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An Optimum Design of Cam Mechanisms with Roller Follower for Combined Effect of Impact and High Contact Loads

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Abstract

The problem in the design of a cam is the analyzing of the mechanisms and dynamic forces that effect on the family of parametric polynomials for describing the motion curve. In present method, two ways have been taken for optimization of the cam size, first the high dynamic loading (such that impact and elastic stress waves propagation) from marine machine tool which translate by the roller follower to the cam surface and varies with time causes large contact loads and second it must include the factors of kinematics features including the acceleration, velocity, boundary condition and the unsymmetrical curvature of the cam profile for the motion curve.

In the theoretical solution the unidirectional impact stress waves with the Mushkelishvilis inverse of the singular integral equation for contact stress have been used for analytical solution and a numerical solution have bean solved using F.E.M (ANSYS 10) for stress analysis in a cam surface at condition of rise-dwell-return (R-D-R) motion of the follower, also to compare the analytical and numerical results that have been used different pressure angles in the rise and return of the motion curves in unsymmetrical cam profile for optimum design.

Keywords: *Stress analysis; dynamic loading; impact loading; cam design.*

1. Introduction

In the design of cam-follower mechanism many factors have been affected on the performance of the translation the working loads from the electric or marine motors to the machine system by the follower system make the dynamic analysis becomes quite complex. These working loads may be classified for useful work performed by a machine to relative categories, gradually applied, suddenly applied and impact forces. The impact load arises in the machines which have either slow speed with heavy loads as in punching holes in tough sheet metal or high-speed system causes as a mechanical shock referring to an extreme abruptly applied force.

In cam follower system the cam size minimization is one of the important topics in cam mechanism design. These problems have been solved by many methods using graphical methods [1, 2], which take the effect of cam motion on the curvature of the cam profile and the amount of offset of the follower that need in a specific design. In recent years [3], designer use CAD digitized packages to generate the cam profile on the drawing and used optimization techniques for cam-follower designs begin with crude sketches using Cartesian coordinates.

A spline functions have also been used to produce very flexible motion for cam synthesis. Splines afford the designer greater local control and therefore are more adaptable when design constraints which must satisfy discrete cinematic constraints and preserve continuity locally. In [4,5] a method used for unsymmetrical cam design employ splines in particular piecewise continuous polynomials called loasis-splines (B-Splines) to have continuous derivatives up to any order and often need only be cubic functions, regardless of the number of constraints present.

Spline functions have been used to produce very flexible motion for cam synthesis taking into consideration the kinematic features such as velocity and acceleration with formulation up to five parameters which are independent functions used to modify the cam profiled.

Automatic assembly machines have many cam-driven linkages that provide motion to tooling, the dynamic behavior of the components includes both the gross kinematic motion and self-induced vibration motion developed by Kalil [6] used solid works CAD software (Pro/Engineering). A three-mass two-degree of freedom dynamic model was created in Simulink taking in a count the impact and the over-travel event of cam follower system machine to obtain improved performance. In [7] several dynamic models that are suitable for the determination of dynamic behavior and residual vibrations within cam-follower systems, as well as providing the means to predict follower jump, the follower motion were defined using 3-4-5 polynomial in rise and return using MATLAB to determines the cam profile as well as cam displacement, velocity and acceleration diagrams. [8] Presented a mathematical model of a cam-driven pattern mechanism by setting differential equations describes the behavior of the system motion of the cam and the follower taking into account the elasticity of its elements and the inertia forces resulting from the oscillatory motion of the pattern cause the speed of driving the cams to fluctuate. [9] Shows that for high-performance engines its need cam shaft design system for computing programs for the design of different types of valve trains. This system is the development of high-quality valve acceleration curves that comply with the hydrodynamic fringe conditions of the charge cycle while providing an oscillation-attenuated valve train which subjected to little dynamic stress.

In this study a present design method for cam synthesis to solve the design task due to the complete non-linear relationships for the constraints including acceleration, velocity and boundary conditions in the process of optimization taking into consideration the effect of impact loading on the curvature of the cam profile and the amount of offset of the follower. A numerical solution using a proper programming for solving the stress analysis in the cam contact using (ANSYS 10) for supporting the analytical solution.

