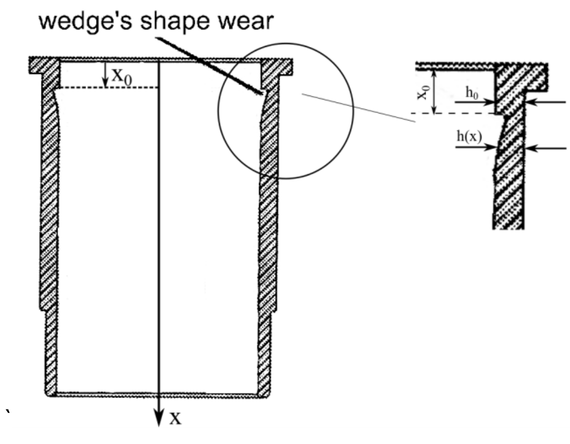

Fig. 5. Mechanical model and geometrical data of piston sleeve.

First numerical example is the reference problem necessary to calibrate material data. The sampling piston sleeve is made of Cr-Ni steel and detailed material data is shown in Tab. 2 and another parameter of analysis is presented in Tab. 1.

Tab. 1. Geometrical and material parameters

Wear coefficient [3] β [-]	Gas pressure in cylinder [5] p_g [MPa]	Medium height of the cylinder rings [5] h_r [mm]	Constant geometric parameter a [1/mm]
$2 \cdot 10^{-6}$	5.0	3.0	$5 \cdot 10^{-2}$
Outer radius shell R [mm]	Length of shell l [mm]	thickness of rib h_b [mm]	Height of rib l_1 [mm]
90.0	140.0	5.0	5.0
Initial thickness of shell h_0 [mm]			
5.0			

Tab. 2. Material data of Cr-Ni steel; after http://www.matweb.com

Young modulus E [GPa]	Poisson's ratio υ [-]	Thermal expansion coefficient α [1/K]	Hardness H [MPa]
170.0	0.33	18.5 x 10^{-4}	66.9

From the experimental point of view, maximum wear (drop of the thickness) is 0.1 mm. If the length of the wedge is equal as below, the piston sleeve will absolutely be unserviceable. Fig. 6 clearly shows wedge-shaped wear, coordinate is equal x/l, where *l* is the length of piston sleeve. It is where the dimensionless worth to notice that the number of cycles leading to destructive wear is equal $N = 2.2 \cdot 10^7$ in the first, reference example.

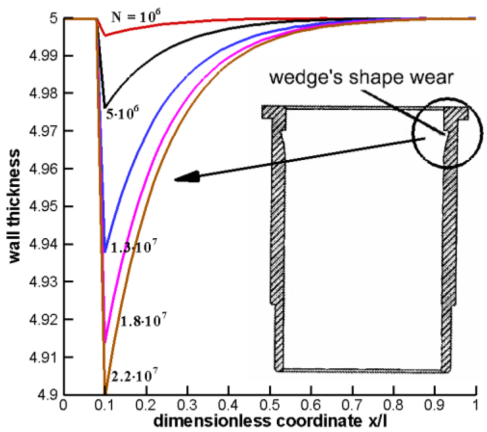

Fig. 6. Distribution of thickness function for piston sleeve made of Cr-Ni steel

5.1. Piston sleeve made of metal matrix composite MMC A356R

In the next numerical example piston sleeve is made of special kind of aluminium-based composite. The chemical formula of this material, called A356R, is $Al\text{-}Si_7Mg_{0.3}$ + 6% TiB_2. Material data of this material is shown in Tab. 3.

Tab. 3. Material data of MMC A356R; after Egizabal [2].

Young modulus E [GPa]	Poisson's ratio υ [-]	Thermal expansion coefficient α [1/K]	Hardness H [MPa]
79.0	0.33	22.4 x 10^{-6}	338.0

Thickness drop is presented in Fig. 7. Character of the plot is the same like in previous example (see Fig. 6). It is well visible that similar wedge-shaped wear appears, however the number of cycles leading to destruction increases ($N = 1.06 \cdot 10^8$). This is the advantage of application of A356R composite.

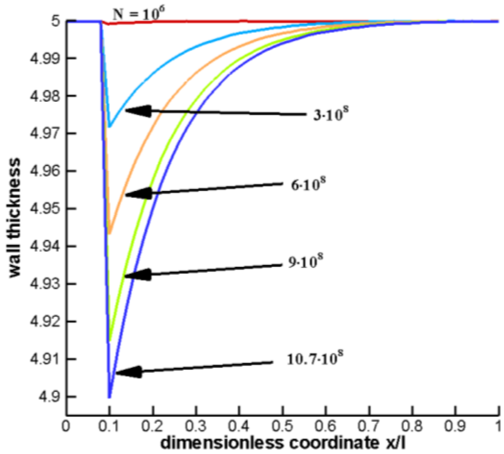

Fig. 7. Distribution of thickness function for piston sleeve made of aluminium composite A356R

