



ESDU 91026

Issued December 1991
With Amendments A to E
July 2010

Analysis of cam roller followers

Associated software: ESDUpac A9126
VIEWpac W9126

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The Mechanisms Committee, which first met in 1975, is continuing the work that previously was the responsibility of the former Gearing and Mechanisms Committee, which first met in November 1964. Like the earlier Committee it has the co-operation of many engineers and scientists in industry, research establishments and universities from whom much assistance and information is being received.

ANALYSIS OF CAM ROLLER FOLLOWERS

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ANALYSIS OF CAM ROLLER FOLLOWERS

1. NOTATION AND UNITS

The units used in this Item are predominantly SI but the user is strongly recommended to check the units given in bearing manufacturers' catalogues and to make any necessary conversions. The programs use units in common usage which differ from those of SI.

		Units
a_1	life adjustment factor	—
a_2	material correction factor	—
a_3	lubrication correction factor	—
C	dynamic load rating of roller	N
C_0	static load rating of roller	N
D	distance between cam rotational centre and follower pivot	m
d_1, d_2, d_3, d_4	lengths used in derivation of roller angular velocity (see Sketches 3.2 and 3.4)	m
E	follower offset	m
F_E	external force acting on moving follower (positive to increase contact force)	N
F_{EO}	external force acting on stationary follower (positive to increase contact force)	N
F_{xi}, F_{yi}	components of resultant of external forces acting on part i in x - and y -directions, respectively	N
f_w	service factor	—
H	follower lift corresponding to one segment	m
I_f	mass moment of inertia of swinging follower system about follower pivot	kg m ²
k	constant of value +1 or -1 (see Section 3.2.1)	—
k_{SC}	compression spring stiffness	N/m
k_{ST}	torsion spring stiffness	N m/rad
L	distance between follower pivot and centre	m
$L_{10(h)}$	basic life rating	h

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$L_{10(h)a}$	modified life rating	h
m_f	lumped mass of translating follower system at roller pivot	kg
N_r	angular velocity of roller	rev/min
n	number of calculation points	—
P	constant load on roller	N
P_C	resultant contact force	N
P_{CO}	contact force exerted on stationary roller	N
P_{ij}	force exerted by part i on part j	N
P_S	compression spring force	N
P_{SP}	compression spring preload	N
P_{xij}, P_{yij}	components of joint force P_{ij} in x - and y -directions, respectively	N
R_b	reference circle radius of cam	m
R_r	radius of roller	m
s_0	static safety factor	—
T_E	external moment acting on moving follower (positive to increase contact force)	N m
T_{EO}	external moment acting on stationary follower (positive to increase contact force)	N m
T_S	torsion spring moment	N m
T_{SP}	torsion spring preload	N m
T_{ij}	moment exerted by part i on part j	N m
w	normalised follower displacement	—
w'	non-dimensional follower velocity	—
w''	non-dimensional follower acceleration	—
X_m, Y_m	axes fixed in machine frame	—
y	translational displacement of follower	m
$\dot{y}$	translational velocity of follower	m/s
$\ddot{y}$	translational acceleration of follower	m/s ²

y_b	height of centre of roller above X_m -axis at inner dwell	m
y_m	instantaneous height of centre of roller above X_m -axis	m
y_0	height of centre of roller above X_m -axis at beginning of a segment	m
z_1	force factor	—
z_2	preload factor	—
α	pressure angle	degree
γ	angle between Y_m -axis and line joining cam and roller centres	rad
Φ	cam angle of rotation corresponding to one segment	rad
ϕ	cam angle of rotation (from start of Segment 1)	degree
Ψ	follower swing corresponding to one segment	rad
$\dot{\psi}$	rotational displacement of follower	rad
$\dot{\psi}$	rotational velocity of follower	rad/s
$\ddot{\psi}$	rotational acceleration of follower	rad/s ²
ψ_b	angle between X_m -axis and line joining follower pivot to roller centre at inner dwell	rad
ψ_m	instantaneous angle between X_m -axis and line joining follower pivot to roller centre	rad
ψ_0	angle between X_m -axis and line joining follower pivot to roller centre at beginning of a segment	rad
ω_c	angular velocity of cam	rad/s
ω_r	angular velocity of roller relative to follower arm	rad/s

Subscripts

e	refers to end value of external load
$equiv$	refers to equivalent value
i	refers to value at a particular calculation point
max	refers to maximum value
min	refers to minimum value
q	refers to value when separating force is a maximum
s	refers to start value of external load

2. INTRODUCTION

This Data Item presents a procedure for assessing the suitability of a user-specified commercial needle-bearing roller follower driven by a disc cam. The choice of an appropriate roller follower is an important step in the design of cam systems used in many industrial machines and the use of the procedure presented in this Item, in conjunction with a suitable manufacturer's catalogue, will ease this choice. A decrease in design time can be achieved together with improvements in life, load/speed capacity and hence machine reliability.

Cam roller followers are subject to conditions of continuously varying angular velocity and load during operation. To assess the performance of a given roller it is necessary to determine, at a series of cam angular positions, the values of load and velocity and then, using the method recommended in manufacturers' catalogues, to convert them to equivalent load (see Equation (3.46)) and equivalent velocity (see Equation (3.45)). This analysis is carried out by a computer program (see Section 7) supplied with this Data Item. The program determines the estimated life, the maximum contact force at the cam/roller interface, the maximum load on the roller stud or spindle and the maximum angular velocity of the roller. These values are then used to assess the suitability of the chosen roller.

