* 0.429	250.005	15.501	-47.735	-245.400	250.000	-50.122	-257.670
* 0.476	255.005	15.501	-68.942	-240.306	250.000	-72.389	-252.321
* 0.524	260.005	15.501	-89.624	-233.383	250.000	-94.105	-245.052
* 0.571	265.005	15.501	-109.623	-224.684	250.000	-115.104	-235.918
* 0.619	270.005	15.501	-128.789	-214.274	250.000	-135.228	-224.988
* 0.667	275.005	15.501	-146.974	-202.234	250.000	-154.322	-212.346
* 0.714	280.005	15.501	-164.040	-188.655	250.000	-172.242	-198.088
* 0.762	285.005	15.501	-179.859	-173.640	250.000	-188.851	-182.322
* 0.810	290.005	15.501	-194.308	-157.304	250.000	-204.023	-165.169
* 0.857	295.005	15.501	-207.278	-139.770	250.000	-217.642	-146.759
* 0.905	300.005	15.501	-218.671	-121.173	250.000	-229.605	-127.232
* 0.952	305.005	15.501	-228.400	-101.653	250.000	-239.820	-106.736
* 1.000	310.005	15.501	-236.391	-81.360	250.000	-248.210	-85.428

FOOTNOTE :

1. A list of all source code is available at Journal's files and can be obtained with the publisher's permission.

2. Dwell-Rise (Return)- Dwell

3. Cam program, line no. 1430.

4. Camlaw program, line no. 2250.

5. Dwell- Rise (Return)- Dwell

6. Choosemat program, line no. 27010.

7. Dwell- Rise- Return- Dwell

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12- ESDU, The estimation of basic dimensions of disk cam with swinging followers, item no. 83008, Engineering Science Data Unit, London, May 1983.

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# 0.778	75.005	36.815	-20.439	292.854	293.566	-21.309	305.324
# 0.889	80.005	36.815	5.163	293.521	293.566	5.383	306.019
# 1.000	85.005	36.815	30.725	291.954	293.566	32.034	304.385

Segment no.= 3
################

Cam angle increment= 5
##

# 0.000	85.005	36.815	30.725	291.954	293.566	32.034	304.385
# 0.042	90.005	36.948	55.729	288.217	285.038	58.118	300.487
# 0.083	95.005	37.238	79.890	282.399	277.580	83.353	294.410
# 0.125	100.005	37.520	103.364	274.472	271.601	107.896	286.121
# 0.167	105.005	37.633	126.234	264.331	266.921	131.828	275.509
# 0.208	110.005	37.453	148.460	251.841	263.121	155.102	262.430
# 0.250	115.005	36.912	169.861	236.880	259.859	177.526	246.753
# 0.292	120.005	36.000	190.139	219.382	257.020	198.786	228.409
# 0.333	125.005	34.756	208.918	199.359	254.714	218.482	207.407
# 0.375	130.005	33.252	225.780	176.913	253.189	236.174	183.857
# 0.417	135.005	31.571	240.314	152.237	252.754	251.427	157.960
# 0.458	140.005	29.793	252.144	125.601	253.713	263.841	130.009
# 0.500	145.005	27.988	260.953	97.346	256.326	273.082	100.368
# 0.542	150.005	26.211	266.500	67.858	260.768	278.898	69.453
# 0.583	155.005	24.504	268.630	37.561	267.088	281.129	37.721
# 0.625	160.005	22.896	267.276	6.902	275.138	279.713	5.647
# 0.667	165.005	21.410	262.467	-23.653	284.471	274.689	-26.272
# 0.708	170.005	20.063	254.331	-53.617	294.232	266.202	-57.531
# 0.750	175.005	18.869	243.114	-82.484	303.069	254.516	-87.606
# 0.792	180.005	17.842	229.191	-109.735	309.146	240.025	-115.970
# 0.833	185.005	16.993	213.074	-134.864	310.380	223.259	-142.111
# 0.875	190.005	16.331	195.397	-157.419	304.975	204.865	-165.580
# 0.917	195.005	15.863	176.863	-177.068	292.181	185.557	-186.049
# 0.958	200.005	15.589	158.146	-193.664	272.951	166.013	-203.378
# 1.000	205.005	15.501	139.770	-207.278	250.000	146.759	-217.642

