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Geneva mechanisms. Part 1: design

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THE PREPARATION OF THIS DATA ITEM

The work on this particular Data Item was monitored and guided by the Mechanisms Committee which has the following membership:

Chairman	
Dr M.R. Smith	— University of Newcastle-upon-Tyne
Members	
Mr P.J. Carden	— Ricardo Consulting Engineers, Shoreham-by-Sea
Mr T.H. Davies	— University of Technology, Loughborough
Dr G. Druce	— University of Surrey
Mr A.A. Fogarasy	— Independent
Prof. F. Freudenstein*	— Columbia University, NY, USA
Dr K.J. Mashford	— Unilever Research, Bedfordshire
Mr P.M. Nityanandan*	— Gears India (Madras) Private Ltd
Dr A. Samuel*	— University of Melbourne, Victoria, Australia
Prof. D.J. Sanger	— University of Salford
Dr C.W. Stammers	— University of Bath

* Corresponding Member

The technical work involved in the assessment of the available information and the initial construction of the Data Item was undertaken by

Mr P.J. Carden	— Ricardo Consulting Engineers, Shoreham-by-Sea
Dr J.N. Hammar	— Senior Engineer, Mechanical Motion and System Dynamics Group.

The person with overall responsibility for the work in this subject area is Mr C.J. Loughton, Head of the Mechanical Motion and System Dynamics Group.

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

GENEVA MECHANISMS. PART 1: DESIGN

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GENEVA MECHANISMS. PART 1: DESIGN

1. NOTATION AND UNITS

The units used in this Item are predominantly SI. All angles are measured in an anticlockwise sense and the crank is considered to rotate in an anticlockwise direction. Units for the computer program input parameters may be different from the units given in this section (see ESDU 96011). This reflects a balance between units in common usage and those necessary for mathematical consistency.

		Units
A	viscosity parameter (see Equation (3.67))	—
a	major axis of contact ellipse (see Equation (3.60))	m
a_1	life adjustment factor (see Equation (3.72))	—
a_2	material correction factor (see Equation (3.72))	—
a_3	lubrication correction factor (see Equation (3.72))	—
B	elasticity parameter (see Equation (3.68))	—
b	minor axis of contact ellipse (see Equation (3.61)) or width of rectangular contact (see Equation (3.53))	m
C	dynamic load rating of needle roller bearing (see Section 4.6)	N
C_o	static load rating of needle roller bearing (see Section 4.6)	N
D_{case}	diameter of casing to enclose Geneva wheel (internal Geneva)	m
D_{in}	inner diameter of wheel (internal Geneva)	m
D_{out}	outer diameter of wheel	m
E_r	Young's modulus for material of roller (or pin)	N/m ²
E_w	Young's modulus for material of wheel slot	N/m ²
E_{eff}	effective modulus (see Equation (3.62))	N/m ²
F_s	frame shaking force	N
F_{sx}, F_{sy}	components of frame shaking force in x - and y -directions, respectively	N
f_w	service factor (see Equation (3.74))	—
G	distance between crank shaft axis and wheel shaft axis	m

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H	slot face width	m
H_v	Vickers hardness (see Equation (5.1))	N/m ²
h	lubricant film thickness	m
$\bar{h}$	non-dimensional film thickness (see Sketch 3.3)	—
I_2	mass-moment of inertia of crank	kg m ²
I_4	mass-moment of inertia of wheel	kg m ²
k_r	material constant for roller (or pin) (see Equation (3.51a))	m ² /N
k_w	material constant for wheel slot (see Equation (3.51b))	m ² /N
L_c	Effective length of contact	m
L_r	roller (or pin) face width	m
L_s	length of wheel slot (see Sketch 3.1b)	m
L_t	thickness of tip (for external Geneva)	m
L_2	distance between crank shaft axis and centre of roller (or pin)	m
l_4	distance between wheel shaft axis and centre of roller (or pin)	m
$L_{10(h)a}$	modified life rating of roller (see Equation (3.72))	hours
m_2	mass of crank	kg
m_4	mass of wheel	kg
N	number of slots	—
n	number of calculation points	—
P_C	contact force	N
$(P_C)_{mod}$	(modified) load per unit contact length (see Section 3.5)	N/m
P_{21}	force at crank shaft bearing	N
P_{41}	force at wheel shaft bearing	N
P_{x21}, P_{y21}	components of force at crank shaft bearing in x- and y-directions, respectively	N