2.1 Assumptions

The procedure presented in this Item is applicable to both translating and swinging roller followers driven by disc cams with spring force closure. It is assumed that:

- (a) forces acting on the roller are due only to follower system inertia, action of the return spring and externally applied load acting on the follower system (friction in the follower system and gravitational force may be modelled as part of the external load if required),
 - (b) friction force at the roller pivot is negligible,
 - (c) slip does not occur between the cam and roller[†],
 - (d) the cam and roller are designed and manufactured so that the roller can accurately follow the cam profile at all times (undercutting does not occur),
 - (e) the cam and roller remain in contact throughout the operating cycle (see Section 3.4)
- and (f) the follower system is sufficiently stiff to be considered rigid (this assumption particularly affects the jerk and will provide unrealistic values).

This list of assumptions is not comprehensive; further specific assumptions are described in Section 3.

2.2 Limitations and Precautions

The use of cam laws that exhibit infinite jerk (see ESDU 82006, Reference 5) is excluded because such motion causes shock loads on the roller and possibly roller slip. The procedure in this present Item assumes that the dominant failure mode for cam roller followers is fatigue of the rolling elements. This is true for perfectly aligned, well lubricated bearings at normal temperatures operating in a clean environment. In practice the life of a roller may be limited by wear or corrosion. Careful choice of cam material, heat

[†] Assumption (c) is obviously questionable. Slip sometimes occurs between the roller and the cam especially at high speed but this is very difficult to predict. Use of the "no slip" assumption results in a good first-order approximation of roller speed and roller life under most operating conditions.

treatment and surface finish using ESDU 94017 (Reference 13), will minimise the possibility of premature failure by these modes. To evaluate the roller fully it is necessary to check that contact stress at the cam surface is acceptable (see ESDU 93001, Reference 12) and that curvature and pressure angle constraints (see ESDU 95001 and ESDU 00014, References 14 and 16, respectively) are not infringed.

2.3 Use of This Data Item

The theoretical background is described in Section 3 of this Data Item. It is recommended that the necessary calculation is carried out using the accompanying computer program described in Section 7. A guide to the choice of values for the inputs to this program is presented in Section 4. The examples in Section 6 provide valuable hints on the effective use of the program.

Analysis of roller performance may be useful:

- (a) during a feasibility study (see ESDU 94017, Reference 13),
 - (b) as part of detail design (see ESDU 94017, Reference 13),
 - (c) to investigate the cause of roller failure,
- or
- (d) to upgrade mechanism performance by retro-fitting.

It is likely that the nature of the information available in the above cases will vary. For instance during a feasibility study the mass of the follower and the externally applied forces may not be known exactly. The analytical procedure is applicable when uncertainty surrounds values of inputs but the accuracy of the results is obviously dependent on the quality of the input information. It is recommended that the program be used repeatedly during the design process as increasingly accurate information becomes available. It may be necessary to run the program with different rollers to identify the most suitable roller (see Section 4.2.1). A bearing manufacturer's current catalogue is essential to obtain values for inputs to the program and to assess the program output.

3. THEORETICAL BACKGROUND

This section presents details of the mathematical model on which the calculation procedure is based. It is strongly recommended that the calculations are carried out using the computer program described in Section 7. The equations used to evaluate contact force and roller angular velocity in this program are derived in Section 3.1, for the case of a translating follower, and in Section 3.2 for that of a swinging follower. Section 3.3 introduces the expression for roller life and Section 3.4 presents the formulae used for initial design of the spring. Recommended values of the quantities needed as inputs to the computer program are given in Section 4.1. The convention adopted for forces and moments in this Item is such that P_{x23} is the component of force in the x-direction exerted by Part 2 on Part 3. Thus

$$P_{x23} = -P_{x32}.$$

The magnitude of the force exerted by Part 2 on Part 3 is given by

$$P_{23} = \left(P_{x23}^2 + P_{y23}^2 \right)^{1/2}.$$

Similarly, T_{45} is the moment (positive in the anticlockwise sense) exerted by Part 4 on Part 5. Thus

$$T_{45} = -T_{54}.$$

3.1 Translating Roller Follower

Sketch 3.1(a) shows a typical sub-system composed of a disc cam driving an offset translating roller follower. A compression spring is used to provide force closure. The convention used is shown in Sketch 3.1(b). Diagrams showing the forces acting on each part of the assembly are shown in Sketch 3.1(c).

3.1.1 Derivation of contact force

It is required to derive an expression for the contact force, P_C , between the roller and the cam due to the action of the spring, the inertia of the follower system and the external force acting on the follower. The follower is constrained to move only in the y-direction and it is assumed that friction force, P_{y14} , between the follower and the guide is zero. However, in practice, friction can be included by considering it as part of the external force, F_{y4} . The sum of the components of force acting on the follower in the y-direction is equal to the product of the mass of the follower (see Section 4.1.1) and the acceleration of its centre of mass in the y-direction. Thus for equilibrium of the follower

