

THE ESTIMATION OF BASIC DIMENSIONS OF DISC CAMS WITH SWINGING FOLLOWERS

1. NOTATION AND UNITS

In this notation, units are given for both SI and a familiar British system. All angles are measured positively in an anti-clockwise sense, and the cam is considered to rotate in an anti-clockwise direction.

		SI	British
B	offset of cranked follower arm, measured in positive sense above X_m -axis	m	in
c	distance from cam centre of rotation to design line, measured along X_m -axis	m	in
D	distance between cam centre and pivot point of swinging arm, measured in positive sense to right of cam centre of rotation	m	in
E	Young's modulus of elasticity	N/m ²	lbf/in ²
f_{zmax}	allowable compressive, or Hertzian, stress (smaller of values for cam and follower materials)	N/m ²	lbf/in ²
I_{cf}	instantaneous centre of rotation of cam with respect to follower		
I_{mc}	instantaneous centre of rotation of cam with respect to machine frame		
I_{mf}	instantaneous centre of rotation of follower with respect to machine frame		
k_j	material constant, given by $(1 - \nu_j^2)/\pi E_j$, ($j = 1, 2$)	m ² /N	in ² /lbf
L	distance between follower arm pivot and roller centre	m	in
M	load torque on follower	N m	lbf ft
P	total normal force at point of contact between roller and profile	N	lbf
R_a	prime circle radius of cam, given by $(R_r + R_b)$	m	in
R_b	reference circle radius of cam	m	in
R_r	radius of roller follower	m	in
R_{REL}	relative radius of curvature between roller follower and cam profile, given by $R_r \rho_p / (R_r + \rho_p)$		
T	parameter facilitating determination of optimum force condition (Section 3.2)		

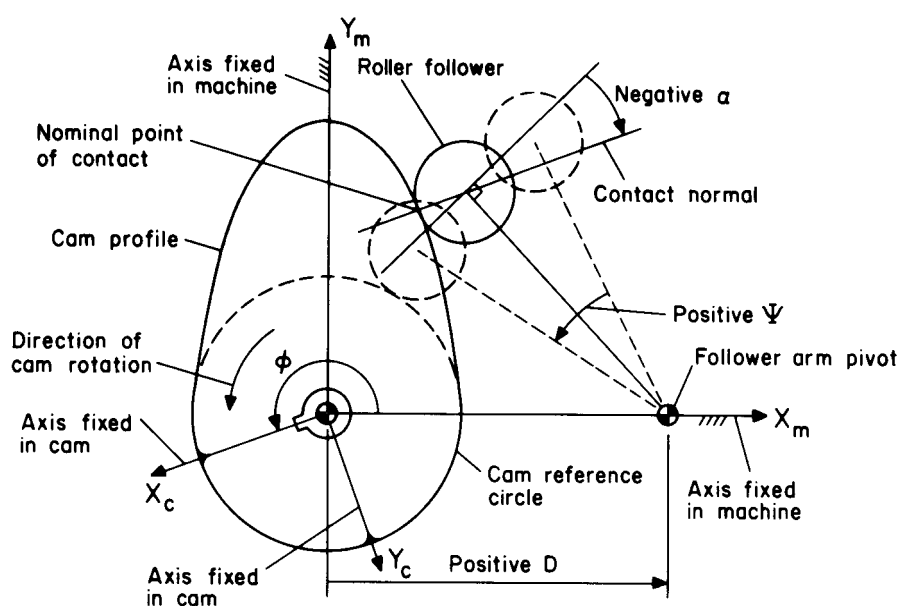
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u	normalised input displacement (cam rotation), given by $(\phi - \phi_0)/\Phi$		
u_i	value of u corresponding to maximum value of T		
W	length of cam/follower contact width, <i>i.e.</i> smaller of cam width and follower (roller or face) width	m	in
w	normalised output displacement of swinging arm, given by $(\psi - \psi_0)/\Psi$		
w'	non-dimensional output (swinging arm) velocity		
X_c, Y_c	axes fixed in cam		
X_m, Y_m	axes fixed in machine frame		
α	pressure angle ($-90^\circ \leq \alpha \leq +90^\circ$)	degree	degree
α_d	pressure angle at design position (Section 3.2)	degree	degree
α_0	pressure angle at beginning of motion segment	degree	degree
α_f	pressure angle at end of motion segment	degree	degree
ν	Poisson's ratio		
ρ	radius of curvature of pitch curve	m	in
ρ_c	critical value of profile radius of curvature (see Section 4.1)	m	in
ρ_p	radius of curvature of cam profile	m	in
σ_{co}	basic contact stress factor	N/m ²	lbf/in ²
ϕ	cam angle of rotation measured from X_m -axis	degree	degree
ϕ_0	value of Φ at start of motion segment	degree	degree
Φ_i	cam angle of rotation for one motion segment producing follower swing Ψ_i	degree	degree
Ψ_i	total angle of swing of follower arm associated with one motion segment	degree	degree
ψ	angle between X_m -axis and line joining follower pivot to roller centre	degree	degree
ψ'	non-dimensional angular velocity of follower arm, given by $w' \Psi/\Phi$		
ψ_0	value of ψ at beginning of motion segment	degree	degree
ψ_f	value of ψ at end of motion segment	degree	degree
ω	angular velocity of cam	rad/s	rad/s