Segment no.= 4
################

Cam angle increment= 5
##

# 0.000	205.005	15.501	139.770	-207.278	250.000	146.759	-217.642
# 0.048	210.005	15.501	121.173	-218.671	250.000	127.231	-229.605
# 0.095	215.005	15.501	101.653	-228.400	250.000	106.736	-239.820
# 0.143	220.005	15.501	81.360	-236.391	250.000	85.428	-248.210
# 0.190	225.005	15.501	60.448	-242.582	250.000	63.470	-254.711
# 0.238	230.005	15.501	39.075	-246.927	250.000	41.029	-259.274
# 0.286	235.005	15.501	17.405	-249.393	250.000	18.276	-261.863
# 0.333	240.005	15.501	-4.397	-249.961	250.000	-4.617	-262.459
# 0.381	245.005	15.501	-26.166	-248.627	250.000	-27.474	-261.058

2	DWELL	45.00	0.00	78.997	78.997	40.005	85.005
3	CYCLOIDAL	120.00	-20.00	78.997	58.997	85.005	205.005
4	DWELL	105.00	0.00	58.997	58.997	205.005	310.005

Segment no.= 1
################

Cam angle increment= 5
##

Nor.dis.U	Cam ang.PHII (deg)	Pres.ang.ALFA (deg)	Rec.Coo. of profile XP	YP	Rad.of curva. of cam contour	Rec.coo.of whe.cen. XG	YG
# 0.000	310.005	15.501	-236.391	-81.360	250.000	-248.210	-85.428
# 0.056	315.005	15.192	-243.637	-56.708	343.620	-255.764	-59.738
# 0.111	320.005	14.433	-250.089	-25.167	430.522	-262.426	-27.181
# 0.167	325.005	13.546	-254.452	10.291	489.721	-266.907	9.234
# 0.222	330.005	12.814	-256.186	45.522	498.239	-268.685	45.348
# 0.278	335.005	12.384	-255.612	76.956	460.285	-268.096	77.597
# 0.333	340.005	12.269	-253.480	103.200	402.778	-265.902	104.600
# 0.389	345.005	12.442	-250.495	124.360	335.534	-262.816	126.473
# 0.444	350.005	12.932	-247.321	140.336	248.526	-259.505	143.127
# 0.500	355.005	13.840	-244.644	150.813	152.032	-256.659	154.260
# 0.556	0.005	15.269	-242.790	156.727	84.966	-254.601	160.819
# 0.611	5.005	17.259	-241.373	160.489	68.006	-252.939	165.230
# 0.667	10.005	19.810	-239.631	164.404	83.611	-250.905	169.803
# 0.722	15.005	22.932	-236.932	169.608	109.552	-247.857	175.682
# 0.778	20.005	26.591	-232.731	176.592	141.331	-243.237	183.366
# 0.833	25.005	30.480	-226.226	185.909	180.249	-236.227	193.409
# 0.889	30.005	33.905	-216.434	197.936	221.015	-225.827	206.184
# 0.944	35.005	36.116	-202.649	212.348	258.494	-211.325	221.346
# 1.000	40.005	36.815	-184.717	228.169	293.566	-192.582	237.884

Segment no.= 2
################

Cam angle increment= 5
##

# 0.000	40.005	36.815	-184.717	228.169	293.566	-192.582	237.884
# 0.111	45.005	36.815	-164.127	243.400	293.566	-171.116	253.764
# 0.222	50.005	36.815	-142.289	256.778	293.566	-148.348	267.712
# 0.333	55.005	36.815	-119.368	268.202	293.566	-124.451	279.622
# 0.444	60.005	36.815	-95.538	277.585	293.566	-99.606	289.405
# 0.556	65.005	36.815	-70.982	284.856	293.566	-74.004	296.985
# 0.667	70.005	36.815	-45.885	289.958	293.566	-47.839	302.305

