Infinitely Variable Valve Lifting

A. Raveendra, K. Sri Noothan Reddy

***Abstract*: A new mechanism of Inlet Camshaft with valve lift varying infinitely for Internal Combustion engines is presented. In this, the valve opening and closing is done by a three-dimensional cam or camoid with a translating spherical follower. And the camoid is actuated by a Governor which rotates about a horizontal axis. Also, the design procedure for spring and analytical expressions for Governor are generated. And the design of camoid profile is defended by employing the theory of envelope. A numerical example is given to explain the application of the approach.**

Index Choice: camoid, governor, variable valve lifting.

I. INTRODUCTION

CAMSHAFT is an integral part of the Internal Combustion engine which helps the cams to actuate the valves. Valve lift, Duration, Lobe separation, and Timing are the main aspects of a camshaft for better performance or efficiency of an Internal Combustion engine. For an efficient engine, there are some advanced technologies used in the camshaft mechanism like VVT (Variable Valve Timing) and VVL (Variable Valve Lifting). Where VVT is a method which helps to alter the valve opening and closing timing. And VVL is method which helps to change the amount of valve lift. Now every company is manufacturing the vehicles with either one of VVT & VVL technology or both. And they have their own type of VVT or VVL mechanisms.

For example, Honda's I-VTEC (intelligent- Variable valve Timing and Lift electronically control) which gives two or three different values of valve lifts with the help of different sizes of cam lobes and rocker arms. Similarly, AUDI developed a mechanism called AVS (Audi Valve System) also uses different sizes of cam lobes to alter the valve lift depending upon the engine speed. Other companies like BMW and SUBARU had developed new mechanisms of camshaft which uses VVT technology. MULTIAIR is latest technology developed by FIAT, which uses hydraulic pressure to operate the inlet and outlet valves.

In all types of mechanisms which uses VVL technology have two to four stages of valve lift. That means at every stage different size of cam profile will operates the valves. Here, smaller cam favors lower rpm performance, while larger cam favors higher rpm performance. Depending upon the engine speed and load conditions, the interchange between any of two stages i.e., the swap between two cam lobes occurs by an external device. These devices include oil pump, ECU (Electronic Control Unit), screw and nut mechanism, rocker arm and etc... Because of these stages the valve lift is same for a given speed range. That means, same amount of air-fuel mixture enters the combustion chamber and same amount of power produces in that given speed range. After that speed limit, swapping will happens between two cam profiles. And then, the power production rate will increases. The above technology causes a time gap between any of two stages. And this will increase engine pickup time from lower rpm to higher rpm.

This research paper presents a new type of VVL technology for inlet camshaft of the engine, called as Infinitely Variable Valve Lifting. This new mechanism helps the engine to supply more amount of air-fuel mixture for increase in every rpm. So, the power production rate increases and engine pickup time decreases. And the mechanism is an assembly of multiple parts like Camoid, Governor, Solid shaft, Hollow shaft as shown in Fig. 1 below.

Fig 1: Inlet Camshaft Full Link Model

Parts Of Camshaft Mechanism

A. Camoid

Camoids are also known as three-dimensional cams, which have two degrees of freedom i.e., rotation and translation. In the regular cams, the profile is two dimensional and have same amount of valve lift across the surface. But camoids have three dimensional profile with number of valve lifts across the surface along with the minimum and maximum amount of valve lifts. To achieve number of valve lifts and to design camoid profile, we consider a translating spherical follower which gives point contact on camoid surface.

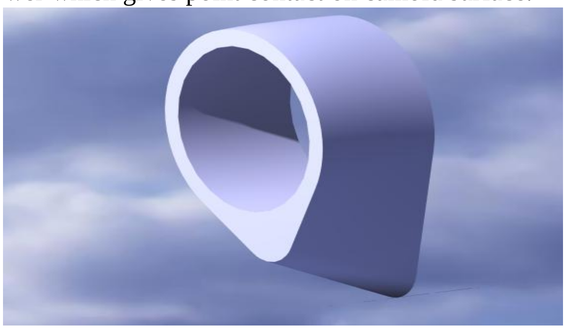

Fig 2: Three-dimensional Cam or Camoid

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Design and manufacturing of camoid is complicated. Dhnade and Chakraborthy (1975) gave a unified approach for determining the camoid profile with different types of follower mechanisms. Also, T say and H Wang (1994) derived equations for camoid profile co-ordinates depending on theory of envelope. The theory says that, the contour of a planar cam is regarded as an envelope of family of follower shape curves in different cam- follower positions when cam rotates for a complete cycle.

