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THE MOTION EFFECT ON THE (EHD) BEHAVIOUR OF CAM MECHANISMS

BY

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ABSTRACT

The scuffing behaviour is one of the main factors associated with the performance of cam follower mechanism. The objective of the present work is to study the scuffing phenomena through the kinematic analysis of various effects of different types of motion on the elastohydrodynamic (EHD) lubrication properties in cam mechanisms. The contact points of cam follower surfaces at which flash temperature reaches maximum due to poor lubrication are specified. An examination had been developed at each point to investigate the possibility of scuffing occurance and mechanisms failure.

INTRODUCTION

The kinematic characteristics at contact of rubbing surfaces play important role in the (EHD) lubrication properties [1]. Bell and Dyson [1,2] discussed the effects of kinematic speeds on the scuffing of lubricated discs. The results show that the scuffing behaviour is governed by the (EHD) properties such as the flash temperature, oil film thickness, and friction power intensity. Recently the study of the (EHD) performance in cam mechanisms has been receiving increased attention by many researches. John and Herbert [3] concluded that the improving of contact surface resistance in cam follower system depends mainly on the quality of surface coating. Barwell and Roy [4] investigated the role of (EHD) lubrication on cam operation by the extension of the theory of lubrication. Dyson [5] analysed the kinematics and wear patterns of cam and follower valve gear. Also he discussed the theortical implications of changes in cam geometry of the film thickness of oil [6]. In references [7,8] attempts have been made to estimate the (EHD) film thickness between a harmonic cam and its follower. Ghoneam [9] analysed the (EHD) film thickness and flash temperature rise for a cycloidal cam.

The present paper outlines the effect of the follower motion on the (EHD) lubrication properties and scuffing behaviour in radial cam mechanism. An efficient computer program has been developed and utilized for analysis

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the effect of cycloidal and simple harmonic motion of reciprocating flat - faced follower With the use of fifth and eighth degrees polynomial the computed numerical results with illustrative figures are presented.

GEOMETRICAL ANALYSIS OF A CAM MECHANISM:

The (EHD) properties are governed mainly by the instantaneous radius of curvature, the effective load including the inertia term, Hertzian contact stress, and the relative speed of the surfaces in the inlet region.

In the case of the radial cam with a flat-faced follower; shown in Fig. (1); the instantaneous radius of curvature at the position cam angle (θ) is given by:

$$R_c = r_b + y + y'' \quad \ldots \ldots \ldots \quad (1)$$

where, $y'' = d^2y/dt^2 . 1/\omega^2$ = the reduced acceleration, ω is the angluar velocity rad/sec., β is the rise or return angle, r_b is the base cam circle, L is the maximum cam lift, and all characteristics of motion (y,y', and y") are given in the appendix.

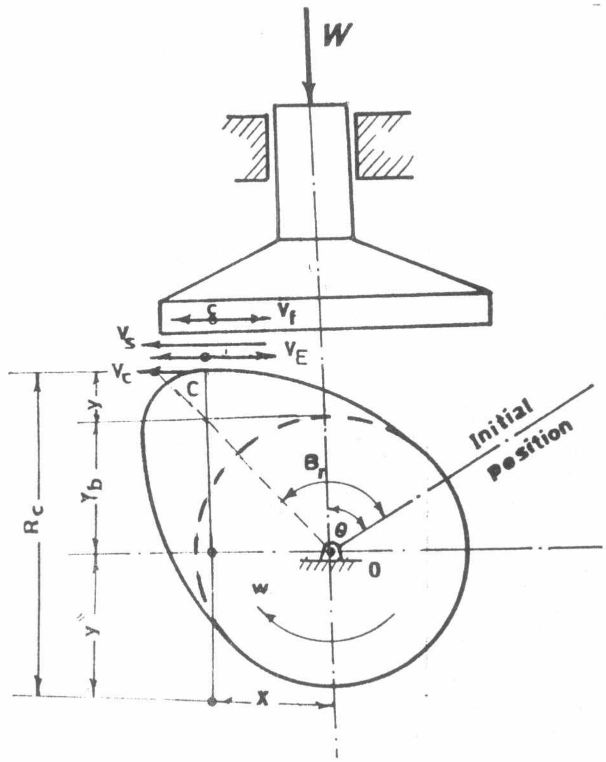

FIG.(1) Geometrical of radial cam with flat faced follower.