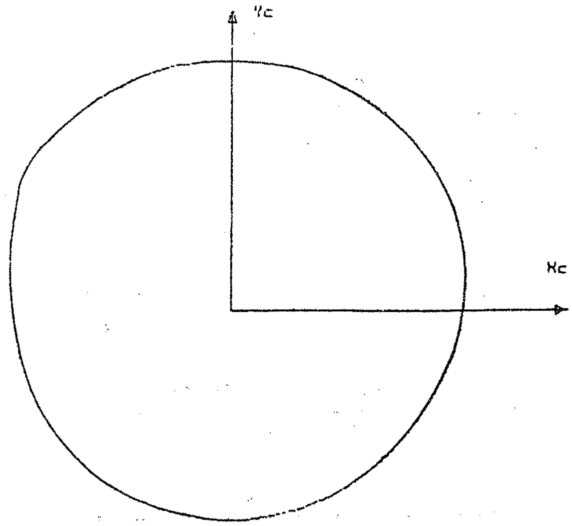

Fig. 3 : Cam profile

*************Example no.4:swinging flat faced follower*************

Swinging flat faced follower,is selected follower type

RECORD OF DESIGN:KNOWN REQUIREMENTS OF SYSTEM

Cam rotation is in an anti-clokwise sense.

QUANTITY	NOTATION	
Cam center to arm pivot	D	-350
Length of arm offset	B	-50
Initial angle from Xm-axis to follower face	SAI0	58.99728
Radius of cutter or grinding wheel	RG	12.5
Radius of reference circle(if required)	R0	250
Initial angle between radius RG and Xc-axis	ANGGO(1)	180

seg. num.	cam law for segment	cam angle of rotation for seg.	fol.ris.(pos.) or fal.(neg.) for segment	ini.hei. of fol. cen. abo. Xm-datum	fin.hei. of fol. cen. abo. Xm-datum	initial cam angle	final cam angle
j	-	PHI	SAI	SAIO	SAIF	PHIO	PHIF
1	3-H MOD.SIN.	90.00	20.00	58.997	78.997	310.005	40.005