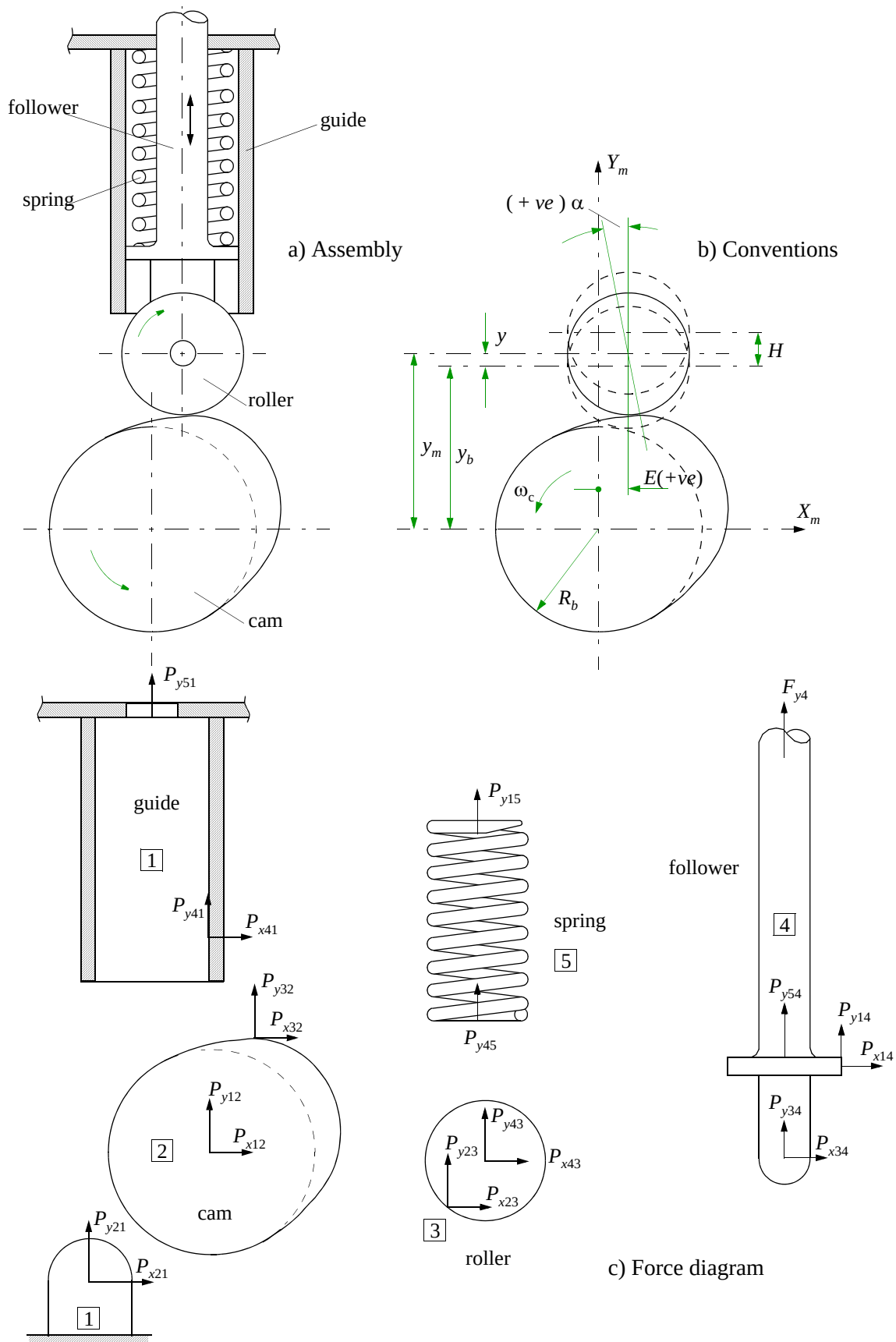
$$P_{y34} + P_{y54} + F_{y4} = m_f \ddot{y}, \quad (3.1)$$

where P_{y34} is the force exerted by the roller on the follower,

P_{y54} is the force exerted by the spring on the follower,

F_{y4} is the external force exerted on the follower

and m_f is the mass of the follower system.



Sketch 3.1 Cam and follower system

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The instantaneous motion (displacement, y , velocity, $\dot{y}$ and acceleration, $\ddot{y}$) of the roller centre during a rise or return segment is required. These quantities are given by

$$y = wH, \quad (3.2a)$$

$$y_m = y_0 + wH, \quad (3.2b)$$

$$\dot{y} = \frac{w' \omega_c H}{\Phi} \quad (3.3)$$

and
$$\ddot{y} = \frac{w'' \omega_c^2 H}{\Phi^2} \quad (3.4)$$

where y_0 is the height of the roller centre above the X_m -axis at the start of the motion segment (rise or return),

H is the total follower lift corresponding to one motion segment (in this Data Item it is taken to be a positive scalar quantity for both rise and return motion segments),

Φ is the total cam angle corresponding to one motion segment (note that units are radians),

ω_c is the angular velocity of the cam

and w , w' and w'' are the dimensionless follower displacement, velocity and acceleration, respectively (expressions for these parameters corresponding to commonly-used cam laws are given in Data Item 82006, Reference 5).

Note that:

(a) w and w' are positive during a rise motion and negative during a return motion

and (b) w'' changes sign during every motion segment (it is positive during the first part of a rise segment and negative during the first part of a return segment).