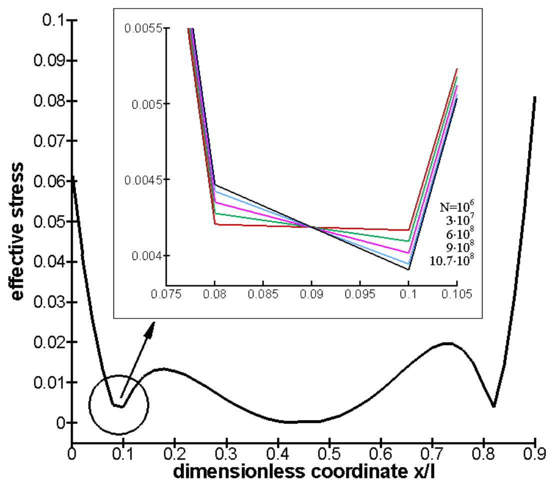

Fig. 8. Distribution of von Mises stress using the shell theory in piston sleeve made of A356R.

Next plot shows von Misses effective stress (see Fig. 8). Detailed analysis reveals existence of smoothness of effective stress near the point where wedge appears. It is a very dangerous effect, because it can lead to fracture in micro level. This is a consequence of stress concentrate at this point. Solution for this problem is applying the anti-abrasion thin layer. However combination of two material of huge differences of material hardness can be provided to huge concentration of stress at this point. Remedy for this problem is introduced the functionally graded materials (FGM) as an interface between the anti-abrasion layer with huge local hardness and virgin material.

5.2. Piston sleeve made of A356R composite material with abrasion resistant thin layer

There was presented the comparison between pistons sleeve made of classical Ni-Cr steel and composite material A356R in the previous sections. Material A356R is very sophisticated, however its wear properties is not perfect. The solution for this problem is introduced the abrasion resistant thin layer.

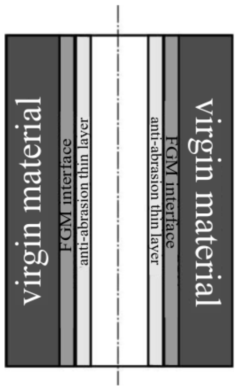

Fig. 9. Material model of piston sleeve with anti-abrasion thin layer

On the other hands connection of two materials with huge difference of properties can lead to micro fracture on interface. In this case application of middle layer from FGM (functionally graded material) is necessary. FGM is special kind of structural concept with spatial varying thermomechanical properties. Varying volume fraction of both constituents ceramic and metal determines the local macroscopic (effective) properties of a composites at the given point of structure. This structures was developed by Japanese engineering and scientists in 1980 (Suresh and Mortensen, 1998).

In the presented example it is introduced layer material of piston sleeve. In the inner side, where the rings contact with piston sleeve, it is applied thin layer of ductile iron as an anti-abrasion layer, because the hardness of ductile iron is huge (see the material data presented in Tab. 4). On the opposite side it is introduced composite material A356R (virgin material) and between them, it is applied the FGM interface with functionally changes of local hardness (see Fig. 9).

Tab. 4. Material data of ductile iron; after http://www.matweb.com

Young modulus E [GPa]	Poisson's ratio υ [-]	Thermal expansion coefficient α [1/K]	Hardness H [MPa]
172.0	0.29	12.0×10^{-6}	583.8

In this example it is applied that the failure wedge is 0.1 mm. This is the same value like in previous examples. From engineering practise point of view, the maximum wear of piston sleeve should not be greater than thickness of thin anti-abrasion layer. However, from the cognitive point of view, the value of thickness of anti-abrasion thin layer is applied as $h_{wear} = 0.09$ mm, which is smaller than maximum wear. The thickness of FGM layer is applied as $h_{FGM} = 0.03$ mm.

The main parameter, which is determined the velocity of wear process, is local hardness, hence this material property is introduced in form presented in Eq. (17) in the FGM interface layer.

$$H(y) = H_c - \tilde{H}[1 + \mathrm{tgh}(y - y_0)] \tag{17}$$

where it is substituted:

$$\tilde{H} = 0.5(H_c - H_{VM})$$
$$y_0 = h_{wear} + \frac{1}{2}h_{FGM} \tag{18}$$

Symbol H_c means the local hardness of anti-abrasion layer and H_{VM} is a local hardness of virgin material. Distribution of hardness function through the FGM interface is presented in Fig. 10.