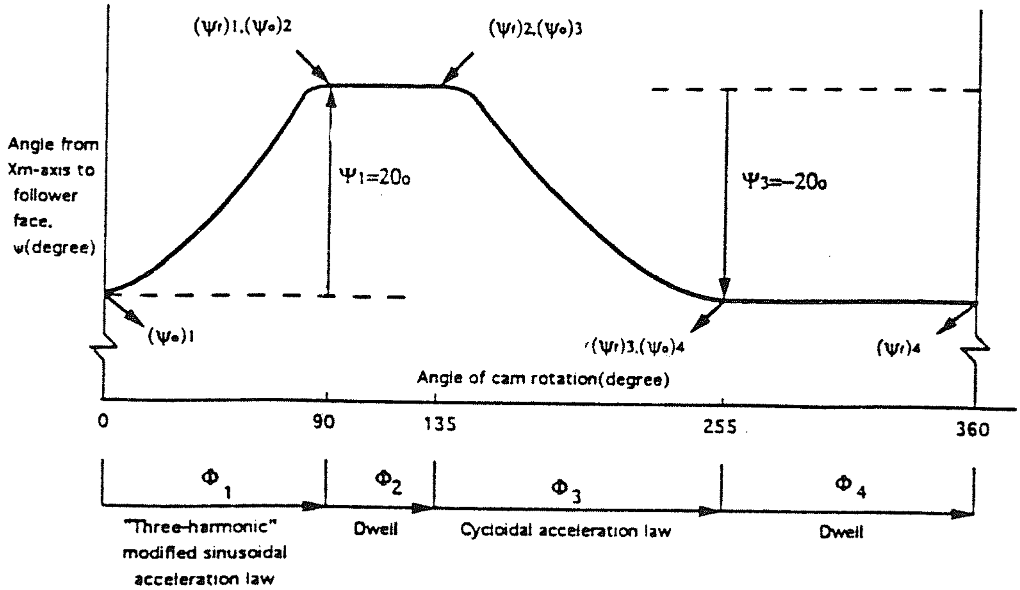

Fig. 1 : Displacement diagram

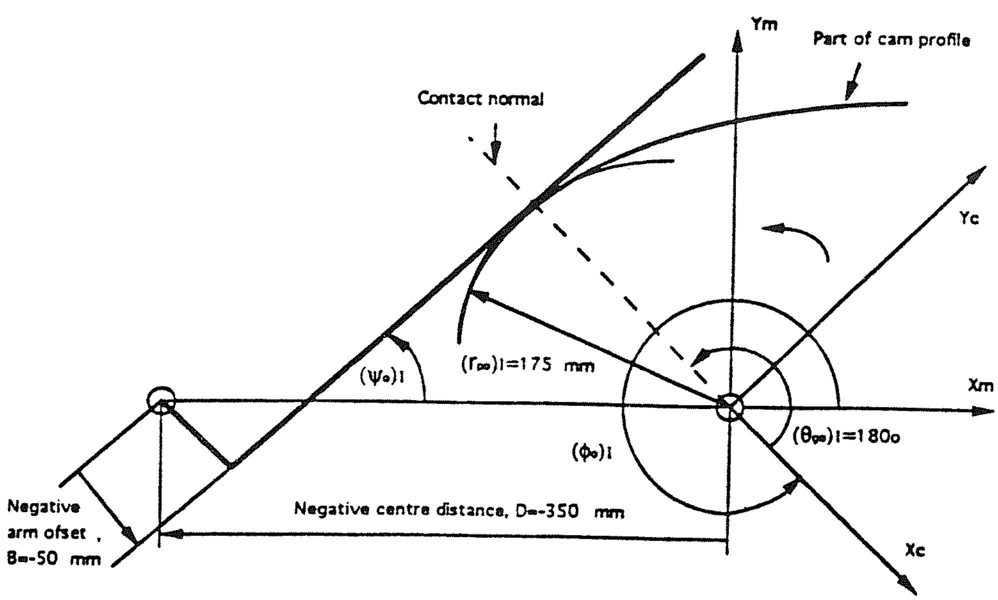

Fig. 2 : System shown at start of rise segment

and accelerations of the follower as well as the pressure angles and cam profile coordinates for each segment and for defined increments.

2. Providing coordinates on the needs of the user and the type of milling machine. (They are: milling machine, milling machine and a rotary table, specialized numerically- controlled machine and x-y numerically- controlled machine.)
3. Calling the Choosemat or. the Plotting program.

● Choosemat Program

1. providing the tables of a sorted list of common materials which are used for cams and followers.
2. Analysing of contact stresses.
3. Optimization of the cam size by rerunning the program with new data provided from contact stress analysis.
4. Calling the Plotting program.

● Plotting Program

1. Plotting all components of design. (e.g. displacement, velocity, acceleration and cam profile.)
2. Exiting from the program or reloading for a new design.

7. Illustrative Example

For a specific design environment, we intend to design a disk cam with swinging flat- faced follower. The displacement diagram, the cam law for each segment, and the situation of the system at start of rise segment are given in Figures 1 and 2, respectively. Other data are as follows:

1. Radius of cutter or grinding wheel, 12.5 mm.
2. Initial angle from X_{mi}-axis (i.e. axis fixed in machine) can be determined from Figure 2 as 58.99 degress.
3. Radius of reference circle, 250 mm.