- In the absence of the frictional force in the slideways of cam follower, the total effective load on the cam is expressed by:

$$W = m.g + m\omega^2 y'' + F_s \quad \ldots \ldots \quad (2)$$

where m is the equivalent reciprocating mass of follower, g is the acceleration due to gravity, F_o is the initial spring force, K is the spring constant, and the spring force is then given by:

$$F_s = F_o + K.y \quad \ldots \quad (3)$$

- Herefrom the Hertizan stress (σ) is given by:

$$\sigma = C \, [W/R_c . b]^{\frac{1}{2}} \quad \ldots \quad (4)$$

where b is the width of cam disc, and constant C depends on the types of contact materials.

- In reference [9] the surface velocities for both cam and follower are given respectively by:

$$V_c = R_c \cdot \omega \quad \ldots \quad (5)$$

, and

$$V_f = y'' \cdot \omega \quad \ldots \quad (6)$$

where (V_f) is the lateral velocity of the point of contact with the follower.

The half summation of the values given by (5) and (6) is the entrainment velocity given by:

$$V_E = \tfrac{1}{2} \, (V_c + V_f) \quad \ldots \quad (7)$$

- The difference of the two velocities V_f and V_c will be taken as the sliding velocity (V_s) here as:

$$V_s \doteq | V_f - V_c | \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad (8)$$

THE (EHD) LUBRICATION PROPERTIES BETWEEN CAM AND FOLLOWER:

In the absence of squeez film lubrication, Dawson and Higginson [10] derived the formula for claculating the (EHD) film thickness (h) given by:

$$h = 1.6\ (V_E\ \zeta)^{0.7}\ \alpha^{0.6}\ E'^{0.03}\ R_c^{0.43}\ W^{-0.13} \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad (9)$$

Where, $E' = (1-\nu_f/E_f + 1-\nu_c/E_c)^{-1}$, ζ is the viscosity of lubricant, α is the pressure coefficient of viscosity, ν_c , ν_f=the possion's ratios of cam and follower mater als, E_c,E_f are their Young's modulii respectively.

- With the help of equations (5) to (8), the total flash temperature (T_f) in the oil film (between a flat follower and a cam) is given by [9,11]:

$$T_f = (C_1 . \mu . W^{0.75}\ |V_c - V_f|) / (\phi^{0.25} b^{0.75} (\alpha_{e_f} / \sqrt{V_f} + \alpha_{e_c} / \sqrt{V_c})\ R^{0.25}) \ldots (10)$$

Where constant C_1 depends on the form of the distribution heat flux between the two rubbing surfaces, μ coefficient of friction, α_{e_c} and α_{e_f} the thermal expansion of cam and follower respectively; and $\phi = 1/E'$.

RESULTS AND DISCUSSION:

The results of the analysis presented above may be described by the variation with either cam shaft angle or with time. With the help of Eq.(1), the cam radius of curvature is plotted against (θ) for the different types of motion as shown in Fig. 2. It can be noticed the critical point to avoid the undercutting of the cam profile at the maximum deceleration. Here from the entrainment velocities (V_E) as function of (θ) is computted and plotted for various cases as shown in Fig. (3). It can be seen that the resulting (V_E) is mostly positive. However, for the simple harmonic motion and eighth polynomial there is a small negative region with two transition points at which the V_E is zero. For the cycloidal and fifth polynomial, there are two negative regions with four transition points at which the V_E is zero. Figure (4) shows the effect of type of motion on the sliding velocities (V_s). It can be seen that the (V_s) is positive value in all cam cycle, and there is no large different of V_s values because the computation was done for the same r_b and maximum lift L. Equation (4) is used to calculate the Hertzian contact stresses (σ) for different motion which are plotted against θ in Fig. (5). It is obvious that the difference in σ is so small using the same spring properties in the calculation.
The very low local EHD film thickness, calculated from Eq. (9) for different motion, is shown in Fig. (6). It indicates that the scuffing and surface damage are to be expected, particularly in the region between the two zeros of the entrainment velocities (as shown in Fig. 7). With the help of Eq. (10), the flash temperatures and the cam shaft angle for different types of motion are computed and plotted in Fig.(8). So it can arrange the different motion with respects to the low temperature as eighth polynomial, simple harmonic, fifth polynomial, and cycloidal. The maximum flash temperature arround the cam profile for different motion is shown in Fig. (9), this indicates that the scuffing and failure may occur. From the results

presented in Figs. (5 through 9) one can deduce that the scuffing and surface damage probably occur at the regions which have poor EHD lubrication and maximum flash temperature (as presented in Fig. 10).