2. Theoretical Analysis of Boundary Conditions

2.1. Impact Forces

Impact is often called mechanical shock referring to an extreme abruptly applied force [10]. The sources of impact in cam-follower mechanisms could be the result of:-

1- Backlash in a positive drive cam and roller follower.
2- High speed systems which are non-linearly elastic so that a rapid change occurs with results similar to impact.
3- The working load action as a cam-driven punching mechanism.

In presented study the contact between the cam and the follower is under dynamic load at low speed of engine, but at high speed the contact load becomes very high at infinitesimal time causes a sudden impulse load. The response of the cam and follower materials to low velocity impact it must be determined the impacter – induced surface pressure and its distribution and then finding the internal stresses distribution in the cam surface which may be causes failure modes in the contact region of cam-follower mechanism.

If the mass and velocity of the follower as m_1 and v_1, and the cam mass velocity as m_2 and v_2 respectively.

Then the rates of change of velocity during impact as:-

$$m_1 \frac{dv_1}{dt} = -p \qquad m_2 \frac{dv_2}{dt} = -p \quad \ldots(1)$$

If y dente the distance that the follower and cam approach each other then for low velocity impact [11]:-

$$y^{\bullet} = v_1 + v_2 \quad \ldots(2)$$

In Fig. 1 it is shown that the contact region between the followers as an elastic rod with the edge of the cam as an elastic half-plane, the roller of the follower at the motion on the cam surface may be extended as a straight part of the roller boundary of length a and the corners are of radius R.

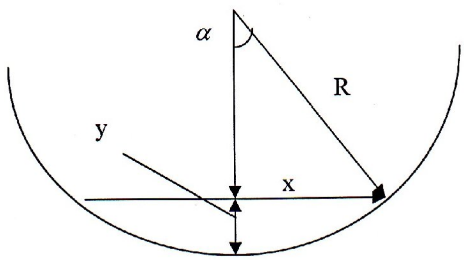


Fig.1, Geometry of Cam-Follower Contact Near Circular Limit.

The general solution in the case of asymmetrical complete or incomplete contact over the range $-b \le x \le b$ is given by Mushkelishvilis inverse of the singular integral equation as [12]:-

$$p(x) = -\frac{x}{\pi\sqrt{(b^2 - x^2)}}[P - \frac{E^*}{2}\int_{-b}^{b}\frac{h^/(t)\sqrt{(b^2 - t^2)}}{t - x}dt] \quad \ldots(3)$$

where

$b =$ the contact length for which the contact load is ceases and depend on the pressure applied between the two surface of contact.

$E^* =$ measure of the composite stiffness of the follower and cam.

$$\frac{1}{E^*} = \frac{1}{E_1}(1 - v_1^2) + \frac{1}{E_2}(1 - v_2^2)$$

$h^/(x) =$ the slope at each point on the profile of the contact region.

From [13] it was developed that the value of $h^/$ could be obtained as:-

$$h^/(x) = \frac{2}{E^*}\frac{1}{\pi}\int_{\lambda}\frac{P(\lambda)d\lambda}{x - \lambda} \quad \ldots(4)$$

and the value of applied load P is in equilibrium with the pressure distribution taking $p(\pm b) = 0$ is obtained as:-

$$P = -\int_{-b}^{b} p(x)dx = -\frac{E^*}{2}\int_{-b}^{b}\frac{h^/(t)tdt}{\sqrt{(b^2 - t^2)}} \quad \ldots(5)$$

now assuming that the cam is stationary and the follower is moving at velocity v_1, then the energy balance

$$\frac{1}{2}m_1 v_1^2 = \int_0^y pdy$$

Since $v_2 = 0$ and $dy = -x.c\sec^2\alpha.d\alpha + \cot\alpha.dx$

$$\therefore \frac{1}{2}m_1 y^{\bullet 2} = \int_{-\alpha}^{\alpha} p.x.c\sec^2\alpha.d\alpha + \int_{-b}^{b} p.\cot\alpha.dx \quad \ldots(6)$$