All of the above data are input using a data file namely "cam4. dat", and are tabulated in first two table of output (page 15). We have chosen Cartesian coordinates for output. There is no sign of undercutting or interference in the design procedure. To see this, it is enough to decrease the radius of reference circle to 175 mm. Pressure angle, coordinates of cam profile, radius of curvature of cam contour, and coordinates of wheel center for specialized numerically- controlled milling machine are calculated for a complete cam cycle by increment of 5 degrees (page 16-17). Finally cam profile is sketched in Figure 3 in a X_c-Y_c axis which are fixed in cam.

8. Some Special Features

In the provided package there are some special features which made the package more useful and efficient. They are:

1. Checking data to ensure that they are in proper range.
2. Asking for correction or improvement of data after a preliminary analysis.
3. Reviewing the results and providing the chance of improvement of data after completion of the design.
4. Options of skipping some parts of the design process in order to get specific information faster than with the normal design procedure.
5. Possibility of running interactively or by inputing a data file.
6. Providing coordinates in Polar or Cartesian form depending on the needs of the user and the method of cam manufacture to be used.

9. Conclusion

● This package can be used to aid in the design of a very wide variety of cams and followers.

● Engineering intuition for estimating initial data is necessary for a successful design. Graphical methods may be used for deriving these initial data. Reference [11], [12] and [13] give some idea about initial dimensions in different lubrication conditions.

● Although obtaining an optimal design was not a goal, improving design is not too difficult when one has access to a computer code for design. Indeed, computer design is the necessary step for optimizing the design. One may interface the provided package with an external design optimizer to obtain an optimal design.

the lower the unbalance in the system.

3. Design Methods

Even if the cam and the follower surfaces are considered to correspond exactly to the theoretically desired shapes, the calculation of the actual motion-time characteristics for a physical cam and follower is a complex problem in nonlinear dynamics of elastic bodies.

Most problems such as interference, sharp corners, space considerations, wear and undercutting which are encountered in cam design, are related to the size and shape of the cam and the follower, and it becomes necessary to consider the cam profile in some detail. If extreme accuracy is necessary, we have to consider the use of special analytical methods. Although "graphical methods", which are based on kinematic inversion, are more convenient to use, but they are useful only for those cases where low accuracy is acceptable. In the package provided, we make use of the methods which are given in references [3], [4], [5] and [6], because they are applicable for more general applications.

4. Contact Stress Analysis

Some experimental graphs for calculating contact stress are given in reference [7], while some others like: "R.J. Roark" [8] and "F.Y.Chen" [9] have given formulas instead of graphs. We make use of formulas[6] which are given by "F.Y. Chen", because of ease implementation of formulas and also because of their compatibility of notation.

5. The Synthesis of Cam Motion by Blending Segments

Several methods are given in reference [10] for using portions of symmetrical DRD cam motion laws in combination with and superimposed on periods of constant velocity to satisfy special motion requirements. These special motions include a requirement for an asymmetrical acceleration characteristic, a period of constant velocity in a motion segment, motion without dwell between follower rise and return movements (DRRD[7] motion), or a precision point where the follower position is given for a particular cam angle. The methods make used of a blend factor which is derived from the standard normalised cam law formulae.

In the program the following syntheses are developed:

1. Asymmetrical DRD motion.
2. Asymmetrical DRD motion with a constant velocity period at the beginning.
3. Asymmetrical DRD motion with a constant velocity period at the middle.
4. One precision point at displacement and a constant velocity period.
5. One precision point at displacement and without a constant velocity period.
6. Symmetrical DRD motion.
7. Asymmetrical DRD motion.

6. Algorithms of Programs

The package provided contains five programs which are called consecutively. Because of the easy access to graphics, GW-BASIC and ease of programming, the BASIC language was selected to produce this package. These programs are called: Cam, Camlaw, Choosemat and Plotting.

The function of each program is as follows:

● Cam Program

1. Selecting the type of output.
2. Asking about running interactively or batch.
3. Displaying of different types of followers.
4. Selecting suitable follower.
5. Calling the Camlaw program.