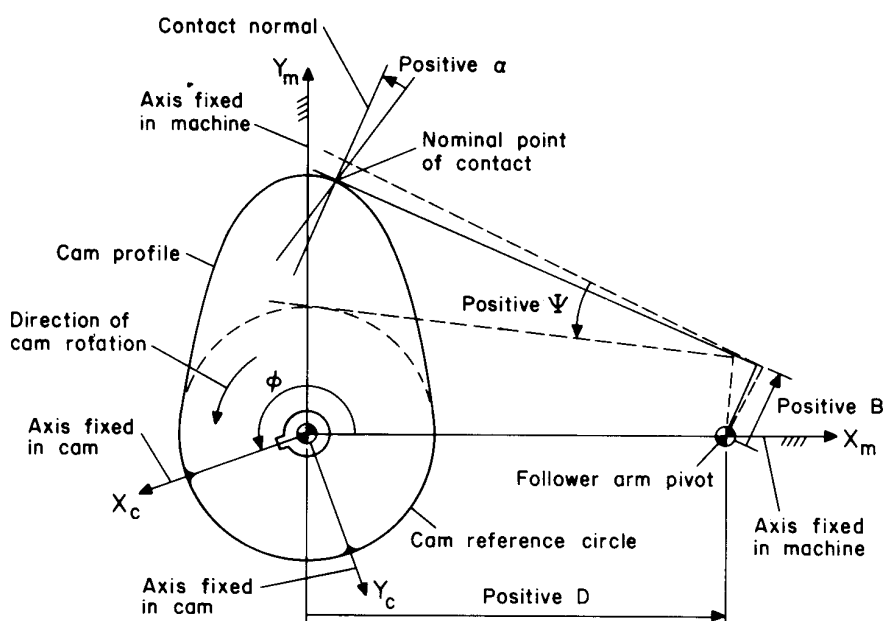
2. INTRODUCTORY NOTES

This document is one of the comprehensive series of Engineering Sciences Data Items (see References 2 to 17) that deals with the design of cam and follower systems. This Data Item is a companion to ESDU Data Item 82023 (Reference 4), which deals with the design of translating follower systems in a similar manner to the treatment of swinging follower mechanisms presented here. All commonly occurring cam/follower configurations are therefore covered by these two Data Items.

This Data Item gives guidance on the estimation of the fundamental dimensions of a disc cam driving either a swinging curved (usually roller) or a swinging flat-faced follower. The data in this Item are supplemented by additional data presented in ESDU Data Item 95001 for both swinging roller and flat-faced followers. Item 82024 provides guidance on the relative merits of swinging and translating followers and rolling or sliding contact.