$$\because y = r_1\sin\theta \quad \therefore y^\bullet = r_1\cos\theta\frac{d\theta}{dt} + \sin\theta\frac{dr_1}{dt}$$

$$y^\bullet = r_1\cos\theta.w + \sin\theta\frac{dr_1}{dt} \quad \ldots(7)$$

Substituting eq. (7) into eq. (6) gives:-

$$\frac{1}{2}m_1(r_1\cos\theta.w + \sin\theta\frac{dr_1}{dt}) = \int_{-\alpha}^{\alpha} p.x.c\sec^2\alpha.d\alpha + \int_{-b}^{b} p.\cot\alpha.dx \quad \ldots(8)$$

2.2. Solving the Boundary Condition

In this research the new technique that have been considered for finding the profile of the cam which is defined in to circles of radius r_i for rise and return stroke and at a distance d from the center of rotation of the cam as shown in Fig. 2.

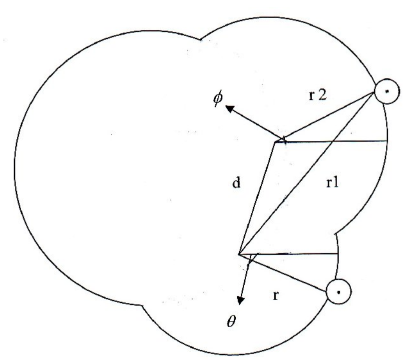


Fig.2. The Effect of Radius of Curvature and Pressure Angle for the Cam Contact with Circular Ended Follower.

This boundary condition required to be solved by two statements:-

1- For the same point on the contact region between the follower and the cam the same velocity must be taken into consideration, that is:-

$$y = d + r_i \sin\phi$$

$$\therefore y^{\bullet} = \frac{dr_i}{dt}\sin\phi + r_i \cos\phi \frac{d\phi}{dt} \qquad \ldots(9)$$

For constant r_i then $\frac{dr_i}{dt} = 0$

2- The pressure angle γ must have two value for the max value of both the rise interval $\gamma_1 \max$ and the return interval $\gamma_2 \max$, with satisfying the condition for determine its value which can be calculated as:-

$$y^{\bullet} = r.w.\tan\gamma \qquad \ldots(10)$$

2.3. The Polynomial Solution for Optimum Design of Cam

The application of algebraic polynomials must satisfy the differential equations of motion of the cam-follower system R-R-D which have the form:-

$$y = C_0 + C_1\theta + C_2\theta^2 + \ldots\ldots\ldots\ldots\ldots + C_n\theta^n \qquad \ldots(11)$$

Eq. (11) is normalized such that the rise h and max cam angle β will both be set equal to unity.

The number of terms of eq. (11) must be limited to desire design so that it will be acceptable at high speeds and this can be solved by satisfying the boundary condition of the motion.

For the strength of the follower and the cam the polynomial equation of y must satisfied eq. (6) and eq. (8) for dynamic loading with eq. (9) and eq. (10) for the curvature of cam profile and specified pressure angle. In the graphical method to minimize the cam size, two important factors that affect the cam size are the permitted maximum pressure angles of both the rise and the return intervals of the cam angle, so that

$$\begin{aligned} \gamma_1 - \gamma_1 \max \geq 0 \\ \gamma_2 - \gamma_2 \max \geq 0 \end{aligned} \qquad \ldots(12)$$

where $\gamma_1 \max$ and $\gamma_2 \max$ can be obtained by substitute eq. (10) in eq. (9) and differentiating with respect to r_i, then equating to zero. This procedure will give the specified value of γ that solved eq. (12).

The acceleration of the follower a is an important factor for reducing the forces exerted by the follower on the cam edge in rise and return section which can be compared with the $a.\max$ for minimum cam size by using the following relation

$$|a| - |a.\max| \geq 0 \qquad \ldots(13)$$

Taking the absolute value for the acceleration in the form of eq. (13) for negative and positive value of acceleration in rise and return sections while $a\max$ can be obtained by differentiate eq. (9) twice and then equating to zero which give min r_i for a max. eq. (12) and (13) are very complicated and must be solved by using two programming methods in (Mathlab 2000 and Mobile) package.