CONCLUSIONS:

- The algorithm for determing the scuffing position on cam follower surfaces, at which the damage and failure can be expected is developed.
- The kinematic analysis for cycloidal, simple harmonic, and various polynomial orders of motion in cam follower system shows the role of types motion on the (EHD) properties and scuffing failure.
- Taking in consideration the suitable (EHD) properties, and low scuffing the study shows that the eighth degree polynomial is the most convenient in practice followed by simple harmonic motion, cycloidal motion, and fifth degree polynomial.
- The requirements of good (EHD) film thickness, low flash temperature, and minimum Hertzian contact stresses are in conflict and required further investigation in future.

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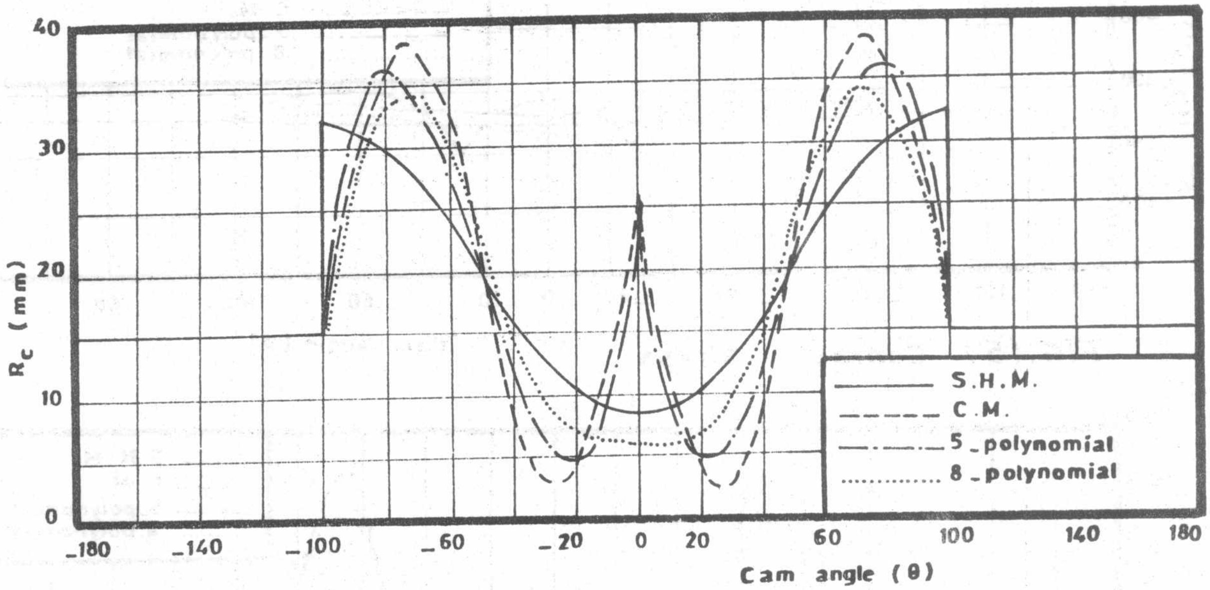

FIG. (2) Cam radius of curvature.