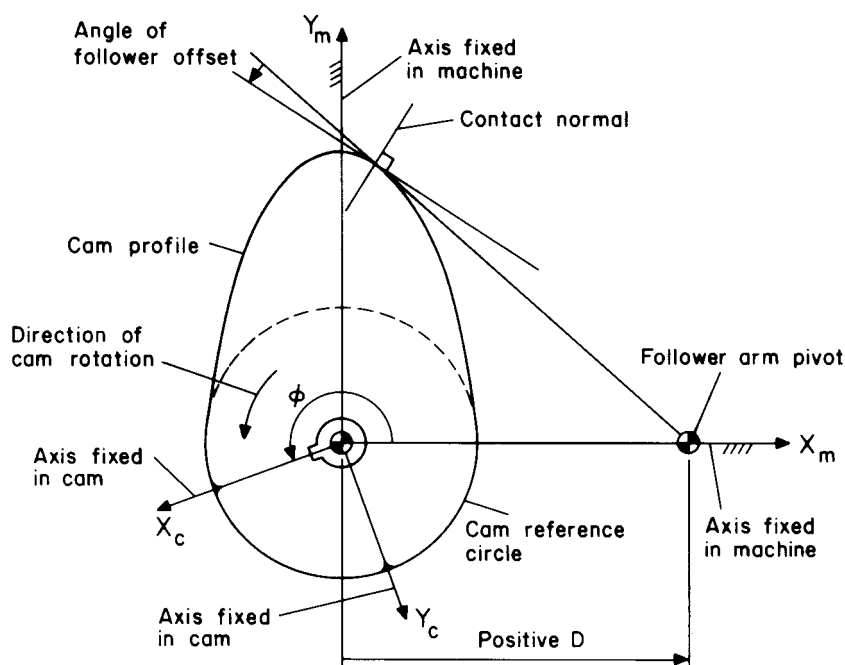


Sketch 2.1 Swinging roller follower



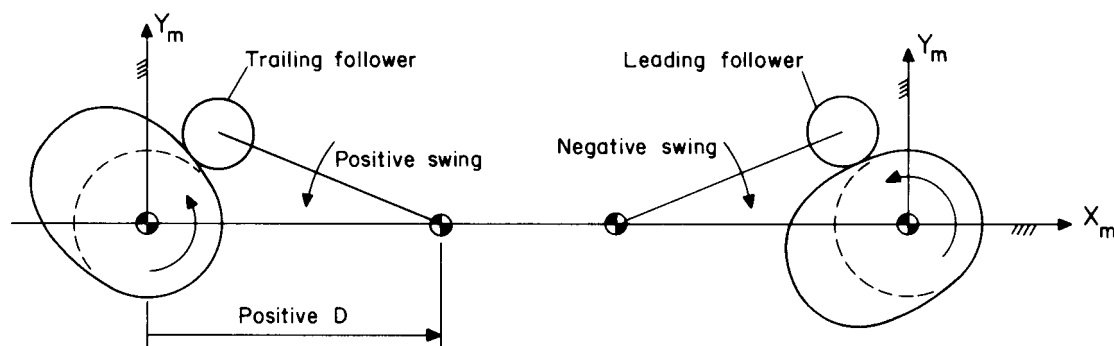
Sketch 2.2 Swinging flat-faced follower

As an alternative to the configuration shown in Sketch 2.2, a swinging flat-faced follower can be thought of as being offset by an angle rather than by a displacement, B . This concept of an offset angle is illustrated in Sketch 2.3. The notation and configuration shown in Sketch 2.2 will however be employed for the analysis in this Item.



Sketch 2.3

Swinging followers have a particular advantage over translating followers in applications where high side thrusts are present. The distribution of forces in a swinging follower mechanism allows much higher side thrusts to be accommodated with no detrimental effect (see also Item 82024). A swinging follower can be arranged in either a leading or a trailing configuration, as shown in Sketch 2.4. Under practical conditions, there is no significant difference between the performance of these two types of follower. It is most likely that the configuration of the follower, *i.e.* whether leading or trailing, will be decided by a consideration of the layout of the machine for which the cam is intended.