3. Sample of Calculation

In this review some restriction must be modified for the curve motion of the follower with the cam profile used for optimum design. For example the system motion of R-R-D used in cutting machine have the max lift of the follower is 16 units, the main radius of the base circle for the cam is 15 units, the permitted pressure angles for both the rise and the return intervals are $\alpha_1 = 30^\circ$ and $\alpha_2 = 45^\circ$,and this is because the force acting in the return section is not from the cam as in the rise, so that the pressure angle in the return is larger value compared with rise motion. The maximum acceleration of the follower is $a_{\max} = 9 \; units/\sec^2$ for the cutting machine.

For solving this problem two boundary conditions have been taken to satisfied and obtained the polynomial functions as follow:-

1- Taking the initial condition as:-

$\theta = 0$, $y = 0$, $y^{/} = 0$, $y^{//} = 0$, $y^{///} = 0$

where $y^{/}, y^{//}, y^{///}$ are the velocity, acceleration and jerk of the motion of the follower,

$\theta = \beta$, $y_{\max} = 16$ units , $y^{/} = 0$, $y^{//} = 0$, $y^{///} = 0$

So the polynomial functions $y = C_n \theta^n$ must be of $n = 7$.

2- For optimum design of the cam profile, the kinetic conditions have been used by solving eq. (8), (9), (10), (12) and (13) added to the 8 boundary conditions that have been taken in step 1.

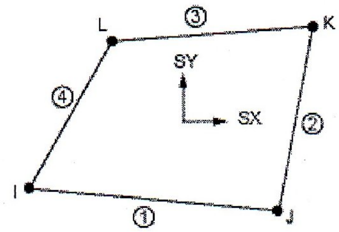
So that the polynomial functions of 12 orders will take the value of $n = 11$ and the velocity and acceleration diagrams have less values than for 8 polynomials as shown in Figures 7, 8, 9, and 10 respectively.

4. Numerical Analysis

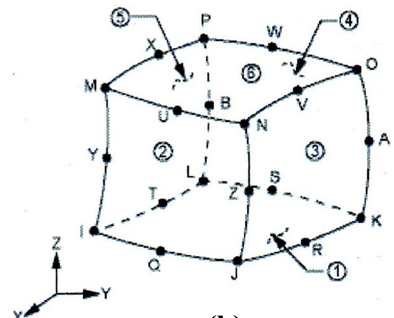
Finite element method is one of the most accepted and widely used tools for the solution of linear and non linear partial differential equations which arises during the mathematical modeling of various processes.

Two models of cam are considered cam(1) and cam(2) using the commercial code ANSYS 10, the cam bodies are modeled using PLANE 42 solid structural includes four nodes and SOLID 95 solid structural with 20 node surface elements, respectively as shown in Fig.3. The material employed in this modeling for the tow types of cam was steel, the element material properties defined as linear-elastic-isotropic with modulus of elasticity of 210 GN/m^2 and poison ratio of 0.3. the two types of cam are identical in the base circle (5 cm diameter), wide surface of cam (1 cm), and the sequence operation of the follower Rise-Return-Dwell (R,R,D), but differed in the position and size of the rise and return circle.

Cam(1) have a rise circle of 16 cm diameter with a center position of 120° from the ground circle start Dwell and a return circle of 12 cm diameter with a center position of 225° from the ground circle, the cam rise-return-dwell motion is as follows: rise angle $14^{\circ} - 184^{\circ}$, return angle $184^{\circ} - 300^{\circ}$ and dwell angle $300^{\circ} - 14^{\circ}$. Cam(2) have a rise circle of 18 cm with a center position of 135° from the ground circle start Dwell and a return circle of 14 cm diameter with the same center position as cam(1), the cam rise-return-dwell motion is as follows: rise angle $8^{\circ} - 192^{\circ}$, return angle $192^{\circ} - 315^{\circ}$ and dwell angle $315^{\circ} - 8^{\circ}$ as shown in Fig. 4.