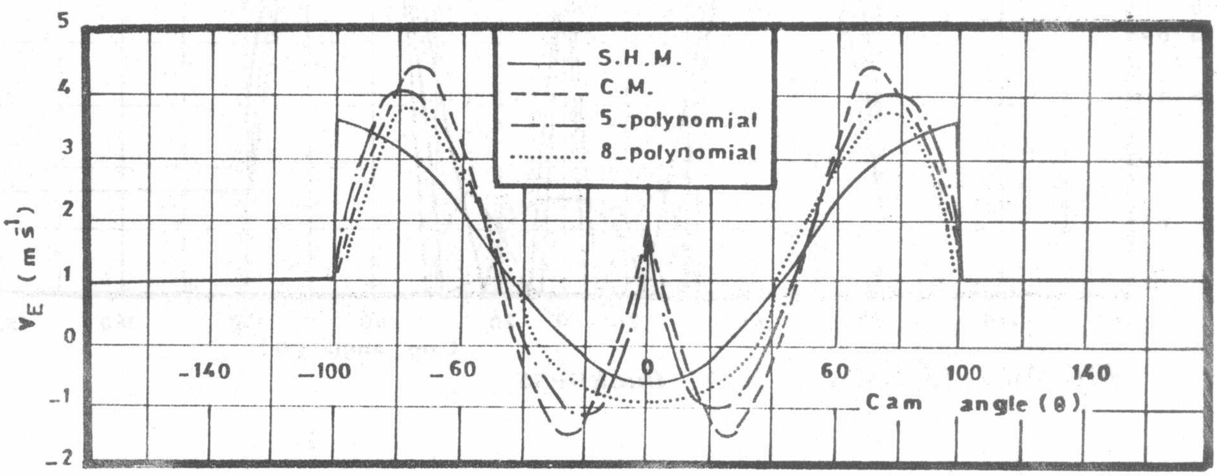

FIG. (3) The entrainment velocities.

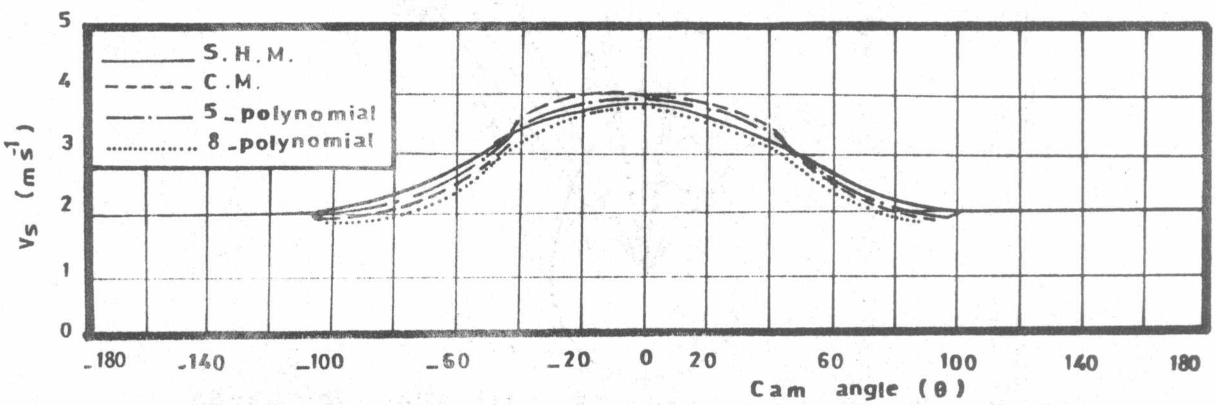

FIG. (4) Sliding velocities.

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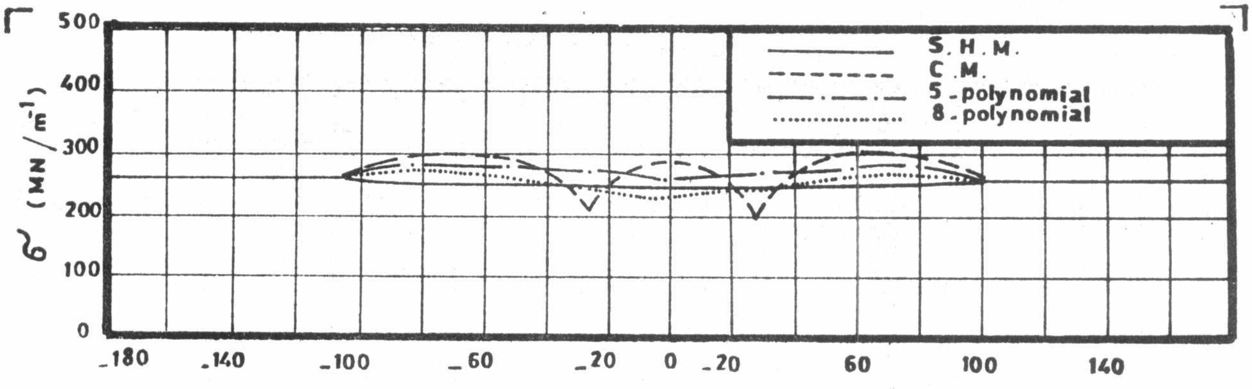

FIG.(5) Contact stress.

