Ways to improve conditions in mineral oil lubrication of the internal combustion engines

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Abstract The paper analyses some phenomena affecting the cam-follower contact conditions, highlights some detrimental influences over the lubrication regime in this contact. The analysis of the theoretical model revealed the significant influence of thermal and starvation effects on the reduction of film thickness. To avoid these negative phenomena and in order to improve the lubrication regime we propose an original technical solution by using additional oil feed with oil jets, directed over the contact area.

Keywords: mineral oils, lubrication regime, cam-follower contact, wear, internal combustion engines

1. Introduction

Current requirements for performance and reliability of automotive engines led to the need for complex analysis of all operating parameters. Engines evolution revealed that one of the sensitive issues for achieving required performance is the gas distribution mechanism and, in particular, the cam-follower mechanism.

A first step towards this direction was the highlighting of all the factors that influence the cam-follower contact conditions. In the meantime the paper aims to render obvious solutions for wear reduction of this couple.

In order to achieve the above mentioned objectives, the authors developed, a complex physical-mathematical model with emphasis on the most important factors affecting the contact conditions. Simulation of the characteristic phenomena, in order to define a procedure for improving the contact conditions was also considered.

2. General physical-mathematical model

Due to the complexity of the processes involved, the **cam-follower contact state concept** was developed in order to evaluate the distribution system of the internal combustion engines.

This concept takes into account the multitude of processes occurring inside the cam-follower couple, emphasizing on their complexity and interdependence, and thus facilitating concomitant analysis and improvement of the contact conditions. Also, this concept provides a way to develop an original physical-mathematical model, useful to study specific phenomena. In order to define the previously mentioned concept the following parameters were taken into account:

- the geometrical parameter of the cam-follower contact state;
- the principal stresses for the contact state;
- the thermal stresses for the cam-follower contact state;
- the lubrication state of the cam-follower couple.

Starting from considerations of Dowson and Toyoda [1] a model with extended degree of generality was developed by adopting more realistic conditions related to the lubricant: a non-isothermal flow, development of the starvation phenomenon of a high elasticity modulus for the bodies in contact. So, we established the general equations defining the physical and mathematical model proposed by the authors for the cam-followers contact in this matrix form:

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$$Ax^2 + By^2 = D$$
$$D = z_1 + z_2 = const.$$
$$A = \frac{1}{4}\left[(c_{11}+c_{12})+(c_{21}+c_{22})+\sqrt{\begin{array}{l}(c_{11}-c_{12})^2+(c_{21}-c_{22})^2+...\\+2(c_{11}-c_{12})(c_{21}-c_{22})\cos 2\omega\end{array}}\right]$$
$$B = \frac{1}{4}\left[(c_{11}+c_{12})+(c_{21}+c_{22})-\sqrt{\begin{array}{l}(c_{11}-c_{12})^2+(c_{21}-c_{22})^2+...\\+2(c_{11}-c_{12})(c_{21}-c_{22})\cos 2\omega\end{array}}\right]$$
$$C = \begin{pmatrix} c_{11} & 0 & c_{12} \\ c_{21} & 0 & c_{22} \end{pmatrix}$$
$$\omega$$

$$\sigma_x = -p_0 \cdot 2\mu \frac{z}{b}\left[\sqrt{\frac{b^2+\lambda}{\lambda}} - 1\right]$$
$$\sigma_y = -p_0 \cdot \frac{z}{b}\left[\sqrt{\frac{b^2+\lambda}{\lambda}}\left(2 - \frac{b^2z^2}{\lambda^2 + b^2z^2}\right) - 2\right]$$
$$\sigma_z = -p_0 \cdot \frac{bz^3}{\lambda^2+b^2z^2}\sqrt{\frac{b^2+\lambda}{\lambda}}$$
$$\tau_{yz} = \tau_{zy} = -p_0 \cdot \frac{byz^2}{\lambda^2+b^2z^2}\sqrt{\frac{\lambda}{b^2+\lambda}}$$
$$\tau_{xz} = \tau_{zx} = 0$$
$$\tau_{xy} = \tau_{yx} = 0$$
(1)