● Camlaw Program

1. Providing different cam laws and providing the user a choice amongst them if designer has already decided on a specific cam law.
2. Choosing a suitable cam law for each segment if desired.
3. providing the synthesis of cam motion by blending segments.
4. Calling the Design program.

● Design program

1. Calculating the normalised displacements, velocities

It should also be obvious that manufacturing tolerance and wear will cause backlash or clearance between the unloaded face of the follower and the cam. Consequently there will be undesirable noise and vibration due to the impact occurring each time the transmitted force reverses direction and the cam contancts a different face of the follower. This problem is more acute for high- speed operation and is one of the reasons why high- speed cams generally operate with spring- loaded followers.

2.4. Torsion of a Flexible Cam-shaft

The shaft exerts torque on the cam during a portion of the cycle and the cam exerts torque on the shaft during another portion of the cycle. This varying torque requirement will cause the shaft to twist, or wind up, as the torque increases during follower rise. During this period, the cam angular velocity is slowed and so is the follower velocity. Near the end of rise the energy stored in the shaft by the windup is released, causing both the follower velocity and acceleration to rise above normal values. The resulting "kick" may produce follower jump or impact. This effect is most pronounced when heavy loads are being moved by the follower, when the follower moves at a high speed, and when the shaft is relatively flexible.

In most cases a flywheel must be employed in the cam system to alleviate the varying torque requirement. Cam- shaft windup can be prevented to a large extent by mounting the flywheel as close to the cam as possible.

2.5. Unbalancing

A disk cam produces unbalance because its mass is not symmetrical with the axis of rotation. This means that two sets of vibratory forces exist, one due to the eccentric cam mass and the other due to the reaction of the follower against the cam. Thus the smaller the size of the cam, the lower the unbalance in the system.

2.6. Cam Laws and Dynamic Performance

Analysing the dynamics of cam and follower motion is extremely complicated. In this package we make use of a "quick reference" cam law dynamic performance which is given in reference [2]. In fact we prevent serious dynamic problems by using a suitable cam law. A comparison of dynamic performance has been made for eight different cam laws using eight variables which define the dynamic performance. Based on the " quick referenc" cam law dynamic performance, these variables and their effects on dynamic performance of cams and followers are:

1. Nominal Acceleration
   Low nominal acceleration results in better dynamic performance.
2. Accelraration at Driven Load
   This variable is the same as nominal acceleration in low speed applications, and has the same effect on dynamic performand.
3. Follower Velocity
   The lower the follower velocity, the better the dynamic performance.
4. Nominal Driven Torque
   The lower the nominal driven torque, the better the dynamic response.
5. Driven Torque
   This variable has the same effect as nominal driven torque, and is very close to that in low speed applications.
6. Impact at Midpoint of Motion Segment
   The lower the jerk at midpoint of the motion segment, the lower the impact.
7. Residual Vibration
   Those cam laws which result in smaller residual vibration after one DRD[5] segment evaluate better.
8. Cam Size for Predetermined Maximum Pressure Angle
   The smaller the size of the cam, the better the dynamic performance.

Also as a general guideline in dynamics of cams and followers, we should point out that: since all real parts have mass, the lowest peak acceleration will result in the lowest peak force acting between the cam and the follower. In addition, the smaller the size of the cam, swinging flat-faced. The package provided includes four comprehensive practical worked examples to help the new users.

1. Kinematics of Cams and Followers

1.1. RDR motion

In many design problems the follower is to move outward during a certain part of a revolution of the cam, to dwell for a part of a revolution of the cam, and then to return to the initial position. This type of motion is often referred to as rise-dwell-return or simply, RDR.

1.2. Kinematic Problems in Design Procedure

During the design procedure we have to ensure that undercutting, sharp corners and interference are eliminated. In this algorithm[3], the presence of these phenomena are automatically flagged during the design procedure and associated suggestions for improvement are given. The suggestions for getting around such problems for a roller follower might be:

- use a smaller roller or
- use a larger cam or
- use a cam law which has a smooth peak acceleration characteristic.