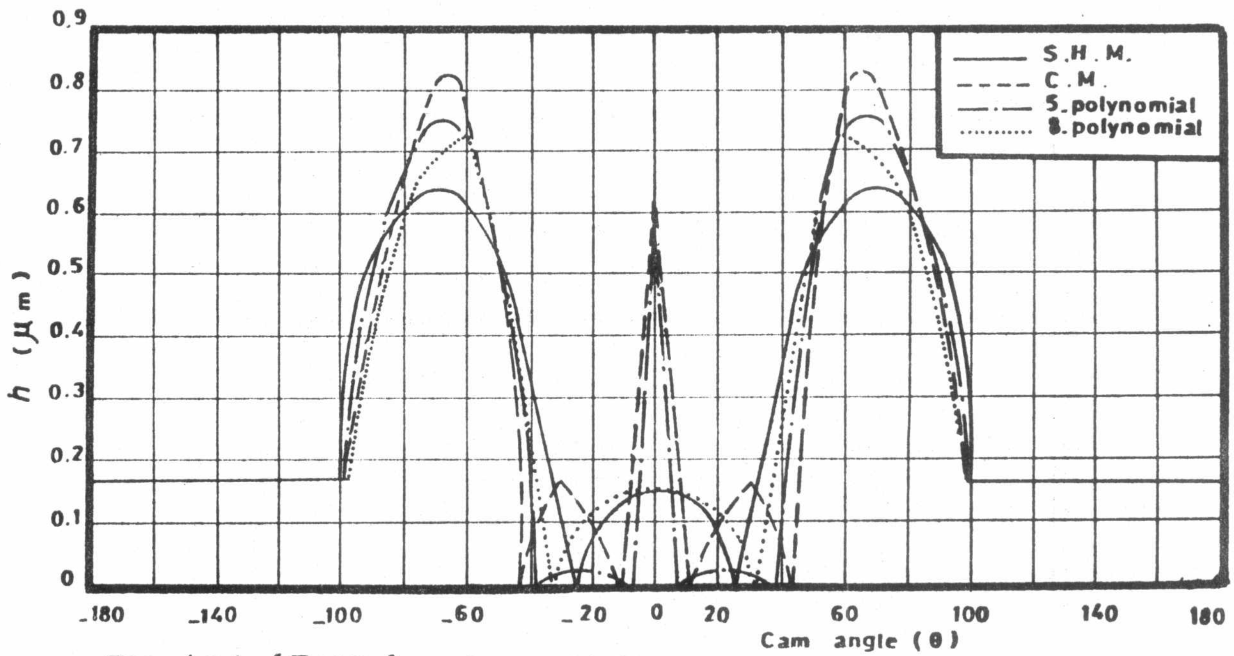

FIG.(6) (EHD) film thickness.

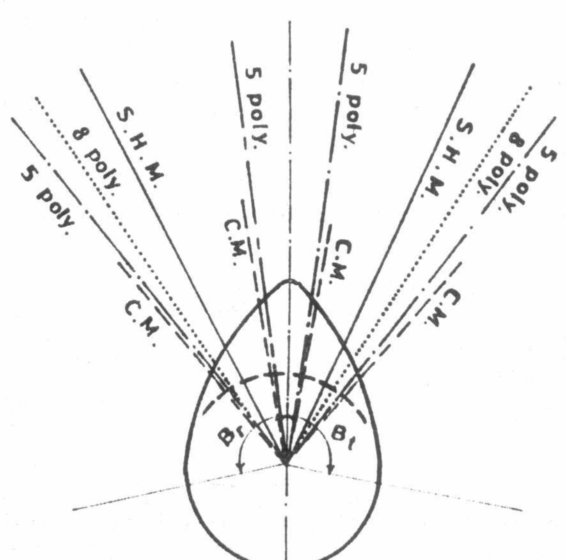

FIG.(7) Position of zero (E H D) film thickness for different types of motion.