$$\bar{\varepsilon}_{xx} = \frac{\partial^2\Phi}{\partial x^2}$$
$$\bar{\varepsilon}_{yy} = \frac{\partial^2\Phi}{\partial y^2}$$
$$\bar{\varepsilon}_{xy} = \frac{\partial^2\Phi}{\partial x \partial y}$$
$$\bar{\varepsilon}_{zz} = \bar{\varepsilon}_{yz} = \bar{\varepsilon}_{zx} = 0$$
$$\bar{\sigma}_{xx} = -2G\frac{\partial^2\Phi}{\partial y^2}$$
$$\bar{\sigma}_{yy} = -2G\frac{\partial^2\Phi}{\partial x^2}$$
$$\bar{\sigma}_{zz} = -2G\nabla^2\Phi = -2G\frac{1+\mu}{1-\mu}\alpha T$$
$$\bar{\sigma}_{xy} = 2G\frac{\partial^2\Phi}{\partial x\partial y}$$
$$\bar{\sigma}_{yz} = \bar{\sigma}_{zx} = 0$$

$$h_{\min} = a \cdot R_x \cdot U_e^{\,b} \cdot G^c \cdot W_e^{\,d}$$
$$h_{cen} = a' \cdot R_x \cdot U_e^{\,b'} \cdot G^{c'} \cdot W_e^{\,d'}$$
$$\lambda = \frac{h_{\min}}{1.15\sqrt{R_{a_1}^2 + R_{a_2}^2}}$$
$$Q_T = \frac{1 - 13.2\left(\frac{p_0}{E_{ef}}\right)\cdot Bk^{0.42}}{1 + 0.213\left(1 + 2.323\cdot C_i^{0.83}\right)Bk^{0.64}}$$
$$Q_{St} = 1 - \frac{1}{\exp\left\{\left[\exp\left(\frac{0.78\ln\psi_{1L}}{1+10^{-3}L_t}\right)\right]\ln\left(4.6 + 1.15\cdot L_t^{\,0.6}\right)\right\}}$$
$$g_e = W^e \cdot U_e^{\,f}$$
$$g_v = G \cdot W^{e'} \cdot U_e^{\,f'}$$

where:

x, y	– xOy coordinates;
D	– distance between the two points on the Oz axis;
z_1, z_2	– distance from a point on body 1, 2 (cam, follower) to the common plane xOy;
A, B, C	– parameters of the contact ellipse;
ω	– angle between the axis of the initial coordinate systems Ox1 and Ox2;
a, b	– the semi-axes of the pressure ellipsoid;
$\bar{\varepsilon}$, $\bar{\sigma}$	– the strains and stresses given by the thermo elasticity equation;
h_{min}, h_{cen}	– minimum and central film thickness;
λ	– specific film thickness;
Q_T	– thermal correction coefficient;
Q_{St}	– coefficient for the reduction of the film thickness due to starvation;
g_e, g_v	– elasticity and viscosity parameter;
R_x	– effective radius on Ox axis;
U_e	– Johnson model speed parameter;
G	– material parameter
W_e	– loading parameter.

3. Customizing of the physical-mathematical model for a linear cam-follower contact

This physical and mathematical model, complex by taking into account a wide variety of phenomena, has a high degree of generality and may thus cover a wide range of situations encountered in cam-followers contact. In the analyzed case, based on a real physical model, the geometric parameters and stresses are known. In these conditions, by customization, the proposed model is reduced to equations describing the lubrication conditions of the cam-follower contact.