The spring force is composed of the preload, P_{SP} , and the product of the spring rate, k_{SC} , and the follower displacement from inner dwell ($y_m - y_b$) and is given by

$$P_S = -P_{y54} = P_{SP} + k_{SC} (y_m - y_b), \quad (3.5)$$

where
$$y_b = \left((R_b + R_r)^2 - E^2 \right)^{1/2}. \quad (3.6)$$

It is assumed that the mass moment of inertia of the roller is negligible which implies that the torque to accelerate the outer race and the rolling elements is negligible, and so only two forces are considered to act on the roller. These forces must be of equal magnitude and opposite in direction. Therefore, for equilibrium of the roller

$$P_{23} + P_{43} = 0 \quad (3.7)$$

and the force exerted on the roller by the cam is given by

$$\begin{aligned} P_{23} &= -P_{43} \\ &= \frac{P_{y34}}{\cos \alpha}, \end{aligned} \quad (3.8)$$

where α is the pressure angle given by

$$\alpha = \tan^{-1} \left[\frac{\frac{(w'H)}{\Phi} - E}{y_m} \right]. \quad (3.9)$$

Note that for a rise segment w' is positive and for a return segment w' is negative (see ESDU ME2, Reference 4).

Substituting Equation (3.1) into Equation (3.8) gives

$$P_{23} = \frac{[-P_{y54} - F_{y4} + m_f \ddot{y}]}{\cos \alpha}. \quad (3.10)$$

The external force, F_E , and the spring force, P_S , are considered as positive (by convention) when they act to increase the contact force on the cam. So

$$F_E = -F_{y4} \quad (3.11)$$

$$\text{and} \quad P_S = -P_{y54}. \quad (3.12)$$

A service factor, f_w (see Section 4.1.4), can be used to account for unknown loads due to shock and vibration and so the contact force is defined as

$$P_C = P_{23} f_w. \quad (3.13)$$

Thus the expression for contact force, as used in the program, is obtained by first substituting Equations (3.10) in (3.13) and then substituting Equation (3.5) and (3.11) into the result to give

$$P_C = \frac{\left([P_{SP} + k_{SC}(y_m - y_b)] + F_E + m_f \ddot{y} \right) f_w}{\cos \alpha}. \quad (3.14)$$

To evaluate the performance of the roller it is also necessary to calculate the static contact force. This is the force that could be exerted on a stationary roller and so does not include effects due to inertia, shock loading or vibration. It does include a static safety factor, s_0 (see Section 4.1.4), and is given by

$$P_{CO} = \frac{\left((P_{SP} + k_{SC}(y_m - y_b)) + F_{EO} \right) s_0}{\cos \alpha}, \quad (3.15)$$

where F_{EO} is the external force that acts on the roller when the cam is stationary, *i.e.* excluding friction and any other dynamic effects.

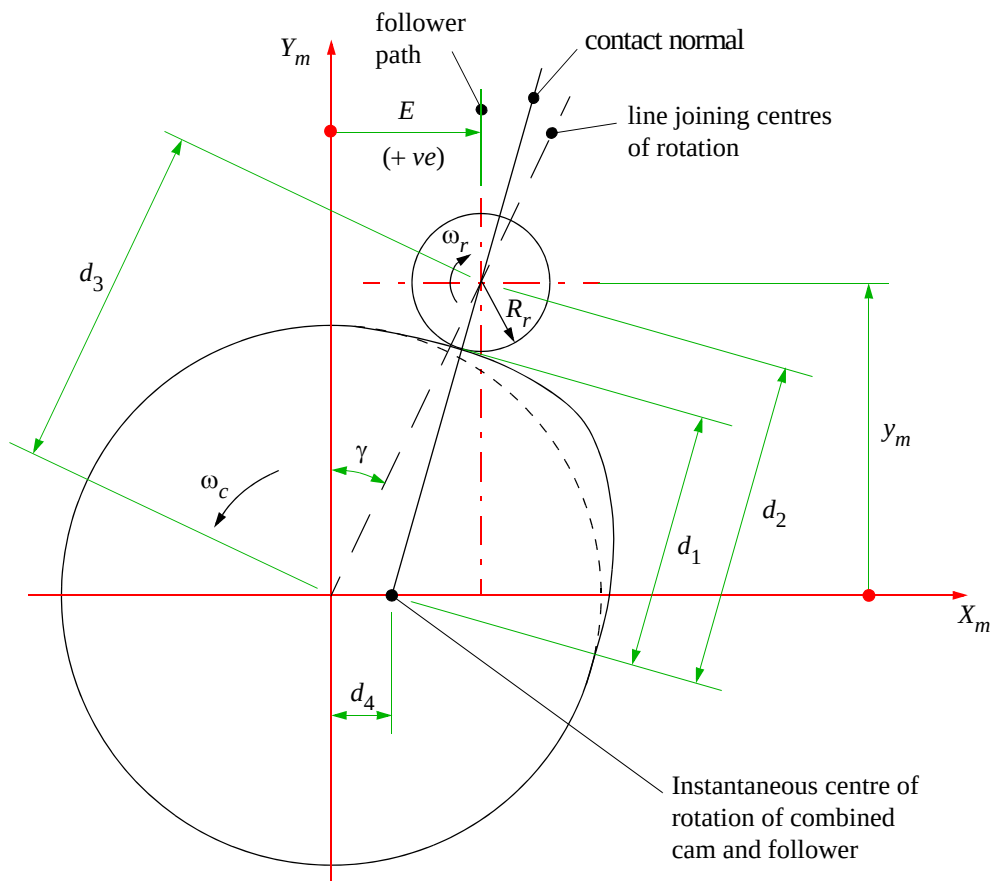
Both static and dynamic contact forces are calculated in the program at intervals of one degree of cam angle during one full rotation of the cam.