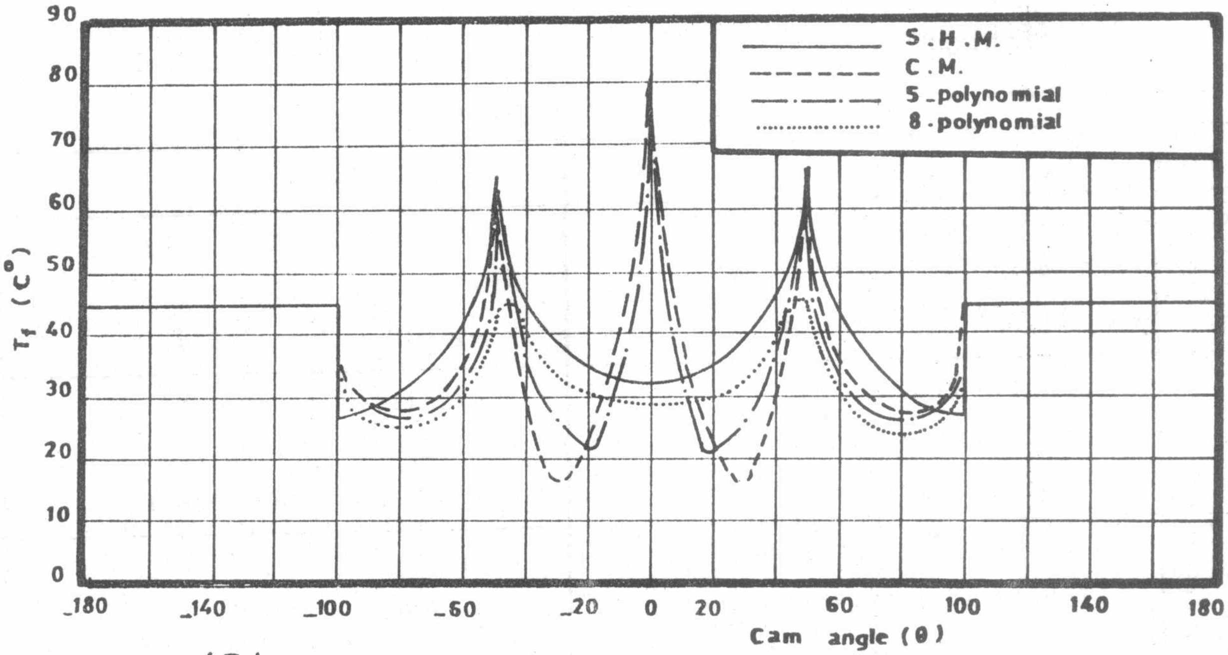

FIG.(8) Flash temperature.

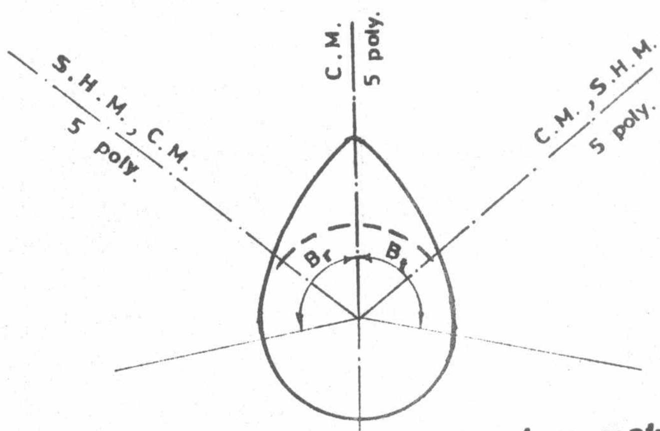

FIG.(9) Position of max flash temperature for different types of motion,

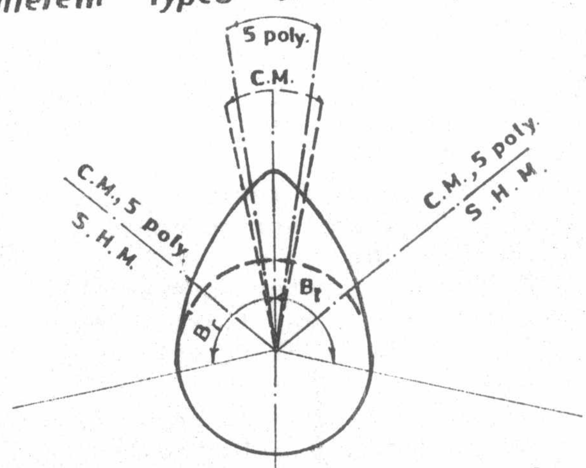

FIG.(10) Position of the scuffing for different types of motion.

APPENDIX:

I: Nominal Characteristics of Types of Cam Curves.