Thus, the equations defining the minimum and central film thickness of oil are:

$$h_{\min} = 2.65 \cdot R_x \cdot U_e^{0.7} \cdot G^{0.54} \cdot W_e^{-0.13} \qquad (2)$$

$$h_{cen} = 3.06 \cdot R_x \cdot U_e^{0.69} \cdot G^{0.56} \cdot W_e^{-0.10} \qquad (3)$$

The thermal correction, quantified by subunitary coefficient, Q_T, will be:

$$h_{\min T} = Q_T \cdot h_{\min} \,. \qquad (4)$$

Evaluation of the undesirable effects of insufficient lubricant quantity at the entrance of the couple is made by starvation factor, Q_{st}, subunitary also:

$$Q_{St} = \frac{h_{\min,\ starvation}}{h_{\min,\ abundant\quad flow}} \qquad (5)$$

An advanced approach for the linear cam-follower contact problems, type cam-roller followers or cam-flat followers, assumed that, in operation and in the presence of lubricant, the coupling cam-followers actually develops four lubrication regimes: *Isoviscous Rigid* - ***IVR***, *Piezoviscous Rigid* - ***PVR***, *Piezoviscous Elastic* - ***PVE*** or ***EHD*** (*Elastohydrodynamic*) and *Isoviscous Elastic* - ***IVE*** [2, 3]. These regimes are dependent by the effects of speed, load, and contact geometry, respectively. In fact, positioning the lubrication regime in one of the above categories determine the quality of the lubrication and the oil film thickness.

In the study case, the delineation of the lubrication regime was obtained by drawing the Johnson diagram [2], show in **Fig.1**, which tracks the changes of the elasticity parameter, g_e, and viscosity parameter, g_v.

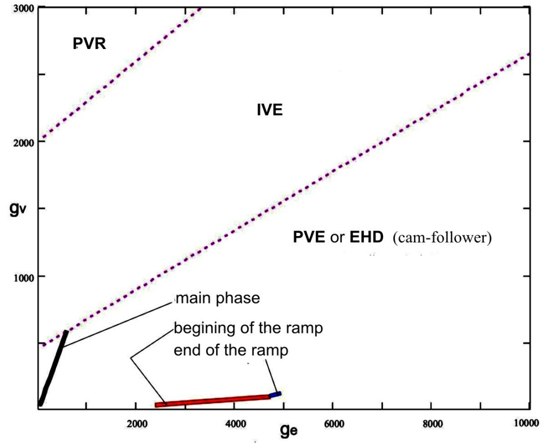

Fig.1. Positioning the lubrication regime in the Johnson diagram

As shown, at the entire movement of the flat follower (ramp and main phase), the cam-follower contact operate in the ***PVE*** lubrication regime (*also named elastohydrodynamic regime* - ***EHD***).

To estimate the heat effect on the lubricant viscosity and hence on the oil film thickness variation, in **Fig.2** was drawn the variation diagram of the thermal correction factor, Q_T.

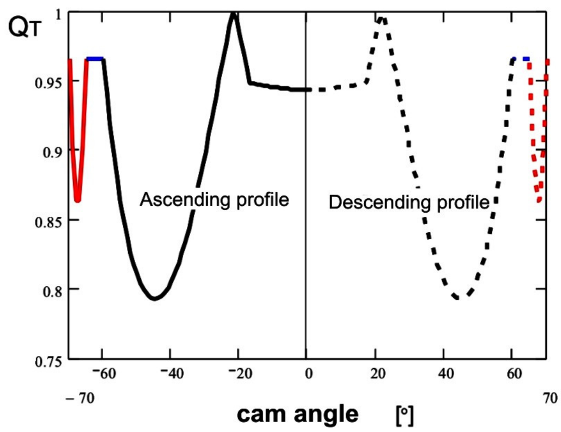

Fig.2. Variation of the thermal correction factor

Note that changing the oil viscosity caused by temperature increase in the film of lubricant can lead, in some areas of the active profile of the cam, to lowering oil film thickness up to 20% ($Q_T \approx 0,8$).