3.1.2 Derivation of roller angular velocity

The angular velocity of the roller must be expressed in terms of the known quantities. Assuming slip does not occur at the cam/roller interface the tangential velocity of the cam at the contact point is equal to the tangential velocity of the roller. Thus, from Sketch 3.2,

$$\omega_c d_1 = -\omega_r R_r \quad (3.16)$$

and so
$$\omega_r = -\frac{\omega_c d_1}{R_r} \quad (3.17)$$



Sketch 3.2 Angular velocity of roller for translating follower

The cam angular velocity, ω_c , and the roller radius, R_r , are known constants. However, the length, d_1 , between the instantaneous centre of rotation (for a definition see ESDU ME2, Reference 4) of the combined cam and roller and the contact point is a variable. This length (shown in Sketch 3.2) may be expressed as

$$d_1 = d_2 - R_r \quad (3.18)$$

and by the cosine rule

$$d_2 = \left(d_3^2 + d_4^2 - 2d_3 d_4 \cos((\pi/2) - \gamma) \right)^{1/2}, \quad (3.19)$$

where $d_3 = \left(y_m^2 + E^2 \right)^{1/2}$ by Pythagoras (3.20)

and, using Kennedy's theorem of instantaneous centres,

$$d_4 = \frac{\dot{y}}{\omega_c} \text{ (from Equation (A1.1) in ESDU ME2, Reference 4)} \quad (3.21)$$

and $\gamma = \tan^{-1} \left[\frac{E}{y_m} \right].$ (3.22)

Substituting Equations (3.20), (3.21) and (3.22) in Equation (3.19) gives

$$d_2 = \left\{ (y_m^2 + E^2) + \left(\frac{\dot{y}}{\omega_c} \right)^2 - 2(y_m^2 + E^2)^{1/2} \left(\frac{\dot{y}}{\omega_c} \right) \sin \left[\tan^{-1} \left(\frac{E}{y_m} \right) \right] \right\}^{1/2}. \quad (3.23)$$

Length d_1 is obtained by substituting Equation (3.23) in Equation (3.18) and the angular velocity of the roller is given (from Equation (3.17)) by

$$\omega_r = -\omega_c \left\{ \left[\frac{\left\{ (y_m^2 + E^2) + \left(\frac{\dot{y}}{\omega_c} \right)^2 - 2(y_m^2 + E^2)^{1/2} \left(\frac{\dot{y}}{\omega_c} \right) \sin \left[\tan^{-1} \left(\frac{E}{y_m} \right) \right] \right\}^{1/2}}{R_r} \right] - 1 \right\}. \quad (3.24)$$

The angular velocity of the roller is calculated in the program using this equation at intervals of one degree of cam angle during one full rotation of the cam. The determination of roller life is presented in Section 3.3.

3.2 Swinging Roller Follower

Sketch 3.3(a) shows a typical sub-system composed of a disc cam driving a swinging roller follower. A torsion spring is used to provide force closure. The convention used is shown in Sketch 3.3(b). The derivations in this section are obtained assuming that length D is positive. When D is negative some sign changes are necessary and these are accounted for in the computer program. Diagrams showing the forces and moments acting on each part of the assembly are shown in Sketch 3.3(c).

3.2.1 Derivation of contact force

It is required to derive an expression for the contact force, P_C , between the roller and the cam due to the action of the spring, the inertia of the follower system and the external moment acting on the follower. The follower system is constrained to rotate about a pivot and it is assumed that friction torque at the pivot is zero. However, in practice friction can be included by considering it as part of the external moment, T_4 . The sum of the moments (anticlockwise positive) acting on the follower about the pivot is equal to the product of the mass-moment of inertia of the follower system (including the roller) and the angular

acceleration of the follower about the pivot. Thus for equilibrium of the follower

$$-P_{34}L \cos \alpha + T_{54} + T_4 = I_f \ddot{\psi}, \quad (3.25)$$

where P_{34} is the force exerted by the roller on the follower in the direction of the contact normal,

T_{54} is the moment exerted by the spring on the follower,

T_4 is the external moment exerted on the follower,

I_f is the mass moment of inertia of the follower system about the follower pivot

and L is the distance between the follower pivot and the roller centre.

The instantaneous motion (rotational displacement, ψ , rotational velocity, $\dot{\psi}$ and rotational acceleration, $\ddot{\psi}$) of the follower during a rise or return segment is required.

These quantities are given (for the configuration in Sketch 3.3(b)) by

$$\psi = -kw\Psi, \quad (3.26a)$$

$$\psi_m = \psi_0 - kw\Psi, \quad (3.26b)$$

$$\dot{\psi} = \frac{w'\omega_c\Psi}{\Phi} \quad (3.27)$$

$$\text{and} \quad \ddot{\psi} = \frac{w''\omega_c^2\Psi}{\Phi^2}, \quad (3.28)$$

where k is a constant of value unity that adopts the sign of D to take account of configurations in which the follower pivot lies to the left of the cam centre ($D - ve$) or to the right of the cam centre ($D + ve$),

ψ_0 is the angle between the X_m -axis and the line joining the follower pivot to the roller centre at the beginning of a segment,

Ψ is the total follower swing corresponding to one motion segment (in this Data Item it is taken to be a positive scalar quantity for both rise and return motion segments),