Sketch 2.4

Once the follower type and configuration has been selected, and the timing chart for the cam cycle has been specified (see Section 2.1), the size of the cam, *i.e.* the radius of the reference circle, R_b , will have a minimum practical value. This minimum cam size is determined by considering a number of parameters,

especially pressure angle, radius of curvature and contact stress. These parameters can be optimised by adjusting the geometry of the cam and follower.

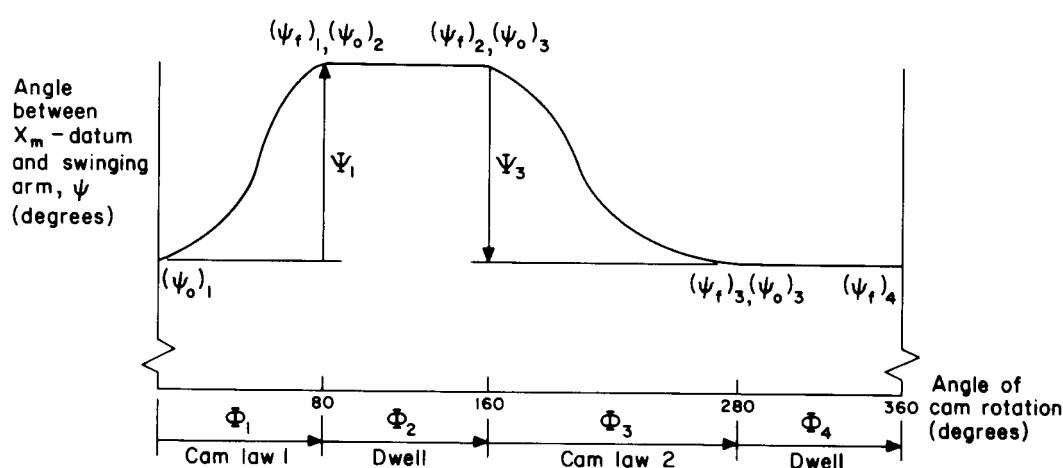
Achieving the smallest practical cam size is generally a good design strategy since it will save space, reduce weight and minimise out of balance and inertia forces. In addition, the sliding velocities will be the least possible, resulting in reduced wear with flat-faced followers and reduced roller angular velocity with roller followers.

2.1 Specification of cam requirements

Several parameters have to be defined or determined when designing a cam and follower system. Some of these can be specified provisionally so that the others can then be determined by the optimising process presented in this Data Item. The following parameters are likely to be specified as part of the machine timing requirements.

- 2.1.1 The rotational speed, or speed range, ω , of the cam shaft (note, in some cases there may exist more than one critical operating speed within a range).
- 2.1.2 The cam angle of rotation, Φ_i , for each motion segment.
- 2.1.3 The follower swing, Ψ_i , associated with each non-dwell motion segment.
- 2.1.4 The cam law for each non-dwell motion segment (see Reference 3).
- 2.1.5 The dimension D between the cam centre of rotation and the follower pivot point, specified either exactly or within maximum and minimum limits. If the latter, a starting value should be taken within the specified range; if acceptable results are not then obtained, a revised value, again within the specified range, can be tried.
- 2.1.6 The available space.
- 2.1.7 The cam shaft diameter.

Factors 2.1.2 and 2.1.3 above may be defined in the form of a timing chart, which relates the cam input rotation to the follower output swing over a complete cam cycle (see Sketch 2.5).



Sketch 2.5 Typical timing chart (DRDRD as shown)