So, in the diagram of oil film thickness variation, this thermal correction leads to the flattening of maximum values area, but the influence is negligible in the minimum values area.

As previously mentioned, in real cases, the oil film thickness depends on the amount of oil available. A reduction of lubricant quantity from entering the couple leads to the starvation phenomenon, favored also by relatively high velocities of the contact surfaces. In **Fig.3** is visible the forming trend of zones with minimum values for the starvation factor. Thus, it appears that, for the portions of the cam positioned at angles of around ± 40^o symmetrically to the top of the profile, Q_{St} is approximately equal to 0.68. This translates into a significant reduction in oil film thickness (with approx. 32%), with obvious adverse effects on friction and wear in this couple.

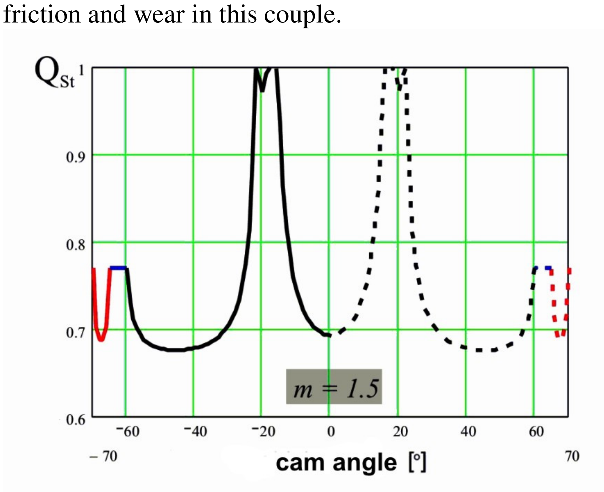

Fig.3. Variation of the starvation factor

For an overview of the cam-followers lubrication regime the Stribeck diagram has to be drawn as shown in **Fig.4.** This diagram, which incorporates lubrication parameter λ changes *vs.* the angle values, highlights comparatively the difference between the size of the oil film thickness and the roughness of the surfaces in contact.

As seen in the left diagram of Fig.4, ignoring the thermal effects and the starvation phenomenon, the lubrication parameter values, λ, are mostly higher than 1 (over 95% of the active profile of the cam), which confirms the elastohydrodynamic lubrication regime. Taking into account the reduction of lubricant viscosity with temperature, and the effects of starvation phenomenon, the actual values of parameters are on average about 50% lower. The right diagram of the figure shows a slight increase in the percentage of subunitary values, for approx. 10% of the active profile of the cam, this indicating the danger of occurrence of limit lubrication regime and an emphasized wear of surfaces, for both elements, in these areas.

In both diagrams of **Fig.4**, areas with subunitary values of the lubrication parameter are hatched and lubrication parameter values within the range [1, 3] are indicated by a gray area.

According to the Stribeck diagram, in this area, there is a danger that an elastohydrodynamic lubrication regime, in case of poor lubrication or high surface roughness, to make the transition to mixed lubrication regime, with obvious negative implications for the degree of wear of coupling. As seen in the left diagram, for an abundant lubrication that reduces the thermal effects and remove the danger of occurrence of the starvation phenomenon, the risk of developing a mixed lubrication regime significantly decreases. An issue worthy of consideration, in a normal or poor lubrication case, is the prevalence of this area in variation diagram of effective lubrication parameter, λ_{ef} .

A suggestive presentation of combined influence of the heat factor and the starvation on the oil film thickness in the couple is visible in Fig.5. As shown, unlike thermal effects that reduce the maximum values recorded for the oil film thickness, an inadequate lubrication, involving starvation phenomenon, induces an overall reduction of oil film thickness, including the minimum values. This phenomenon favors the interruption of the elastohydrodynamic lubrication regime.

Concluding over the results obtained in the analysis made within the theoretical model, the influence of thermal effects and the starvation phenomenon to reduce the oil film thickness can be significantly minimized if an additional contribution of oil is introduced between the cam and cam followers from the beginning of ascending race until the end of the descending race of the cam follower.