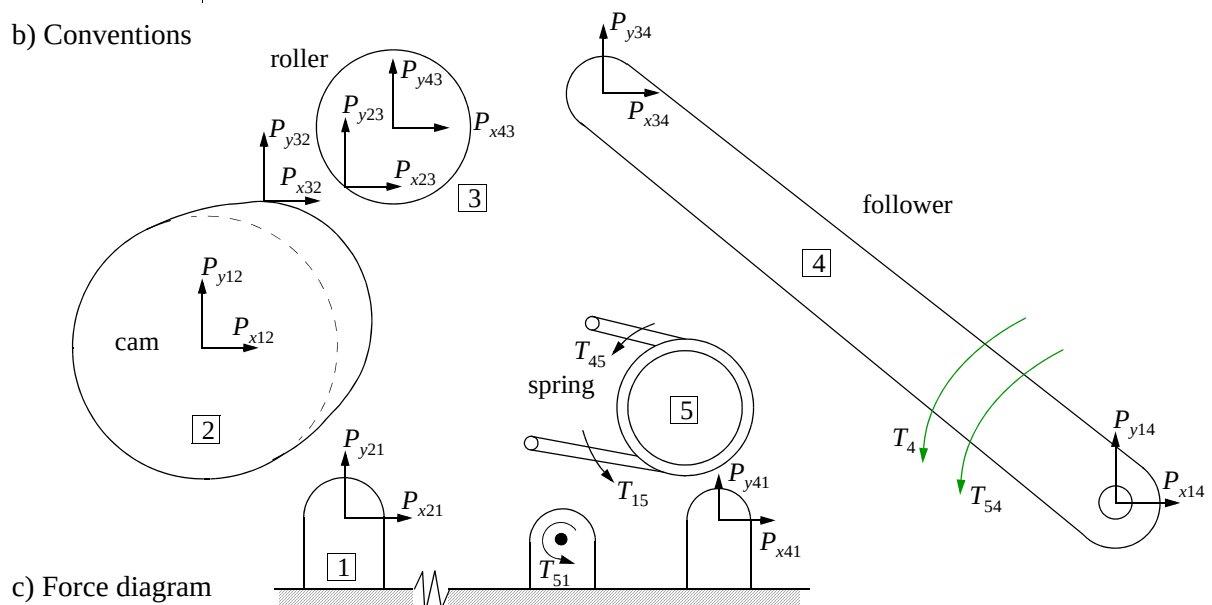
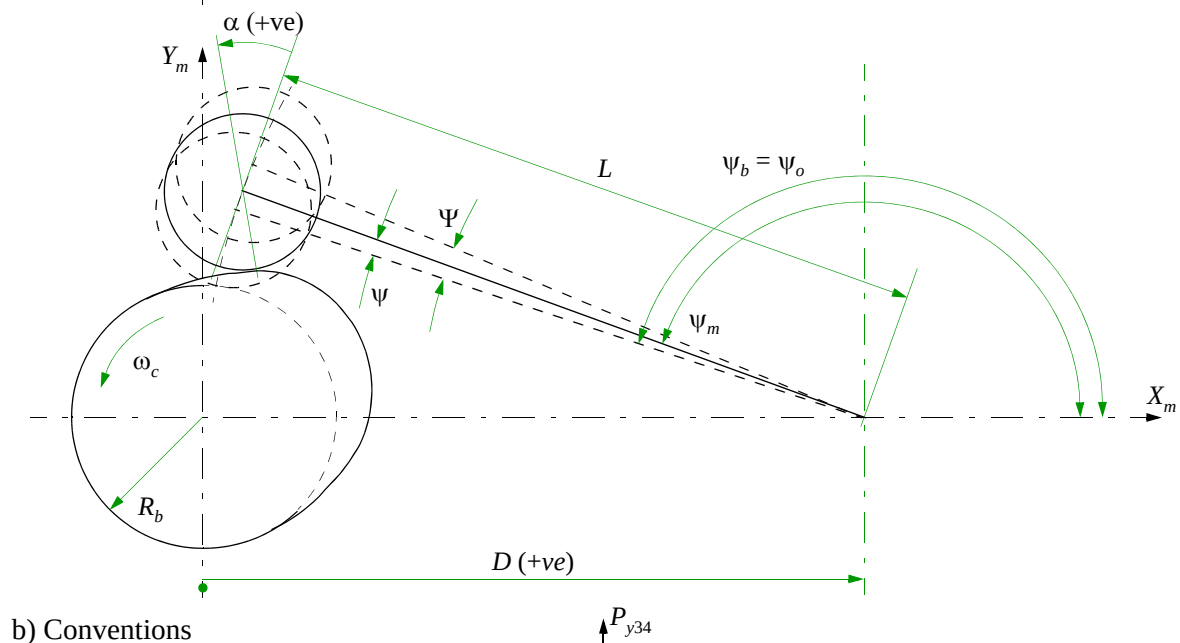
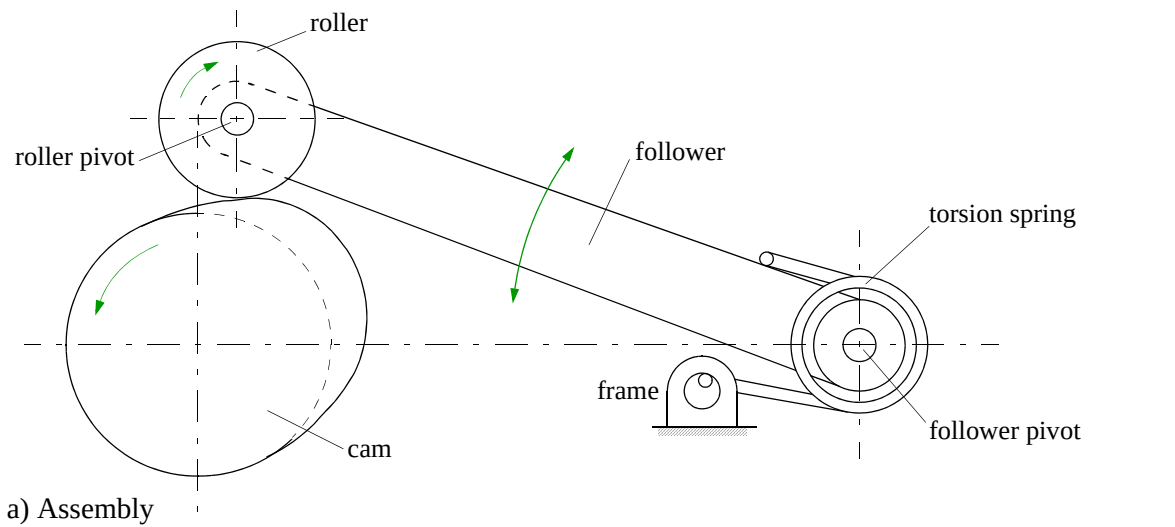
Φ is the total cam angle corresponding to one motion segment (note that units are in radians),