P_{x41}, P_{y41}	components of force at wheel shaft bearing in x - and y -directions, respectively	N
p	pressure of lubricant	N/m ²
p_o	maximum Hertzian pressure (see Equation (3.64))	N/m ²
R'	radius ratio (see Equation (3.66))	—
R_L	locking radius	m
R_{qr}	rms roughness of wheel slot surface	m
R_{qt}	composite roughness (see Equation (3.71))	m
R_{qw}	rms roughness of wheel slot surface	m
R_r	radius of roller (or pin)	m
R_{rc}	crown radius of roller (or pin)	m
R_{eff}	effective radius (see Equation (3.65))	m
r_2	distance from crank shaft axis to centre of mass of crank	m
r_4	distance from wheel shaft axis to centre of mass of wheel	m
s_o	static safety factor	—
T_{LOAD}	externally applied torque	N m
T_2	crank shaft torque	N m
V_{ent}	lubricant entrainment velocity	m/s
X_m, Y_m	axes fixed to machine frame with origin at crank shaft axis	—
Z_1, Z_2, Z_3	intermediate calculation parameters	—
α	pressure exponent of lubricant viscosity (see Equation (4.1))	m ² /N
β	semi-axis ratio of contact ellipse	—
γ_2	angle between the crank datum and line joining crankshaft axis to centre of mass of crank	rad
γ_4	angle between wheel datum and the line joining wheel shaft axis to centre of mass of wheel	rad
δ	decrease in separation between roller (or pin) and wheel slot due to elastic deformation	m

η	dynamic viscosity of lubricant	N s/m ²
η_0	dynamic viscosity of lubricant at atmospheric pressure and operating temperature (see Equation (4.1))	N s/m ²
λ	specific film thickness (see Equation (3.70))	—
σ_r	Poisson's ratio for material of roller (or pin)	—
σ_w	Poisson's ratio for material of wheel slot	—
ϕ	crank angle	rad
ϕ_e	crank angle at end of segment of crank rotation over which externally applied torque is non-zero (see Section 4.9)	rad
ϕ_s	crank angle at start of segment of crank rotation over which externally applied torque is non-zero (see Section 4.9)	rad
$\dot{\phi}$	angular velocity of crank	rad/s
$\ddot{\phi}$	angular acceleration of crank	rad/s ²
ψ	angle between wheel slot axis and X_m axis during indexing period	rad
ψ_l	angle between wheel slot axis and X_m axis during dwell period	rad
$\dot{\psi}$	angular velocity of wheel	rad/s
$\ddot{\psi}$	angular acceleration of wheel	rad/s ²
ω_c	angular velocity of crank	rad/s
ω_r	angular velocity of roller	rad/s

Subscripts

<i>equiv</i>	refers to equivalent
<i>i</i>	refers to value at particular calculation point
<i>max</i>	refers to maximum value
<i>min</i>	refers to minimum value
<i>x, y</i>	refers to x- and y-directions, respectively