The principle of ***additional oil contribution*** of into couple is defined as an ***original solution*** for improving the contact conditions for the pair cam-follower and is named by the authors, ***LUJET***.

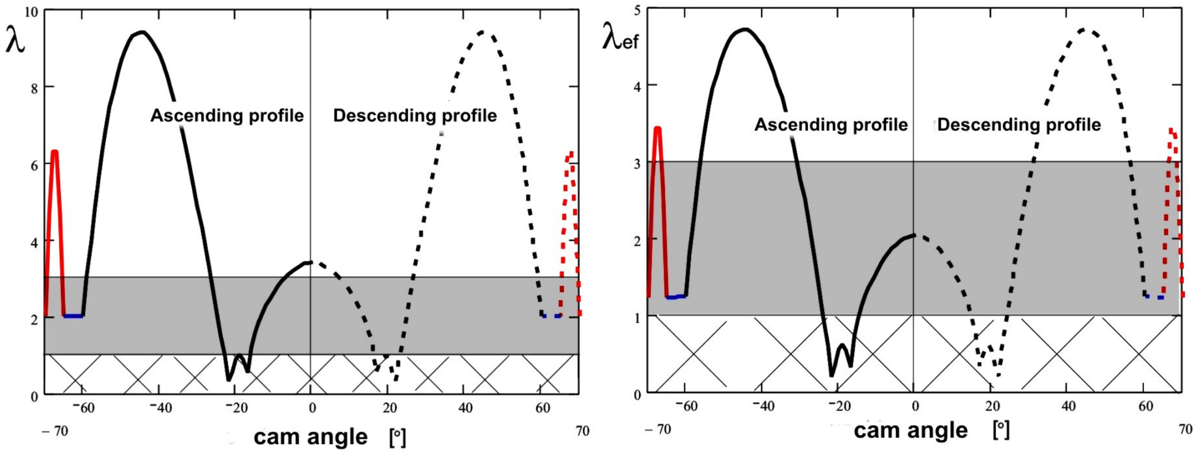

Fig.4. Comparative analysis of the lubrication parameter variation

This solution was the basis for a real model modified from a classical variant of the mechanism of distribution, considered as the standard model. In the next step, the experimental researches were focused on validate the theoretical results.

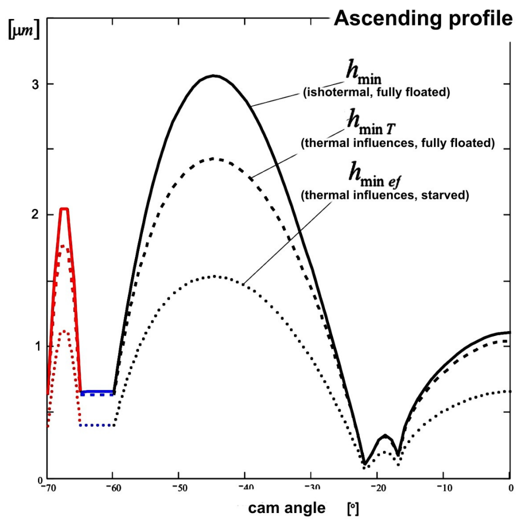

Fig.5. Reduced oil film thickness by heat and starvation effect

4. The experimental validation of physical-mathematical model

According to the requirements imposed to validate the proposed theoretical model and its analysis, the following determinations were established as necessary, determinations made before and after the testing regime, for both the standard and modified model:

- determining the geometric shape of the cam-followers contact and the follower law of motion;
- determining the inertial mass of the follower and the elastic constant of the spring;
- determining the oil temperature into the lubricating circuit and the average temperature of surfaces for cam and follower;
- measuring the roughness for the active profile of the cam and determining the lubrication parameters of the oil.