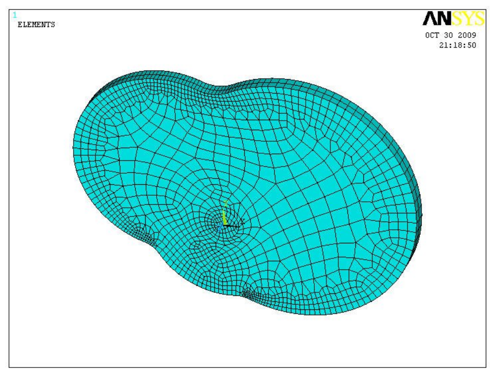


(a)

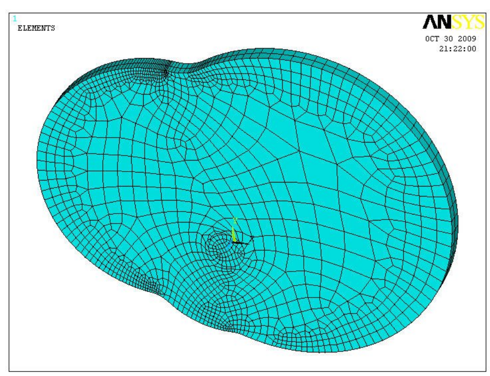


(b)

Fig.3, Element Geometry; (a) Plane 42, (b) SOLID 95.



a- Cam (1)



b- Cam(2)

Fig.4, The Curvature of Cam Profile for; (a) First Polynomial, (b) The Size Optimization of Cam.

The accuracy of the FEM depends on the density of the mesh used in the analysis; the outer surface of cam loaded by the follower so that to obtain the correct stress and displacement function in the region of high stress it was necessary to have a more refined mesh in this region.

The roller follower is the most commonly follower used, so that the contact surface between the follower and cam assumed to be a line contact. The forces are taken from the calculation of the dynamic polynomial function are used as input for the follower rotation to calculate the stress and deflections. The pressure on the top surface of the cam applied as a line pressure with parabolic shape having a maximum value at the center, with 25 different position / angles each interval have 15°.

Figures 5 and 6 gives a sample of stress and deflection ANSYS analysis of cam (1) with a pressure load in rise cycle at angle of 105°, respectively.

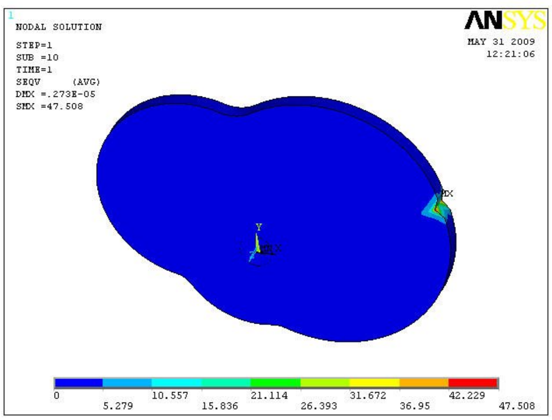


Fig.5. Stress Analysis.

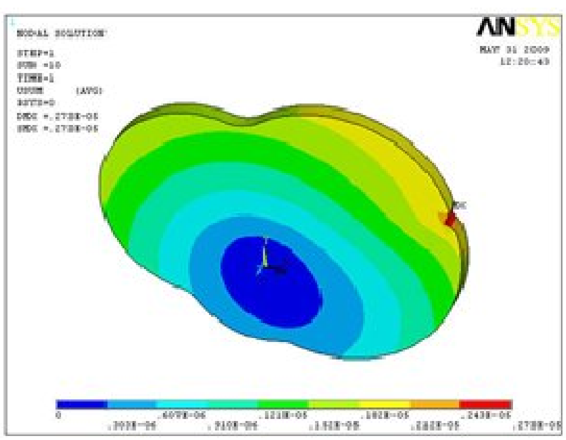


Fig.6. Deflection Analysis.

5. Results

For the optimization of the results it can be shown from Fig.7 and 8 that the act of additional parameters on the boundary conditions which have been used in this research give minimization in the radius of curvature r_i for the cam in the rise and return motions of the follower which are bound by max pressure angles α_1 max and α_2 max.