A bar above a force variable denotes the modulus of that variable

2. INTRODUCTION

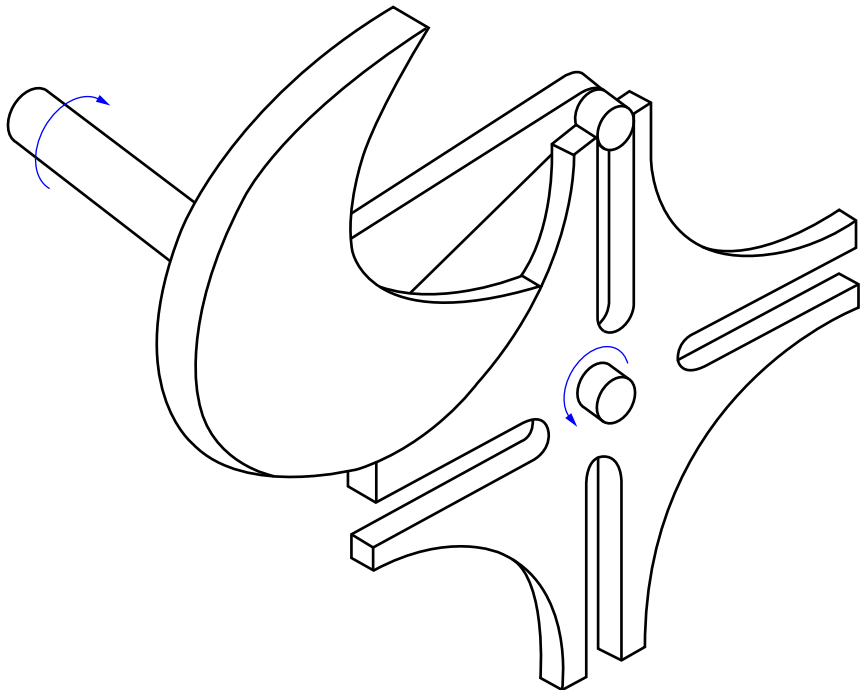
Geneva mechanisms are used widely to generate intermittent rotary motion from a continuously rotating input shaft. They have many features of interest to designers of industrial machinery including simplicity of design, cost effectiveness, precise control of the output position during motion and dwell periods, a wide range of feasible sizes, good reliability and long life.

The principal disadvantage is the lack of versatility. Once the number of the dwells per revolution of the wheel is chosen (typically between 3 and 18), the shape of the motion curves and the ratio of motion period to dwell period are fixed. Finite accelerations at the ends of the motion period will produce shock loads. Although cam based indexing mechanisms can be designed with better motion characteristics, these are usually more expensive and in many applications a well designed Geneva mechanism is the most cost-effective method for generating intermittent motion.

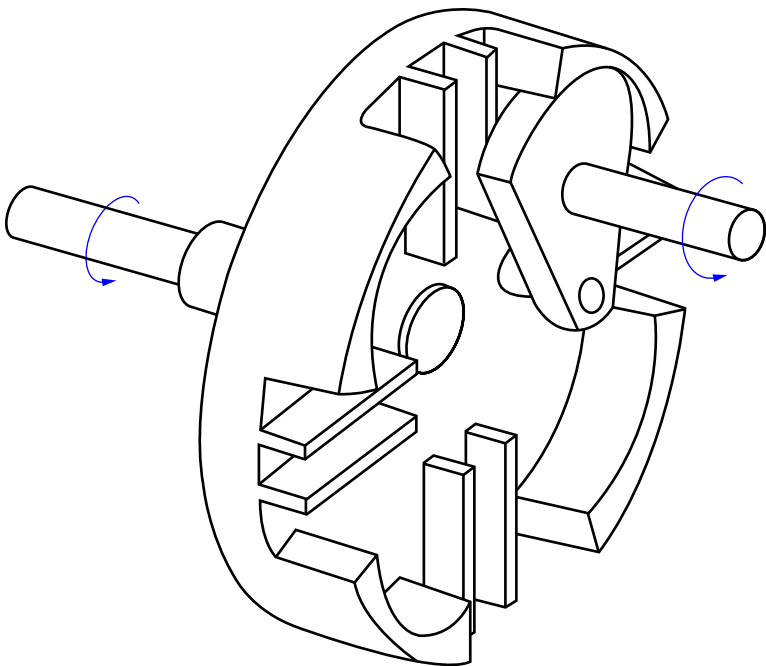
There are two basic designs of Geneva mechanism; the external Geneva (see Sketch 2.1a) and the internal Geneva (see Sketch 2.1b). Many variations on these two types have been suggested (see References 6 and 14) and most represent attempts to modify the motion characteristics for particular applications or to reduce shock loads in mechanisms intended for high speed operation. However, the basic designs have found wide use in many types of instruments and in industrial machinery and it is with these basic designs that this Item is concerned. It is useful to calculate values of many quantities (geometric parameters, kinematic characteristics, forces, torques, stresses etc.) at the design stage so that service failure by known modes can be avoided.