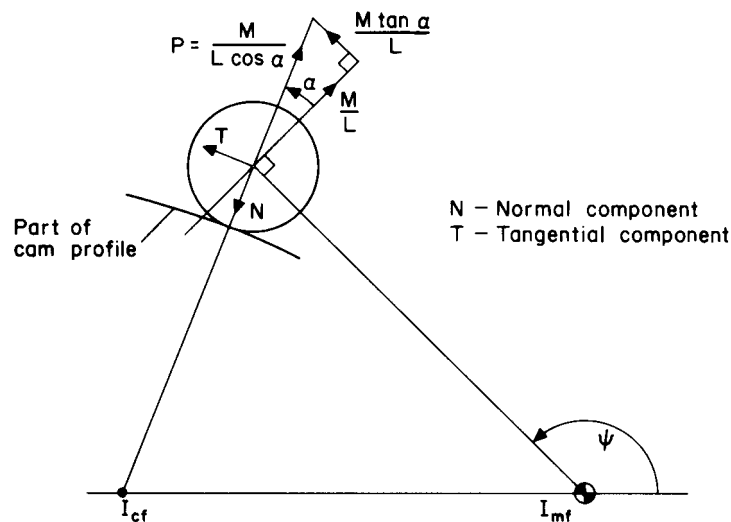
2.2 Contact Force and Contact Stress

In the design method for swinging roller followers explained in Section 3.2, the load on the follower is defined as a function of u , the non-dimensional input parameter, for each motion segment under

consideration. The forces on the roller, its bearing and the follower pivot are equal and given by Equation (2.1), where M is the load torque on the follower expressed as a function of u .

$$P = \frac{M}{L \cos \alpha} \quad (2.1)$$

No general expression for the load torque is given in this Item because of the wide range of different ways in which the driving effort can be transmitted from a swinging follower to other machine elements. Sketch 2.6 illustrates the various components of the force acting on a swinging roller follower. The load torque may be an inertia load, a friction load or an external load or any combination of these, all of which can be expressed as a function of u . If the application requires the cam to operate at several shaft speeds, the nature of the load will show which speed is critical for stress. For example, at very low operating speeds, inertia forces are likely to be negligible and spring forces dominant.



Sketch 2.6 Components of force on swinging roller follower

Care must be taken to make the follower arm stiff enough to avoid unacceptable distortion of the cam laws and alteration of the machine timing, with possible adverse dynamic effects.

The follower load torque throughout every motion segment needs to be known to enable the contact stress for both cam and follower to be checked.

2.3 Pressure Angle

The pressure angle, α , for a swinging roller follower can be defined as the angle between the normal to the line joining the arm pivot to the roller centre, and the normal to the cam profile passing through the nominal point of contact. Sketch 2.6 illustrates the way in which the pressure angle affects the components of force acting on the follower roller. For a given magnitude of load torque, a small pressure angle will give rise to a smaller force, and hence contact stress, at the contact point than will a large one.

The size of the pressure angle is only critical during the periods of the motion when the cam is active, *i.e.* when the load is acting against the motion. During inactive periods of the motion, the pressure angle is not critical, provided that the forces involved are small.

A swinging follower mechanism can be operated at larger pressure angles than equivalent translating follower mechanisms because of the superior distribution of forces. As a result, smaller cams often result from the use of a swinging follower.

The pressure angle must not be so large that the cam normal force passes through or close to the follower pivot. A general guide is not to exceed 45 degrees.

For swinging flat-faced followers, the pressure angle is the acute angle between the contact normal and the normal to the line joining the arm pivot to the contact point; this is therefore the angle of follower offset of Sketch 2.3. If there is no follower offset the pressure angle is zero throughout the cam motion; any offset will result in a pressure angle varying with cam rotation as the contact point moves along the follower face. The same limitations on pressure angle apply as on roller followers.

2.4 Cutter Interference

Cam design has to take account of the manufacturing facilities to be used. With roller followers for instance, where the cam profile can be hollow, any milling cutter or grinding wheel used to form the profile must have a radius smaller than the minimum negative (concave) radius of curvature of the cam profile, otherwise interference will occur. Item 95001 presents a calculation procedure for determining the radius of curvature of the cam profile throughout the cam cycle, and this check should be made once all other criteria have been considered.

2.5 Comparison between Design Methods used for Swinging Roller and Swinging Flat-faced Followers

The design procedure for swinging roller followers presented in Section 3.2 is based upon the theory of instantaneous centres of rotation (Derivation 18). Employing this method will enable the user to minimise loads on the follower and roller pivots. If such an approach was employed for a swinging flat-faced follower configuration, the resulting minimum load condition would require that, at the design position, the follower face should be parallel to the axis through the cam centre and the follower pivot. With flat-faced followers however, it is contact stress rather than load that is the critical parameter to control, and for this reason a different method, which optimises radius of curvature and hence contact stress, is employed in Section 4.