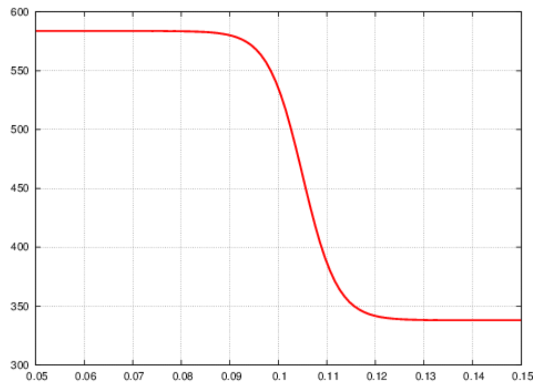

Fig. 10. Distribution of local hardness function $\boldsymbol{H(y)}$

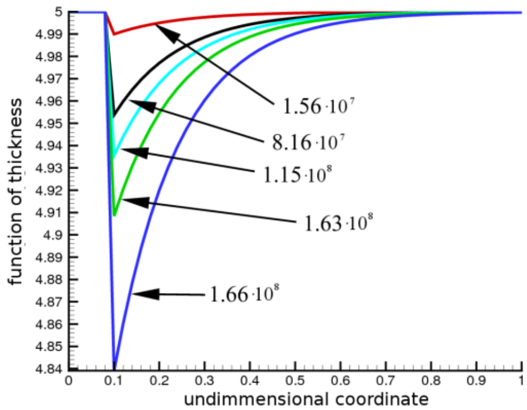

Fig. 11. Thickness function of piston sleeve with anti-abrasion layer and FGM interface

The first layer from $y = 0.05$ to $y = 0.09$ is made of ductile iron with huge value of local hardness. The next layer of thickness 0.03 is FGM interface, where the local hardness varying spatial to the value of local hardness of virgin material – A356R.

Applying an anti-abrasion layer in the piston sleeve leads to more effective resistant of wear (see Fig. 11). The number of cycles, after that the wear reaches the critical value

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is $N = 1.66 \cdot 10^8$. It is worth to noticed that the velocity of wear process progresses rapidly, when the value of wear is greater than the thickness of anti-abrasion layer.

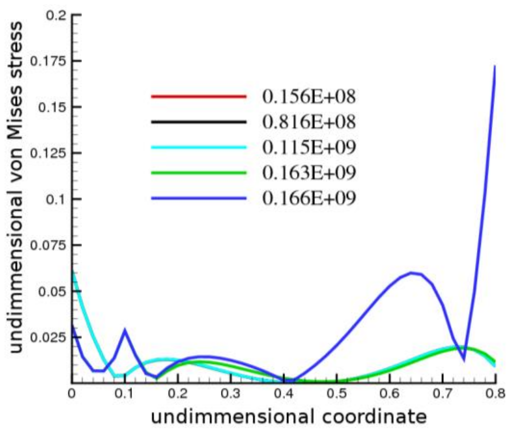

Fig. 12. Von Mises stress of piston sleeve with anti-abrasion layer and FGM interface

The next figure is presented the distribution of equivalent H-M-H stress. When the degradation of surface is grater then anti-abrasion layer, the distribution of stresses are changed. It is a consequence of modification of material. It is going to change from the ductile iron to the composite material A356R, hence the material coefficients change too.

6. CONCLUSIONS

– Numerical model presented in this paper with complex thermo-elastic phenomena and linear wear law (Archard's law) is proposed to describe wear process in piston sleeve.
– Wear resistance of piston sleeve made of composite material A356R is higher in comparison to the wear of the sampling piston sleeve made of Cr-Ni steel.
– The layered structure of material with FGM interface between the anti-abrasion layer and virgin composite material A356R improves the strength of piston sleeve, because the graded interface eliminates the concentration of stresses.

List of symbols: *V* – total volume of material, *S* – sliding distance, *n* – material constant in wear law, *F* – applied load, *β* – wear coefficient, σ_y – yield surface limit, *H* – local hardness of the material, *T* = *T*(*x*, *y*) – function of temperature, *x* – axial coordinate, *y* – radial coordinate, $T_x(x)$ – mean value of temperature of the shell wall, $\bar{T}(x)$ – temperature gradient inner and outer layer, h_0 – initial thickness of shell, δh - infinitesimal change of thickness, $h = h(x, N)$ – function of thickness of cylindrical shell, T_w – approximately the temperature of boiling of the cooling liquid, T_z - temperature at the outer layer of shell, *R* – radius of shell, E – Young modulus, ν – Poisson's ratio, α – coefficient of thermal expansion, $w(x)$ – radial displacement function, *D* = *D*(*x*, *N*) – bending stiffness, *p* – pressure, *N* – number of cycles in wear process, l- length of piston sleeve, $\theta = \theta(r)$ – function of temperature in ribs, *q* – transverse force in ribs, l_1 – height of rib, x_0 – coordinate of points where the wear process is start, a – material constant, h_r – medium height of cylinder rings, p_g – constant gas pressure in the cylinder in the one cycle of sliding, H_c – local hardness of anti-abrasion layer, H_{VM} – local hardness of virgin material, h_{wear} – thickness of anti-abrasion layer, h_{FGM} – thickness of FGM layer.

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