B. Governor

Fig 3: Governor Assembly

Governor is a self-controlling device. In the Internal combustion engines, governor is used to control the air-fuel mixture supply into combustion chamber depending upon the load and speed conditions of engine. In this mechanism, the main function of governor is to translate the camoid. And here the governor rotates about horizontal axis. It consists of two collars, one is fixed to solid shaft and the other is movable and connected to hollow shaft. The moving collar has keys which guide the collar to translate in keyways on solid shaft. These two collars connected to rotating masses with the help of rectangular bars. These bars are also useful to limit the moving collar. And a spring is placed between two collars for controlling the movement of collar.

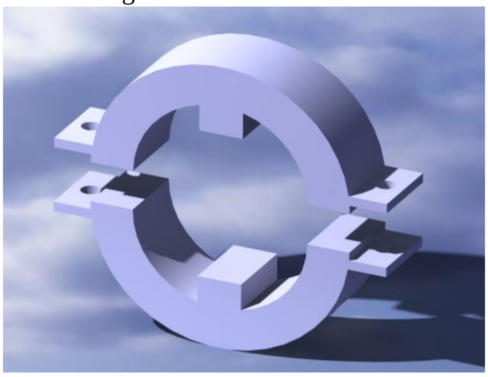

Fig 4: Moving Collar of governor

C. Solid shaft

Solid shaft is central part of total assembly. It helps the camshaft to rotate about its axis. It consists of keyways to translate the governor's collar. And the length of keyways depends on the length of camoid. It also consists of triangular grooves which helps the hollow shaft to translate.

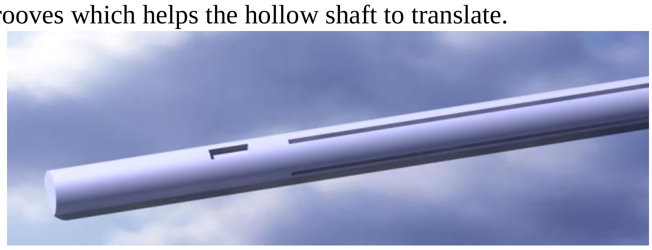

Fig 5: Soli shaft

Fig 6: Hollow shaft

D. Hallow shaft

Hollow shaft consists of camoids which are mounted on it or the shaft can be manufactured along with camoids. One end of shaft is connected to moving collar of governor and the other end is connected along with the solid shaft to engine structure. It also consists of an extended triangular grooves inside the hollow section, which helps to translate on solid shaft.

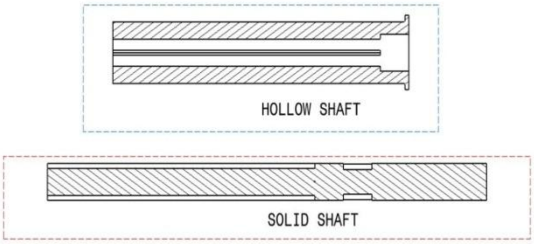

Fig 7: Sectional view of shafts

I. Assembly And Working

First, the hollow shaft and solid shaft are assembled with the help of triangular grooves. Then camoids and bearings are mounted on the hollow shaft (if they are separate). Now the governor is assembled by fixing the collar on the solid shaft. The other collar which is movable is placed in the key ways on solid shaft. And a spring is attached in between collars to control the moving collar even at rest position. Then two collars are connected by rectangular bars at pivot. And the other end of rectangular bars are connected to rotating masses. The connection should be like that, the masses are always in same plane or axis. Now the moving collar is connected to hallow shaft with the help of bolt joint. If collar slides in its keyways, it also translates the hollow shaft. Now one end of solid shaft is connected to gear drive or chain drive from crankshaft. And the other end of solid shaft and hollow shaft are connected to engine structure with the help of bearings.