ω_c is the angular velocity of the cam

and w , w' and w'' are the dimensionless follower displacement, velocity and acceleration, respectively (expressions for these parameters corresponding to commonly-used cam laws are given in Data Item 82006, Reference 5).

Note: (a) w and w' are positive during a rise motion and negative during a return motion

and (b) w'' changes sign during every motion segment (it is positive during the first part of a rise segment and negative during the first part of a return segment).



Sketch 3.3 Cam and follower system

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Pressure angle is given by (see ESDU ME2, Reference 4)

$$\alpha = \tan^{-1} \left[\frac{L \left(1 + \frac{w' \Psi}{\Phi} \right) + D \cos \psi_m}{D \sin \psi_m} \right]. \quad (3.29)$$

The spring moment is composed of the preload, T_{SP} , and the product of the spring rate, k_{ST} , and the follower displacement from inner dwell and is given by

$$T_S = T_{54} = T_{SP} + k_{ST} (\psi_b - \psi_m), \quad (3.30)$$

where

$$\psi_b = \pi - \cos^{-1} \left[\frac{L^2 + D^2 - (R_b + R_r)^2}{2LD} \right].$$

It is assumed that the mass moment of inertia of the roller about its centre is negligible which implies that the torque to accelerate the outer race and the rolling elements is negligible, and so only two forces are considered to act on the roller. These forces must be of equal magnitude and opposite in direction. Therefore, for equilibrium of the roller

$$P_{23} + P_{43} = 0 \quad (3.31)$$

and the force exerted on the roller by the cam is given by

$$P_{23} = -P_{43} = P_{34}. \quad (3.32)$$

Substituting Equation (3.25) into Equation (3.32) gives

$$P_{23} = \frac{(T_{54} + T_4 - I_f \ddot{\Psi})}{L \cos \alpha}. \quad (3.33)$$

The external moment, T_E , and the spring moment, T_S , are considered as positive (by convention) when they act to increase contact force on the cam. So

$$T_E = T_4$$

and

$$T_S = T_{54}.$$

A service factor, f_w (see Section 4.1.4), can be used to account for unknown loads due to shock and vibration and so the contact force acting normal to the cam profile is defined as

$$P_C = P_{23} f_w. \quad (3.34)$$

Thus the expression for contact force, as used in the program, is obtained by substituting Equations (3.30) and (3.33) in Equation (3.34) to give

$$P_C = \frac{[T_{SP} + k_{ST}(\psi_b - \psi_m)] + T_E - I_f \ddot{\Psi}}{L \cos \alpha} f_w. \quad (3.35)$$

To evaluate the performance of the roller it is also necessary to calculate the static contact force. This is the force that could be exerted on a stationary roller and so does not include inertia effects or shock and vibration. It does include a static safety factor, s_0 (see Section 4.1.4), and is given by

$$P_{CO} = \frac{\left[\left(T_{SP} + k_{ST}(\psi_b - \psi_m) \right) + T_{EO} \right] s_0}{L \cos \alpha}, \quad (3.36)$$

where T_{EO} is the external moment that acts on the roller when the cam is stationary, *i.e.* excluding friction and any other dynamic effects.

Both static and dynamic contact forces are calculated in the program at one degree intervals during one full rotation of the cam.