Dimensional measurements of the cams were made using a meter, type TESA Micro-Hite, equipped with optical sensor. Measuring accuracy is up to 0.1 μm for a length of 715 mm measuring stroke. Actual measurements were aimed to determine the law of motion for the cam followers.

In the measurement of the roughness was used a portable digital roughness meter, type TR 200. Measurements were made in 20 points, by dividing the angular profile of the cam.

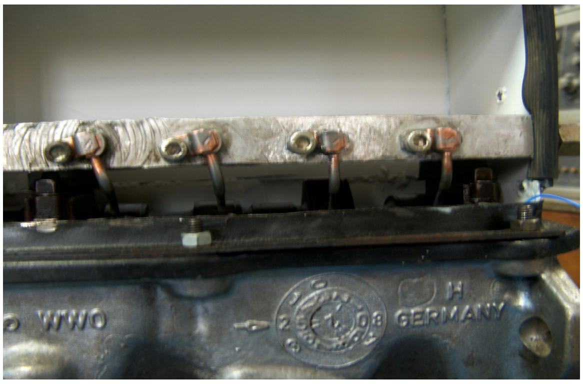

Fig.6. Rail for additional lubrication

Experimental test rig was designed to imitate with accuracy the actual operation of the distribution mechanism. For test was chosen a cylinder head coming from an 1800 cm^3 *VW* engine, type *ABS*, with camshaft mounted into cylinder head and hydraulic valve lifters. From rail, the oil reaches to the cylinder head through two circuits: one that provides lubrication standard model, the other provides the oil necessary to additional lubrication of the cam-followers contact. As seen in Fig.6, anointing further contact is made by means of nozzles (commonly used in piston bottom oil cooling, in 2000 cm^3 *VW* engines, mounted on a second common rail). These nozzles were chosen because they are composed of a valve ball which makes the oil stream format to be consistent in terms of flow. Additional flow of oil can be modified with a valve mounted on the supply pipe.

Doing a global assessment were found that, by using the proposed solution were obtained an effective reduction of the average wear of 25[µm] on the ascending profile and about 23[µm] on the descending profile, for the active surface of the intake cam. These results are found also for the exhaust cam. On the ascending profile the effective reduction of the average wear is about 30[µm], while in the descending profile it is about 22[µm]. Visualization of these experimental results was made in **Fig.7** and **Fig.8**, Confirming the predicted issues in the theoretical model, concerning the lubrication and wear thus contributing to its validation.

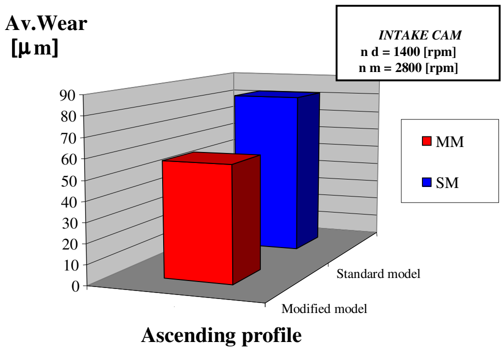

Fig.7. The wear average of the ascending profile - intake cam

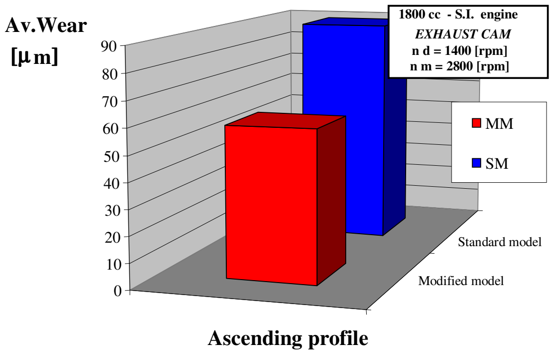

Fig.8. The wear average of the ascending profile - exhaust cam

The wear reductions provided by the proposed solution in relation to the standard solution may be express based on these absolute values into a more suggestive form by percentage differences, illustrated in **Fig.9** and **Fig.10.**