If the engine starts, the crankshaft will rotate the camshaft with the help of solid shaft. And the valve opening and closing is done by camoid and spherical follower, where the valve lift is minimum. At this time. The whole assembly of camshaft is only in rotating motion. The speed of engine will increases by time, because the power production in every cycle is always higher than required even at minimum valve lift condition. When the engine reaches a certain speed, where the centrifugal force of rotating masses equals the compressive load of spring on collar, then translating motion starts to takes place in camshaft assembly.

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Then the moving collar translates in the keyways on solid shaft and also helps the hollow shaft to translate. That means the camoid also starts to move and this will change valve lift continuously. So, more amount of air-fuel mixture enters the combustion chamber and hence more power will be produced. This is continuous process from the minimum rotating speed at which collar starts to translate to the maximum rotational speed. That means, for increase in every rpm the valve lift is more than prior condition.

As described earlier, the length of keyways on solid shaft must be equal to length of camoid. If not, the camoid exceeds the translating axis of the follower. And the maximum valve lift also depends upon camoid length. That means, the rotating speed of camshaft may increase even there is no translating motion in assembly.

Now, in case of deceleration process the engine has to reach its minimum valve lift i.e., lower speed conditions as quickly as possible. Here, the spring helps the assembly to reach its minimum conditions. If the engine speed reduces, the centrifugal force by rotating masses will also reduce. This will release the compressive load of spring on collar. Hence the sleeve translates away from the fixed collar so that the minimum amount of valve lift should takes place.

II. Design Procedure For Camshaft

A. Governor

Let N be the rotational speed of governor and r is radius of rotation of masses, which are connected with rectangular bars of length l. And $e$will be the distance between axis of rotation and pivot point. As shown in Fig. 8, P is compressive load of spring on collar and θ is the angle made by bars with horizontal axis. And T is tension in rectangular bar.

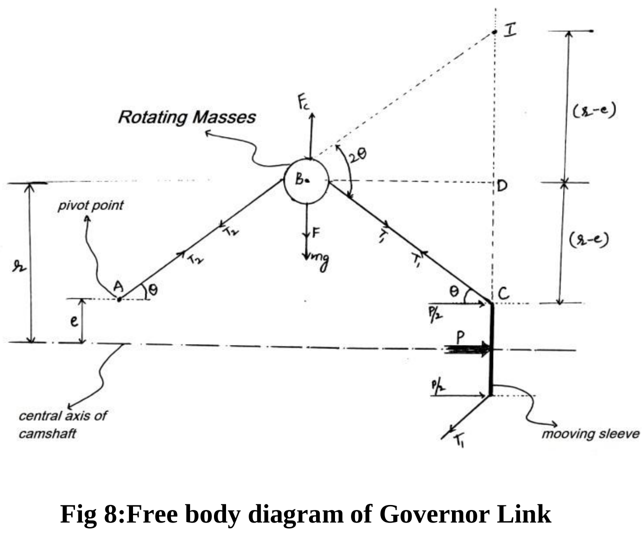

Fig 8:Free body diagram of Governor Link

Governor is always balanced by Control Force (F) which is a function of radius of rotation. Fig. 8 gives the free body diagram of governor link.

A. Taking moment about I;

$M_I = 0$

$F.\,(BD) = mg.\,(BD) + T_1.\,(IB.\sin 2\theta)$

$F = mg + T_1.\,[\frac{IB}{BD}].[\sin 2\theta]$

 $\frac{IB}{BD} = \frac{1}{\cos\theta}$

$F = mg + T_1.\,[\frac{1}{\cos\theta}].\,[2.\sin\theta\cos\theta]$

$F = mg + T_1.\,(2.\cos\theta.\tan\theta)$

Now horizontal forces acting on sleeve;

$2.\,T_1.\cos\theta = P$

 $F = mg + P.\tan\theta$…....…………………………… (1)

But control force is function of ‘r’, therefore from Fig 8;