Type of Mot.	Rise	Return
cycloidal (C.M).	$y = L(\theta/\beta - 1/2\pi \sin 2\pi\theta/\beta)$ $y' = L/\beta(1-\cos 2\pi\theta/\beta)$ $y'' = 2\pi L/\beta^2 (\sin 2\pi\theta/\beta)$	$y_r = L(1-\theta/\beta+1/2\pi \sin 2\pi\theta/\beta)$ $y_r = L/\beta(1-\cos 2\pi\theta/\beta)$ $y''_r = -2\pi L/\beta^2 (\sin 2\pi\theta/\beta)$
S.H.M.	$y = L/2 (1 - \cos \pi\theta/\beta)$ $y' = \pi L/2\beta (\sin \pi\theta/\beta)$ $y'' = \pi^2 L/2\beta 2 (\cos \pi\theta/\beta)$	$y_r = L/2 (1 - \cos \pi\theta/\beta)$ $y'_r = -\pi L/2\beta(\sin \pi\theta/\beta)$ $y''_r = -\pi^2 L/2\beta^2 (\cos \pi\theta/\beta)$
Fif.-deg. pol.(5poly)	$y = 10L[(\theta/\beta)^3 - 1.5(\theta/\beta)^4 + 0.6(\theta/\beta)^5]$ $y' = 30 L/\beta[(\theta/\beta)^2 - 2(\theta/\beta)^3 + (\theta/\beta)^4]$ $y'' = 60 L/\beta^2[\theta/\beta - 3(\theta/\beta)^2 + 2(\theta/\beta)^3]$	$y_r = L - y$ $y'_r = -y'$ $y''_r = -y''$
Eighth-degree mation 8 poly	$y = L[6.09755(\theta/\beta)^3 - 20.78040(\theta/\beta)^5 + 26.73155(\theta/\beta)^6 - 13.60965(\theta/\beta)^7 + 2.56095(\theta/\beta)^8]$. $y' = L/\beta[18.29265(\theta/\beta)^2 - 103.902(\theta/\beta)^4 + 160.3893(\theta/\beta)^5 - 95.26755(\theta/\beta)^6 + 20.4876(\theta/\beta)^7]$. $y'' = L/\beta^2[36.5853(\theta/\beta) - 415.60800(\theta/\beta)^3 + 801.94650(\theta/\beta)^4 - 571.6053(\theta/\beta)^5 + 143{,}4132(\theta/\beta)^6]$.	$y_r = L[1.0 - 2.63415(\theta/\beta)^2 + 2.78055(\theta/\beta)^5 + 3.17060(\theta/\beta)^6 - 6.87795(\theta/\beta)^7 + 2.56095(\theta/\beta)^8]$ $y'_r = L/\beta[-5.2683(\theta/\beta)^2 + 13.90275(\theta/\beta)^4 + 19.023(\theta/\beta)^5 - 48.14565(\theta/\beta)^6 + 20.4876(\theta/\beta)^7]$ $y''_r = L/\beta^2[-5.2683 + 55.611(\theta/\beta)^3 + 95.118(\theta/\beta)^4 - 288.8739(\theta/\beta)^5 + 143.4132(\theta/\beta)^6]$.

II. Input Data:

1. Motion program is D-R-R-D. as 80°-100°-100°-80°-respectively maximum cam lift (L) 10.76 mm, Diameter of a cam base circle 30 mm, and width of cam disc (b) is 12 mm.
2. Dynamic properties, the equivelent of reciprocating follower mass is 1.005 kg, cam-shaft speed 1400 r.p.m, spring constant(k) is 22352 N/m, Perload of spring (F_o) is 181.0 N.
3. Material constant of cam follower system is steel have, $E_c=E_f=21\times10^{10}$ N/m², $\nu_c=\nu_f=0.3$, contact conductivity $\alpha_{e_c}=\alpha_{e_f}=1.3\times10^7$ erg. cm^{-2} Sec$^{-\frac{1}{2}}$. C$^{0-1}$ and $\sigma_{max}=1.035\times10^9$ N/m², the coefficient of friction(μ) is 0.05 in the (EHD) condition.
4. The straight minerial oil is used, and its properties as follow. The pressure exponent of viscosity(α) is 1.5×10^{-8} pa.$^{-1}$, the viscosity at the condition of entry to contact (ζ) is 1.3×10^{-2} pa. Sec.