3. DISC CAMS WITH SWINGING ROLLER FOLLOWERS

The first estimate of the basic dimensions of a swinging roller mechanism may be made by employing the simple graphical approach presented in Section 3.2. The design will then be “tuned” by examining in detail several of the mechanism characteristics. The graphical method adopted for use in this Item will result in a condition of minimum contact force for a given loading cycle. The results of this graphical approach provide a suitable basis for the subsequent determination of cam and roller size.

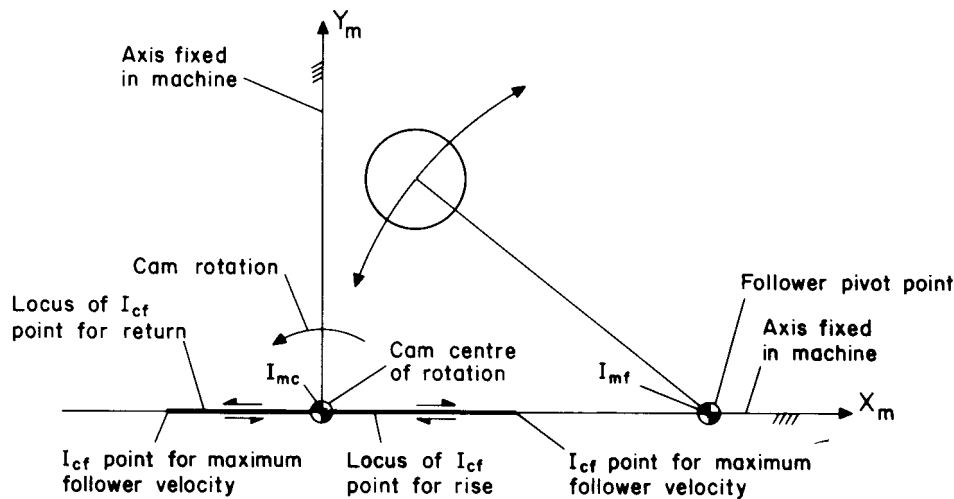
3.1 Instantaneous Centres of Rotation

The design in method presented in Section 3.2 is based upon the theory of instantaneous centres of rotation (Derivation 18). A different method is recommended for the design of swinging flat-faced follower mechanisms (Sections 2.5 and 4).

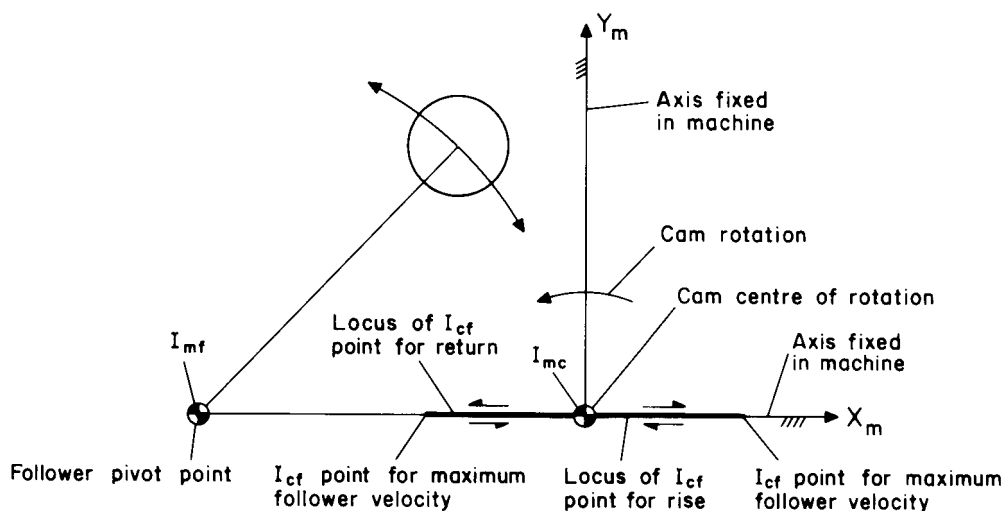
In a cam and follower mechanism, there is a point known as the instantaneous centre of rotation of the cam with respect to the follower; this point is designated the I_{cf} -point. At the beginning and end of a dwell-rise-dwell or dwell-return-dwell motion segment, the I_{cf} -point will coincide with the cam centre of rotation. A line drawn from the I_{cf} -point to the roller centre will be normal to the pitch profile and will therefore pass through the point of contact of the roller with the cam. Assuming that the X_m -axis