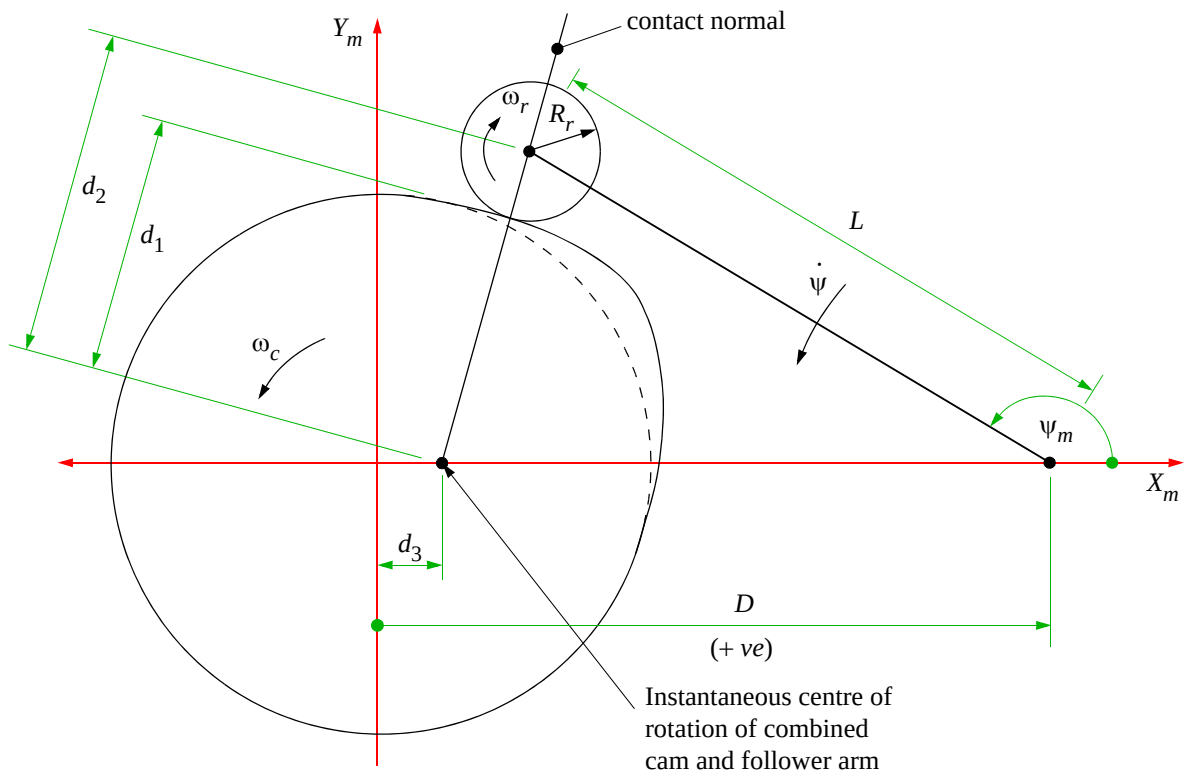
3.2.2 Derivation of roller angular velocity

The angular velocity of the roller must be expressed in terms of the known quantities. Assuming slip does not occur at the cam/roller interface the tangential velocity of the cam at the contact point is equal to the tangential velocity of the roller. Thus

$$(\omega_c - \dot{\psi})d_1 = -\omega_r R_r, \quad (3.37)$$

or

$$\omega_r = -\frac{(\omega_c - \dot{\psi})d_1}{R_r}. \quad (3.38)$$



Sketch 3.4 Angular velocity of roller for swinging follower

The cam angular velocity, ω_c , and the roller radius, R_r , are known constants. However, the length, d_1 , between the instantaneous centre of rotation (for a definition see ESDU ME2, Reference 4) of the combined cam and roller and the contact point is a variable. This length (see Sketch 3.4) may be expressed as

$$d_1 = d_2 - R_r \quad (3.39)$$

and by the cosine rule

$$d_2 = \left((D - d_3)^2 + L^2 - 2(D - d_3)L \cos(\pi - \psi_m) \right)^{1/2}, \quad (3.40)$$

where, using Kennedy's theorem of instantaneous centres,

$$d_3 = \frac{D}{\left(1 - \frac{\omega_c}{\dot{\psi}}\right)} \quad \text{(from Equation (A1.2) in ESDU ME2, Reference 4).} \quad (3.41)$$

Substituting Equation (3.41) in Equation (3.40) gives

$$d_2 = \left\{ \left[D - \frac{D}{\left(1 - \frac{\omega_c}{\dot{\psi}}\right)} \right]^2 + L^2 + 2L \left[D - \frac{D}{\left(1 - \frac{\omega_c}{\dot{\psi}}\right)} \right] \cos \psi_m \right\}^{1/2}. \quad (3.42)$$

Length d_1 is obtained by substituting Equation (3.42) in Equation (3.39) and the angular velocity of the roller is given (from Equation (3.38)) by

$$\omega_r = -(\omega_c - \dot{\psi}) \left\{ \frac{\left\{ \left[D - \frac{D}{\left(1 - \frac{\omega_c}{\dot{\psi}}\right)} \right]^2 + L^2 + 2L \left[D - \frac{D}{\left(1 - \frac{\omega_c}{\dot{\psi}}\right)} \right] \cos \psi_m \right\}^{1/2}}{R_r} - 1 \right\}. \quad (3.43)$$

The angular velocity of the roller is calculated in the program at intervals of one degree of cam angle for one complete rotation of the cam.

3.3 Evaluation of Roller Life

Roller bearings are subject to failure by fatigue and so have a limited life even when the operating conditions are ideal. The fatigue life of a particular roller cannot be predicted exactly but a well-established technique (see Derivations 1 and 2) exists for the determination of a basic rating life. This rating is defined as the life in hours that a group of apparently identical bearings operating under the same conditions will attain based