;$\tan\theta = \frac{r-e}{\sqrt{l^2-(r-e)^2}}$

 $F = mg + P.\,[\frac{r-e}{\sqrt{l^2-(r-e)^2}}]$…………………………….. (2)

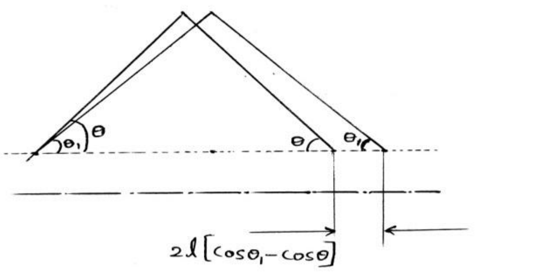

Fig 9 :Distance covered by moving collar

Where, P is the sum of initial load before sleeve moment and load after sleeve moment.

$P = P_0 + K.\,[2.l.\,(\cos\theta_1 - \cos\theta)]$

Where;

K = spring stiffness

θ_1 = angle between links and horizontal axis at radius of rotation ‘r_1’

θ = angle between links and horizontal axis at radius of rotation ‘r’

P_0 = initial spring force on sleeve

;$P_0 = P$ when $\theta = \theta_1$

And from Fig. 8; $\cos\theta = \frac{\sqrt{l^2-(r-e)^2}}{l}$

$\sin\theta = \frac{(r-e)}{l}$

 $P = P_0 + 2K.\,[\sqrt{l^2-(r1-e)^2} - \sqrt{l^2-(r-e)^2}]$ … (3)

Now we apply Equilibrium condition where the control force will be equal to centrifugal force (F_C). Also it helps to find out radius of rotation at any given rotational speed. And Fig. 10 shows characteristics of control force with respect to radius of rotation.

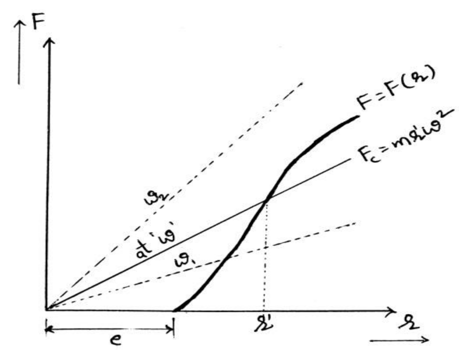

Fig10: Relation between Control force and radius of rotation

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□ Centrifugal Force (F_C) = Control Force (F)……… (4)

Since Centrifugal force; $F_C = mr\omega^2$.......................... (5)

$mr\omega^2 = mg + P.\tan\theta$

$mr\omega^2 = mg + P.[\frac{r-e}{\sqrt{l^2-(r-e)^2}}]$ [□ form eq. (3)]

$mr\omega^2 = mg +$

$[\ P_0 + 2K.\ [\ \sqrt{l^2-(\mathrm{r1}-e)^2} - \sqrt{l^2-(r-e)^2}\ ]\ ]\ [\frac{r-e}{\sqrt{l^2-(r-e)^2}}]$

B. Spring

Let us consider a spring which is compressed initially of length *X* and stiffness *K.* The spring wire diameter is *d* where spring coil diameter is *D*. And *P* be the compressive load developed in spring.

(a).Spring load:

From the equation (1), by neglecting the weight of rotating masses;

$F = P.\tan\theta$…………………………………….. (6)

By equating (5) & (6); $mr\omega^2 = P.\tan\theta$

□ Initial spring load; $P_1 = \frac{\mathrm{mr}_1\omega_1^2}{\tan\theta_1}$

And Final or maximum spring load;$P_2 = \frac{\mathrm{mr}_2\omega_2^2}{\tan\theta_2}$

(b).Stiffness of spring:

□ $K = \frac{P_2-P_1}{X_1-X_2}$

Where $(X_1 - X_2)$ = [2.l. ($\cos\theta_1$-$\cos\theta_2$)]

(c).Torsional moment of spring:

□ $M_t = f_s.\frac{\pi d^3}{16}.\frac{1}{\sigma}$

And $M_t = \frac{P_2.D}{2}$

Where; σ = stress concentration factor = $\frac{4S-1}{4S-4} + \frac{0.615}{S}$

S = spring index = $\frac{D}{d}$

(d).Deflection of spring:

□ $\delta = \frac{8.F.S^2.Z}{C.d}$

And $\delta = \frac{load}{stiffness}$

Where; Z = number of effective coils

$F = P_2 - P_1$

C = rigidity of modulus

d = diameter of spring wire

(e).Maximum deflection:

□ $\delta_{max} = \delta.[\frac{P_2}{P_2-P_1}]$

(f). Free length of spring:

□ Free length = [$(Z^1 - 1).g^1$] + [$Z.\ d$] + δ_{max}

Where; Z^1 = total number of coils

g^1 = gap between each coil of spring

(g). Initial compression of spring = $\frac{P_1}{K}$

C. Camoid

In this design of camoid, we are going to use the design approach derived by M.T. Say and H. Wang by using Theory of Envelope. Let us consider S_1 be amount of valve lift which is function of camoid displacement S_2 and angle of rotation $\emptyset_2$ of camoid.

$S_1 = S_1\ (\emptyset_2, S_2)$…………………………… (7)

Now, from theory of envelope, a family of surfaces of follower with two independent parameters of camoid for different positions can be expressed as;

$f\,(x, y, z, \emptyset_2, S_2) = 0$

And the co-ordinates of camoid can be obtained by eliminating the parameters $\emptyset$& s from equation (10) with the help of partial differentiation.

□ $\frac{\partial f}{\partial \emptyset_2} = 0$ …………………………. (8)

$\frac{\partial f}{\partial S_2} = 0$ ……………………………………. (9)

Therefore by solving Eq. (8) and (9), we have (*X, Y, Z*) co-ordinates for camoid surface with translating spherical follower as;

$$\left.\begin{aligned} X &= A\cos\emptyset_2 \pm \frac{r[B.\sin\emptyset + A.\cos\emptyset]}{\sqrt{A^2+B^2+A^2C^2}} \\ Y &= -A\sin\emptyset_2 \pm \frac{r[B.\cos\emptyset + A.\sin\emptyset]}{\sqrt{A^2+B^2+A^2C^2}} \\ Z &= -S_2 \pm \frac{r.A.C}{\sqrt{A^2+B^2+A^2C^2}} \end{aligned}\right\}\text{………… (10)}$$

Where A = (a+S_1)

B = $\frac{\partial S_1}{\partial \emptyset_2}$

And C = $\frac{\partial S_1}{\partial S_2}$

D. Valve lift

Below method of approach gives the values of valve lift, if the camoid surface is generated. From the Fig. 11 the total length of camoid is *X* and the distance between top and bottom surfaces at point of contact will be L_1at minimum end and L_2 at maximum end.

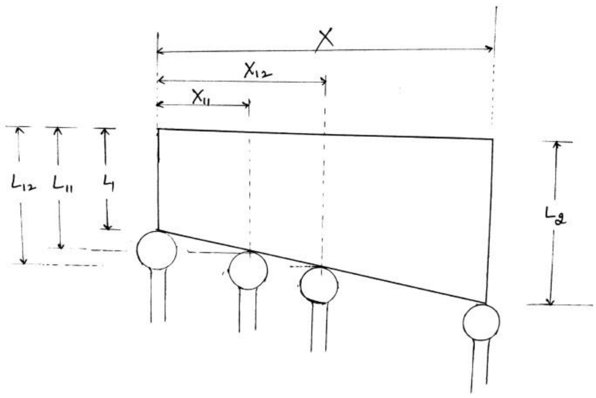

Fig 11: Point contact between camoid and follower

And L_{11}= distance b/w surfaces at next point contact

$L_{11} = L_1 + [(\frac{L2-L1}{X}).X_{11}]$……………………… (11)

Similarly; $L_{12} = L_1 + [(\frac{L2-L1}{X}).X_{12}]$……………etc.

And we have minimum valve lift i.e. 'y'

□ Second valve lift after displacement 'X_{11}';

$y_1 = y + (L_{11} - L_1)$………….. (12)

$y_2 = y + (L_{12} - L_1)$…………etc.