For flat-faced rollers choosing a larger cam is suggested.

1.3. Pressure Angle

It is obvious that the forces and couples acting on the guide may lead to problems related to friction and wear and thus are undesirable. The larger the cam, the smaller the pressure angle and, therefore, the smaller the force and couple on the guide . It is usually considered good pratice to limite the pressure angle to 30 degrees, but for swinging flat-faced followers it is possible to use pressure angles as high as 60 degrees without difficulty. Large cams and large rollers also result in lower contact stresses and thus permit the use of cheaper materials or material treatments.

1.4. Sliding of the Roller

When our materials are subject to wear, we have to prevent sliding of the roller. It is obvious that with a roller follower if we do not have step changes in acceleration, we will not have any sliding. We have to use two complete cycles of motion in the accelearation part and the deceleration part; that is, at the combined point acceleration is zero. This blending task is automatically performed in the program[4] upon designer demand.

2. Dynamics of Cams and Followers

2.1. Vibration

At high speeds there is a tendency for machine members to vibrate. Such vibrations increase the operating forces and noise in a machine, and may be the principal factor limiting a machine's useful operating speed. This phenomenon has been observed in cam mechanisms. In cam driven systems the performance is strongly dependent on the choice of cam profile (i.e., type of input motion). It is this correlation between input motion and dynamic performance that makes the study and design of motions important to the design of high speed machinery.

"J.L. Wiederrich" as quoted in reference [1] has shown that it is necessary to eliminate or minimize the "cam profile's jerk" (i.e., the derivative of acceleration of input motion.), to achieve low vibration motions.

2.2. Positive Contact

One obvious disadvantage of cams is that there is no positive contact between the cam and the follower. If the outward motion of the follower is upward and the speed of rotation of the cam is slow, the dead weight of the follower may be sufficient to maintain contact between the cam and the follower at all times.However, if the rotation speed of the cam is so high that the acceleration of the follower on its downward stroke is greater that that due to gravity, an additional force must be present to maintain contact at all times. Springs are ordinarily used in this situation, even though cams may be desinged to ensure positive return of the follower.

2.3. Clearance

The Development of a Computer Program for Cam and Follower Design

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ABSTRACT

An algorithm[1] for designing constant velocity DRD[2] disk cams with four different followers has been developed. Based on the design environment, a suitable disk cam and follower is selected as well as an appropriate cam law.

A general sine- constant- consine acceleration cam law as well as eight other common cam laws are embedded in the program. In addition, properties for a small number of suitable materials have been included. Finally, contact stresses between the cam and the follower are checked in order to optimize the design.

Cam profile data is provided for four different milling machines.

Furthermore, an algorithm for synthesis of cam motion by blending segments has been developed to meet specified requirements for the follower motion.

Introduction

computers were used in machine design only in progressive industries in the 1960's while in the 1970's this became standard practice. Computer aided machine design involves more than design calculations. A large amount of data, and standard tolerances, might be tabulated in the computer, and can be accessible in the design procedure. Some of the advantages of using computers in design are:

- reduction in design time.
- providing a systemized logical design procedure.
- ease in dealing with a wide variety of design variables and constraints which are difficult to visualize using tables and formulas.

One common problem in machine design is moving or positioning components. With few exceptions, the most convenient means for imparting a specific motion to a member is by means of a cam-and-follower mechanism.

One of the attractive features of cam systems is their versatility and flexibility in the design.

Technological developments in manufacturing methods, measurements, materials and material treatments, for example, often arising in completely different applications, are finding their way into the field of cams. Among the wide variety of applications of cam systems, some industries such as machine tool engineering, automobile engineering, mining equipment, printing packaging, textile and special purpose machinery are gaining form the benefits of the cam-and-follower mechanism more than other fields.

Presently, there is no readily available comprehensive cam and follower design package. This study attempts to fill this gap for the design of disk cams with four common types of followers: translating roller, translating flat - faced, swinging roller and