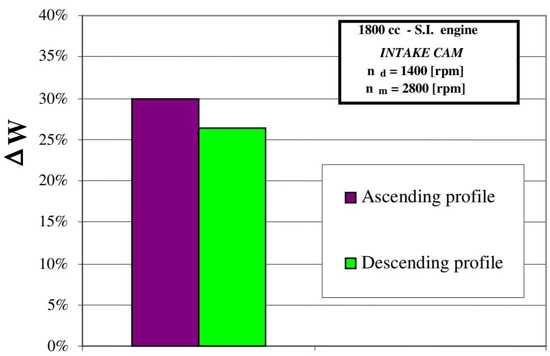

Fig.9. Reduction of the wear for the modified model versus standard model - the intake cam

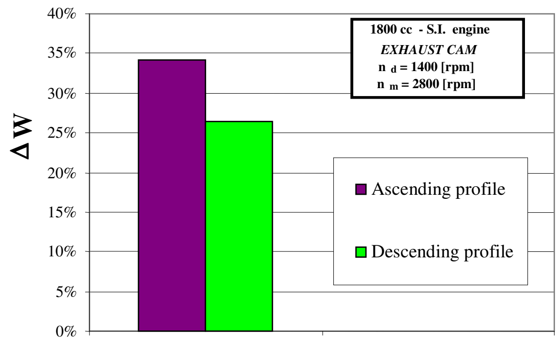

Fig.10. Reducing wear for the modified model versus standard model - the exhaust cam

Thus, for the intake cam, highlights a wear reduction about 30% for the ascending profile and 26% for the descending profile. For the exhaust cam, the wear reduction is about 34% on the ascending profile and about 24.5% for the descending profile.

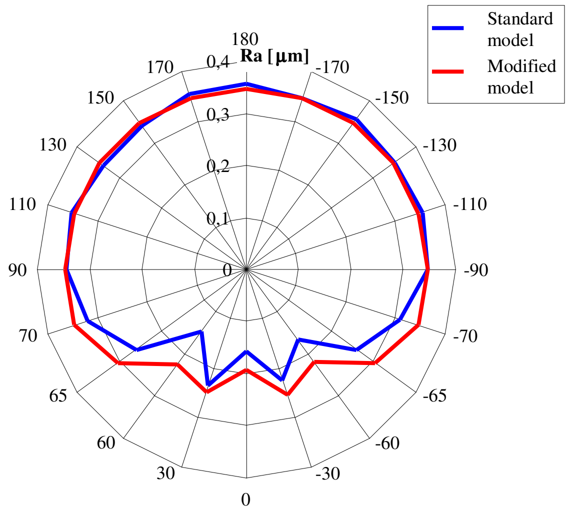

Fig.11. Average roughness of the intake cam on the circumference profile

Figure 11 shows the average roughness of the intake cam corresponding to both standard and modified models.

5. Conclusions

Because the theoretical model, in fact quite complex, does not enter the direct influence of roughness on the wear process, in the experimental measurements, the authors have proposed to highlight this possible influence. Thus, the changes in roughness size for the profile of ***modified model*** [***MM***] by increasing the supply of oil is smoother than in the ***standard model*** [***SM***], which shows that in reality there is a mutual conditioning between these two phenomena.

References

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[1]. C.M. Taylor, "Engine Tribology", Elsevier Science Publisher B.V., Tribology Series, Amsterdam, **26**, 75-88, (1993);

[2] .J.C. Tournier, "Tribological Analysis of Injection Cams Lubrication in Order to Reduce Friction and Wear", SET Halstad, 2007;

[3]. D. Zhu, H.S. Cheng, "An Analysis and Computational Procedure for EHL Film Thickness, Friction and Flash Temperature in Line and Point Contacts", STLE Tribology Transactions, Park Ridge, IL, USA, **32**, 364-370, (1989).

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