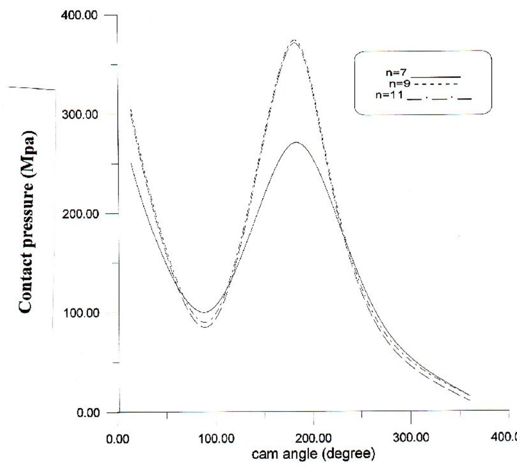


Fig.7. The Distribution of Contact Pressure Along The Edge of Cam (1) by Analytical Solution.

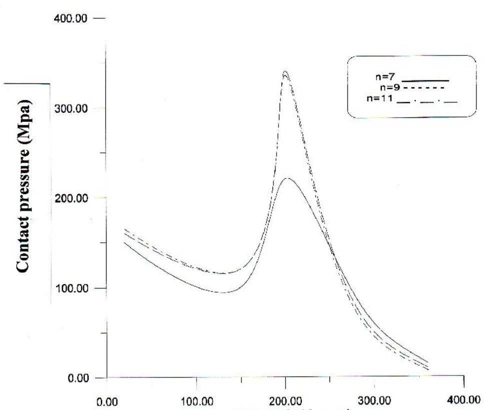


Fig.8. The Distribution of Contact Pressure Along The Edge Of Cam (2) by Analytical Solution.

Both the velocity continuity along the edge of the curvature of the cam and the discontinuity of acceleration at the rise and return section with using the conservation of energy which result from low velocity impact between the follower and the cam give some simplicity for solving the integration of the equations which used to find the optimum size of positive and negative curvature.

The results shows also that the stress distribution in the cam surface must be minimized especially that the contact force have different values along the rise and return motion of the follower and different accelerations which results from this variation, the external forces will be minimized for optimum design of the cam. In Fig. 7 and 8 it can be shown the position of θ that at which the stress will be minimum depend on the order of the polynomial functions used to solve the equations.

In Fig. 7 the analytical solution for cam (1) shows that the pressure contact for polynomial $n=7$ have great values and become stable for polynomial degree of varying $n=11$ which has less difference when compared with polynomial $n=9$.

When the dimensions of the cam change with respect to the same radius of curvatures but with different pressure angle as in cam (2) it can be shown in Fig. 8 that for for the same contact load as in cam (1), the pressure will be decrease and become more stable for polynomial function of $n=9$ and $n=11$, this is because the kinetic motion which have induced in the design of the radius of curvature for the cam become more stable than cam (1).

For any cam mechanism it must be drawn the acceleration diagram with cam angle (θ) to illustrate the type of acceleration for the motion of the follower and this can be shown in Fig. 9 and 10 , for cam (1) and cam (2). The cycling of the acceleration for the polynomial functions are more stable for cam (2) and this is because the parameters of the analytical solution for each of rise and return intervals are limited to max and min pressure angle but the different between the six polynomials resulting cam profiles have limiting values of the constraint on the radius of the positive curvature have less effect than the negative curvature and give a critical factor in the size minimization.

These results have confirmed by the numerical solution using ANSYS 10 which have been shown in Fig. 11 and 12, for the distribution of the effective stress with the angle of contact along the cam edge which give low variation of the values of the stress with the optimum design for cam (2) rather than for cam (1). It can be seen when comparing figure 7 with figure 11 for cam (1) the analytical and numerical results shows the same minimum value of stress at angle 100˚ with stress 100 N/cm² and then become maximum at angle 200˚ with 350 N/cm² while for cam (2) the analytical solution gives minimum value at angle 140˚ with 110 N/cm² and maximum value of 310 N/cm² at angle 205˚ as shown in fig (8) when compared with the numerical one which have minimum stress of 100 N/cm² at 150˚ and 300 N/cm² at 210˚ with the same trend as shown in fig. (12).