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III. EXAMPLE

In this section a numerical example is given, where we will find out rotational speed and valve lift for different radius of rotations. Let us consider a camshaft which has maximum speed (N_{max}) of 3000rpm. And the governor is assembled with the help of rectangular bars of length (l) 100mm at a distance (e) 50mm from the rotational axis. At a radius (r_1) of 100mm and speed (N_1) of 1000rpm, the collar starts to move towards the fixed collar. And the maximum movement of collar (l_C) will be 25mm. This moving will also help the camoid to alter the valve lift in between 12.75mm and 19.05mm. Camoid will rise in $0^0 - 85^0$ and in $95^0 - 180^0$ also Dwell in $85^0 - 95^0$.Also consider the weight of rotating masses as 9.8N and spring coli diameter as 40mm.

SOLUTION:

(A).First we have to calculate positions of rectangular bars and rotating masses

; $\tan\theta_1 = \frac{r_1 - e}{\sqrt{l^2-(r_1-e)^2}} = \frac{100-50}{\sqrt{100^2-(100-50)^2}} = 0.57$

∴ $\boldsymbol{\theta_1} = 30^0$

; l_C = [2.l. ($\cos\theta_1$-$\cos\theta_2$)]

∴ $\boldsymbol{\theta_2}$ (or) $\boldsymbol{\theta_{max}} = 42.2^0$

; $\tan\theta_2 = \frac{r_2-e}{\sqrt{l^2-(r_2-e)^2}}$

; $\tan 42.2^0 = \frac{r_2-50}{\sqrt{100^2-(r_2-50)^2}}$

∴ $\boldsymbol{r_2}$ = 117.26 mm (i.e., maximum radius of rotation)

(B).Design of spring:

 Initial spring load; $P_1 = \frac{m r_1 \omega_1^2}{\tan\theta_1}$ = 1899.02 N [$\omega = \frac{2\pi N}{60}$]

Similarly Maximum spring load;

$P_{max} = \frac{m r_{max}\, \omega_{max}^2}{\tan\theta_2}$ = 11871.8 N

 Stiffness of spring; $K = \frac{P_2-P_1}{X_1-X_2} = \frac{11871.8-1899.02}{25}$

∴$\boldsymbol{K}$ = 398.912 N/mm

And we have Deflection; $\delta = \frac{8.F.S^2.Z}{C.d}$

Since $\delta = (X_1 - X_2)$ = 25mm

Consider S= spring index= 6

; $d = \frac{D}{S} \cong 7$ mm

$F = (P_{max} - P_1)$ = 9972.78 N

No. of effective coils (Z) = $\frac{\delta.C.d}{8.F.S^2}$ = 5.11

 $\boldsymbol{Z} \cong \mathbf{5}$

 Free length of spring = [$Z.\ d$] + δ ≅60mm

 Initial compressed length = $\frac{P_1}{K}$ = 4.76 mm

(C).Now we have to calculate the rotational speed of camshaft at radius of rotation 'r_2 =117.26mm'.

From equilibrium condition i.e. from Eq. (4)

$m r_2 \omega_2{}^2$ = mg+ [P_0 + 2K. [$\sqrt{l^2-(r_1-e)^2}$ - $\sqrt{l^2-(r_2-e)^2}$]][$\frac{r_2-e}{\sqrt{l^2-(r_2-e)^2}}$]

$\boldsymbol{\omega_2}$ = 281.59rad/sec

 $\boldsymbol{N_2}$ **= 2849 rpm**

Therefore at;

r_2=100mm	N=1000rpm	l_C =0mm	y=12.75mm
r_2=102mm	N=1215rpm	l_C =2.4mm	y=13.36mm
r_2=110mm	N=2246.28rpm	l_C =13.2mm	y=16.72mm
r_2=115mm	N=2689rpm	l_C =21.22mm	y=18.18mm
r_2=117.6mm	N=2849rpm	l_C =25mm	y=19.17mm

Where; $P_0 = P_1$ =1899.02 N

r_1 = 100 mm

l_C = [2.l. ($\cos\theta_1$-$\cos\theta_2$)]

=2.l. [$\frac{\sqrt{l^2-(r_1-e)^2}}{l}$-$\frac{\sqrt{l^2-(r_2-e)^2}}{l}$]

$y_1 = y + (L_{11} - L_1)$ (from eq. (12))

$L_{11} = L_1$ + [($\frac{L2-L1}{X}$).X_{11}] (from eq. (11))

From the results, it is shown that, for different values of r we can find out the rotating speed N of camshaft.