(Sketch 3.1) has been drawn through both the cam centre of rotation and the follower pivot then one of the following will also apply.

- 3.1.1 If the cam is rotating anti-clockwise and the follower is rising during the motion segment (Sketch 3.1), the I_{cf} -point will move along the X_m -axis to the right of the cam centre of rotation until an extreme position corresponding to maximum follower velocity is reached. The I_{cf} -point will then move back to the cam centre along the same line as the motion segment is completed.
- 3.1.2 If the follower is falling and the cam is rotating anti-clockwise, then the I_{cf} -point will have a locus as in 3.1.1 but to the left of the cam centre of rotation.
- 3.1.3 If the follower pivot is to the left of the cam centre of rotation and the cam still rotates in an anti-clockwise sense, then 3.1.1 and 3.1.2 will still apply, but will be reversed from the point of view of follower direction (Sketch 3.2).



Sketch 3.1 Loci of I_{cf} point for positive D (rise and return)



Sketch 3.2 Loci of I_{cf} point for negative D (rise and return)

3.2 Cam Size Estimation

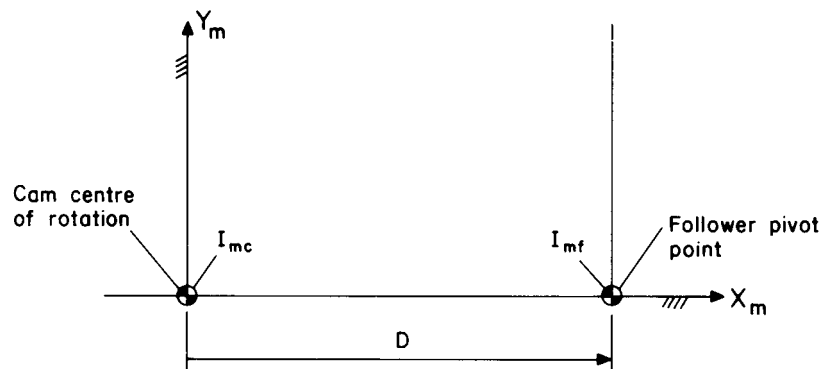
The layout of the cam and follower is determined as follows.

- 3.2.1 The first stage in the design is to examine the load acting on the follower and determine the value of the parameter, T , throughout each active region of the cam motion. Equation (3.1) gives the expression for T ,

$$T = \frac{M}{|D|} (1 - \psi'). \quad (3.1)$$

Section 2.2 discusses the determination of the magnitude of the load torque, M . ψ' is determined from the relevant cam law (see Reference 3). Once the value of T has been determined throughout each active motion segment, a graph should be plotted showing the variation of T with u to determine the value of u_i corresponding to the maximum value of T . Determine the value of u_i for each active motion segment.

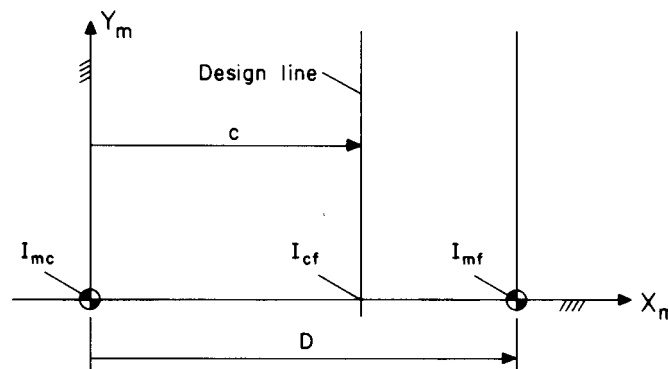
- 3.2.2 Lay out to scale the cam and follower axes, making the X_m -axis pass through both the cam centre of rotation and the follower pivot point (Sketch 3.3).