Also it can be seen the variation of the dynamic displacement along the contact angle between the follower and the cams (1) and (2) as shown in Fig. 13 and 14, which confined with the distribution of the absolute values acceleration because the acceleration multiplied by the mass of the follower give the forces which as a result indicate the deflection and displacement in the direction of the force at any (θ) of cam angle.

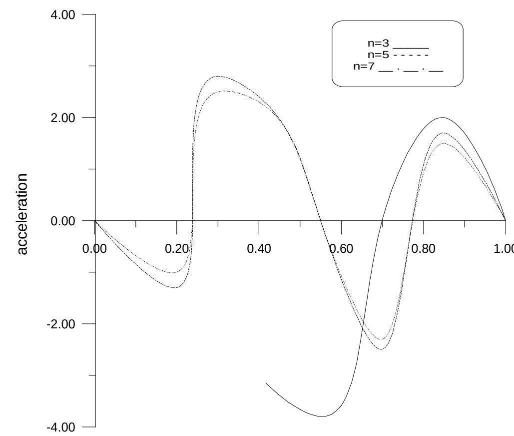


Fig.9. The Acceleration Curve of The Optimum Results with Normalized Cam Angle for Cam (1).

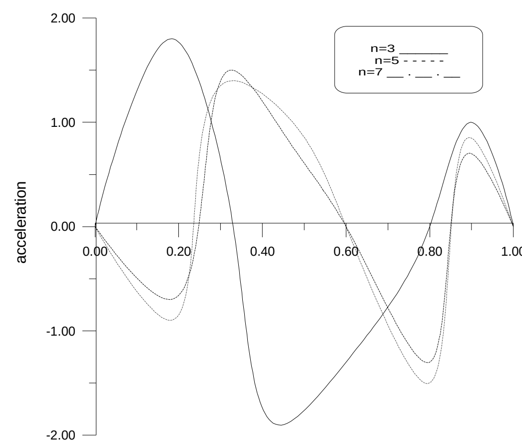


Fig.10. The Acceleration Curve of The Optimum Results with Normalized Cam Angle for Cam (2).

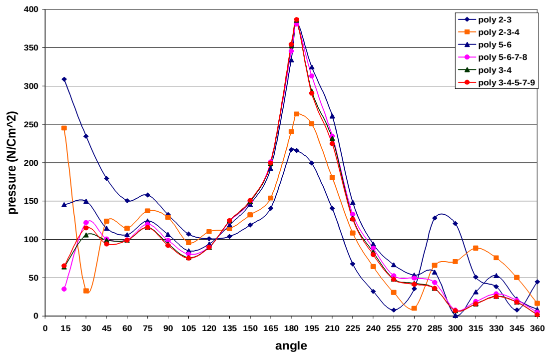


Fig.11. Stress Function Cam (1) FEM.

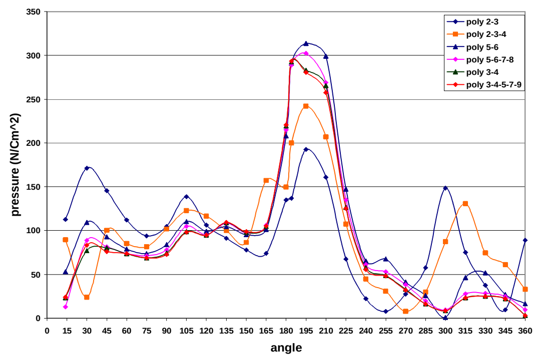


Fig.12. Stress Function Cam (2) FEM.

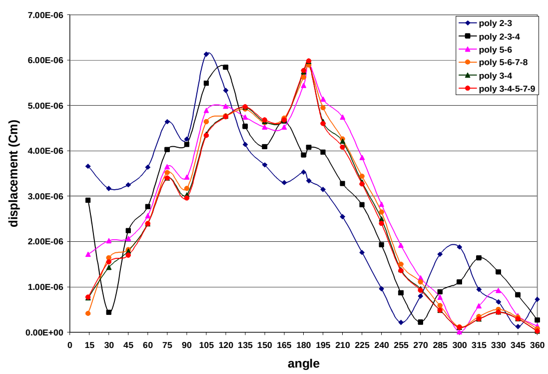


Fig.13. Deflection Function Cam (1) FEM.