When masses rotates at maximum radius i.e., 117.6 mm, they reach a speed of 2849 rpm with a valve lift of 19.17 mm.

But the maximum speed of camshaft is 3000 rpm. Even though the valve lift is same in speed range of 2849-3000 rpm. And the value of can be determined by using Uniform acceleration and retardation diagram (same as regular cam profiles) with the help of given values of lift, rise, and dwell.

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IV. APPENDIX

In this section an example is illustrated to design a three-dimensional camoid as in T say and H wang (1994). Let us consider, Normal distance between follower and camoid centers (a) as 10 cm and Length of camoid (l) 20 cm and Radius of spherical follower (r_S) 1 cm.

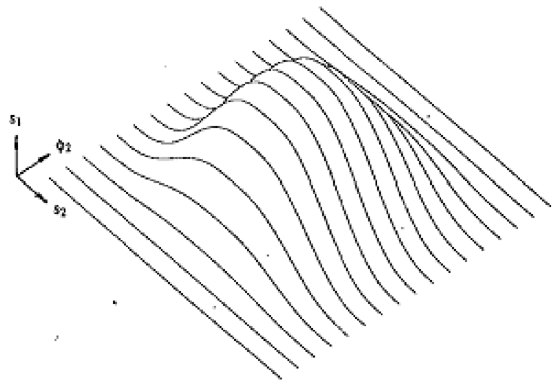

Fig12: Follower motion program

And the motion program of spherical follower is assumed to be given as,

$$S_1 = S_1\ (\emptyset_2, S_2);\ 0 \le \emptyset_2 \le 2\pi \ \&\ 0 \le S_2 \le 20$$

$$S_1 = \left[\frac{\emptyset_2}{\pi} - \frac{1}{2\pi}\ .\sin 2\emptyset_2\right]\left[\frac{S_2}{10} - \frac{1}{2\pi}\ .\sin\frac{\pi S_2}{5}\right]$$

$$= \left[\frac{\emptyset_2}{\pi} - \frac{1}{2\pi}\ .\sin 2\emptyset_2\right]\left[\frac{20-s_2}{10} - \frac{1}{2\pi}\ .\sin\frac{\pi(20-S_2)}{5}\right]$$

$$= \left[\frac{2\pi-s_2}{\pi} - \frac{1}{2\pi}\ .\sin(2(2\pi-\emptyset_2))\right]\left[\frac{S_2}{10} - \frac{1}{2\pi}\ .\sin\frac{\pi S_2}{5}\right]$$

$$= \left[\frac{2\pi-s_2}{\pi} - \frac{1}{2\pi}\ .\sin(2(2\pi-\emptyset_2))\right]\left[\frac{20-s_2}{10} - \frac{1}{2\pi}\ .\sin\frac{\pi(20-S_2)}{5}\right]$$

And the Fig. 12 shows the graphical representation of above equation. Therefore by applying analytical Eq. 10 to above motion program of follower we get the camoid profile as shown in Fig. 13.

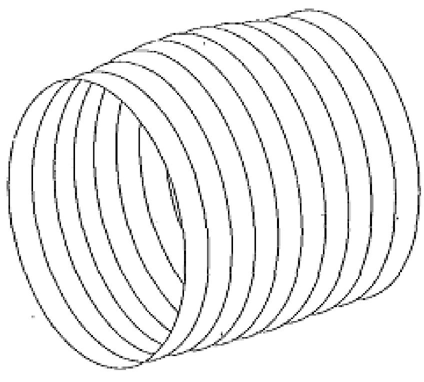

Fig 13: Camoid profile by theory of envelope for above motion program

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