Sketch 3.3

- 3.2.3 Locate the position of the I_{cf} -point corresponding to the value of u_i (Sketch 3.4),

$$c = \left[\frac{D}{1 - \frac{1}{w'(\frac{\psi}{\Phi})}} \right]_{u = u_i} \quad (3.2)$$

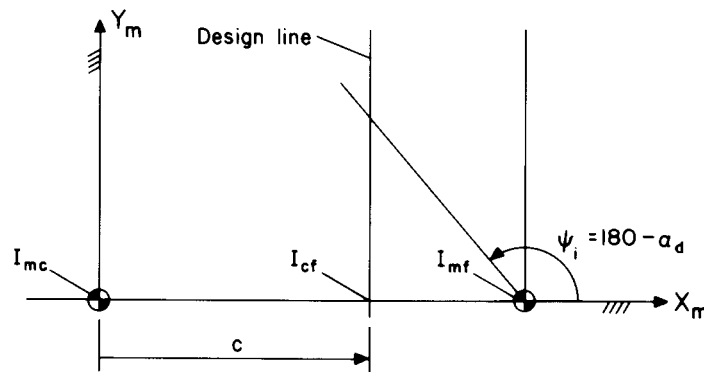


Sketch 3.4

- 3.2.4 Draw a line perpendicular to the X_m -axis passing through the I_{cf} -point. This line is termed the design line.
- 3.2.5 Using the guidance in Section 2.3 select an allowable pressure angle at the design position, α_d , for the first motion segment under consideration, and hence determine the value of ψ_i (the angle from the X_m -axis to the follower arm) at input displacement u_i .

$$\left. \begin{aligned} \psi_i &= 180 - \alpha_d, \text{ for positive } D, \\ \psi_i &= \alpha_d, \text{ for negative } D. \end{aligned} \right\} \quad (3.3)$$

Then draw in the follower arm at this position (Sketch 3.5).

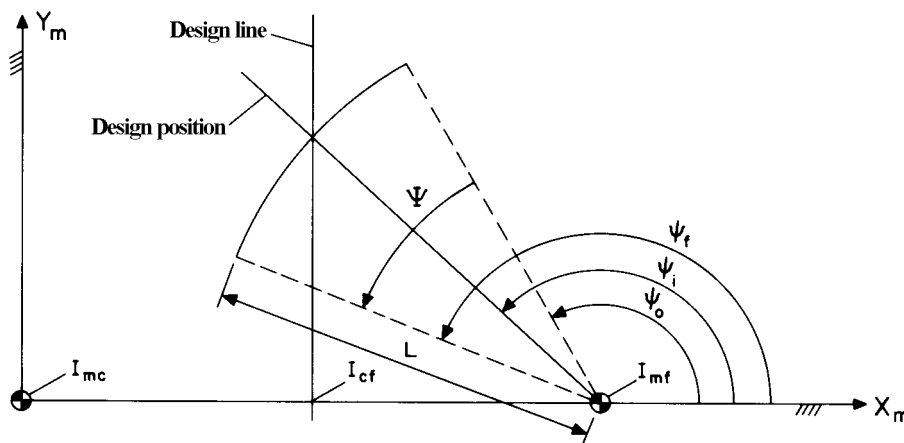


Sketch 3.5

- 3.2.6 Putting $\psi = \psi_i$ and $w = w_i$ into ,
- $$\psi = \psi_0 - w\Psi \quad (3.4)$$

yields the value of ψ_0 . The value of ψ_f is then readily obtained.

Draw the position of the follower at the beginning and end of the motion segment (Sketch 3.6).



Sketch 3.6