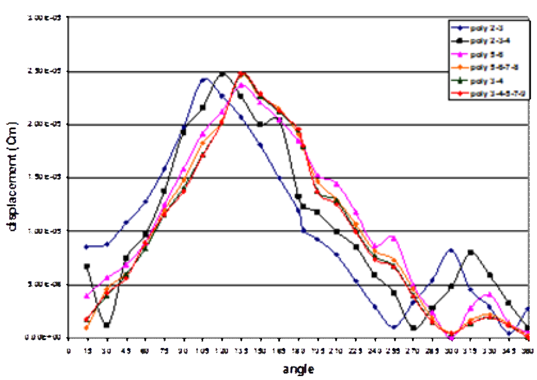


Fig.14. Deflection Function Cam (2) FEM.

6. Conclusions

For optimum design of a cam with roller follower a modified method have been studied in this paper taking in the consideration the kinetic boundary condition with low velocity impact with the kinematic condition for solving the parametric polynomial function that proposed to satisfy the curvature and minimum cam size.

In this method it have been taken the optimum stress distribution along the contact between the follower and the cam edge which give good agreement with the numerical analysis using F.E.M. and the results shows some advantage in solving non-linear programming for prescribing the motion, velocity, and acceleration curves as well as the various constraints related to the pressure angle for return-rise-dwell system of motion of the follower.

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الخلاصة

أن تصـ ميم الحـ دبات يحتـ اج الـ ى تحليـ ل ميكانيكيـ ة حركـ ة الحدبـ ة مـ ع تـ أثير القـ وى الديناميكيـ ة الموزعـ ة علـ ى طـ ول حافـ ة الحدبـ ة وعلـ ى العوامـ ل الخاصـ ة بالمعادلات الرياضية لسلسلة المحددات (parametric polynomials) والتي تصف الحركـ ة الدورانيـ ة للحدبـ ة. فـ ي هـ ذا البحـ ث تـ م اسـ تخدام طـ ريقتين لغـ رض تحليل المعادلات الرياضية والحصول على أفضـ ل تصـ ميم لشـ كل الحدبة. الطريقـ ة الأولـ ى يـ تم فيهـ ا ادخـ ال القـ وى الديناميكيـ ة العاليـ ة القيمـ ة والخاصـ ة بالصـ دمة والموجات المرنة الناتجة من هذه الصدمة والتي تؤثر على حركة التابع وتكون متغيرة مع زمن تلامس التابع مع حافـ ة الحدبـ ة خـ لال زمـ ن الصـ دمة. والطريقـ ة الثانية هي أدخال العوامل الكينامتيكية في المعادلات الرياضية والتي تشمل التعجيل والسـ رعة وغيرهـ ا مـ ن الظـ روف المحيطـ ة بميكانيكيـ ة حركـ ة الحدبـ ة وعـ دم تماثل المنحنيات الخاصة بشكل الحدبة.

أن الحـ ل الرياضـ ي لهـ ذه المعـ ادلات يحتـ اج الـ ى اسـ تخدام قـ انون الصـ دمة بأتجـ اه واحـ د مـ ع قـ انون الـ تلامس الخـ اص بـ Mushkelishvilis مـ ع حـ ل سلسـ لة المحددات الرياضية. لقد تم استخدام حلول عددية بأستخدام ANSYS10 لمطابقة نتائجه مع حلول معادلات وصف حركة الحدبة لأفضل تصـ ميم لشـ كل الحدبـ ة الخاصة بنـ وع حركـ ة التـ ابع (Rise-Return-Dwell) ولأنـ واع مـ ن زوايـ ا الضـ غط الخاصـ ة بأرتفـ اع ورجـ وع التـ ابع مـ ع عـ دم تماثـ ل انحنـ اء حافـ ة الحدبـ ة للحصول على أفضل تصميم لها.