This Data Item presents methods for calculating the important indicators of performance of both internal and external Geneva mechanisms. The analysis is time consuming and is best carried out using the associated computer program (ESDUpac A9611 – see Reference 11). The program determines values of wheel motion, bearing forces, input torque, frame shaking force, contact stress between the roller (or pin) and the slot, lubricant film thickness (if applicable) and roller life (if applicable). In addition the program finds limiting values of certain geometric parameters and makes a number of checks on the dimensions of the mechanism. In order to communicate with other computer aided design software, a DXF file (from which the outline of the Geneva wheel can be plotted) is generated (for external Genevas only). This facility is described in greater detail in Section 5.2.4.

Correct implementation of the calculation procedures suggested in this Item should result in a decrease in design time together with improvements in machine reliability.



Sketch 2.1a Typical example of an external Geneva mechanism



Sketch 2.1b Typical example of an internal Geneva mechanism

2.1 Scope

The methods presented in this Data Item are applicable to the design of Geneva mechanisms of the external and the internal type where the input link is a simple crank rotating with either constant or varying angular velocity. The computer program is intended for the analysis of Geneva mechanisms in which the wheel has between 3 and 18 (inclusive) equally-spaced slots of equal length and the crank has a single roller (or pin) that engages with one wheel slot per revolution of the crank. In some Geneva mechanism designs these conditions do not prevail, but, with care, the program can be used to determine the performance in some of these cases (see Reference 18).

The mechanism may be enclosed in a sealed lubricated unit or not enclosed at all. The part of the crank that engages with the slot in the wheel may be a roller or a pin. If a commercially obtained (needle) roller bearing is specified then the program can estimate the expected life of that component. Both rollers and pins may be cylindrical or crowned. The program is capable of analysing mechanisms constructed from any engineering material and lubricated by one of a range of industrial lubricants.

The loads at the joints of the mechanism are expected to be primarily due to inertia but an additional external torque can be applied to the wheel to model the effects of friction or interaction with other machine parts.

2.2 Use of This Data Item

The equations required to perform all calculations are presented in Section 3 of this Data Item and full derivations are given in a separate document, ESDU ME3 (Derivation 5). It is not necessary to have a deep understanding of the theory to carry out the calculations using the accompanying computer program, ESDUpac A9611, as described in Section 2. A guide to the choice of input values for this program is given in Section 4 and the use of the results when analysing the likely performance of a specified mechanism is presented in Section 5. Guidance on design improvements is also given in Section 5. Examples illustrating the use of the computer program are presented in a separate document (see Reference 18).

The calculations may be useful in design situations such as:

- (a) during a feasibility study, especially when determining the space required for a mechanism with specified motion and load characteristics,
 - (b) part of a detail design, especially for the selection of critical dimensions and the evaluation of materials,
- or
- (c) investigation of the causes of in-service problems such as deterioration of the roller (or pin) or the slot surface, shaft breakage, bearing failure, frame vibration, crankshaft vibration, wheel breakage, *etc.*

It is likely that the nature of the information available in situations (a) to (c) will vary. For instance during a feasibility study, the mass moment of inertia of the wheel and the externally applied torque may not be known accurately. The analytical procedure is applicable when uncertainty surrounds values of inputs but the accuracy of the results is entirely dependent on the quality of the input information.

3. THEORETICAL BACKGROUND

3.1 Introduction

This section presents the equations on which the calculation procedures are based. It is strongly recommended that the calculations are carried out using the computer program, ESDUpac A9611, described in ESDU 96011 (Reference 11). The equations used to evaluate the geometric parameters, the motion of the wheel, the force at each joint, the input torque and the frame shaking force are different for the external and internal mechanisms and these are given in Sections 3.2 and 3.3, respectively. The derivation of these equations is given in ESDU ME3 (Derivation 5). Expressions for contact stress, lubricant film thickness and roller life are given in Sections 3.4, 3.5 and 3.6, respectively. The derivation of these equations is given in other ESDU documents (see Derivations 1 to 4). Recommended values of the quantities needed as inputs to the computer program are given in Section 4.

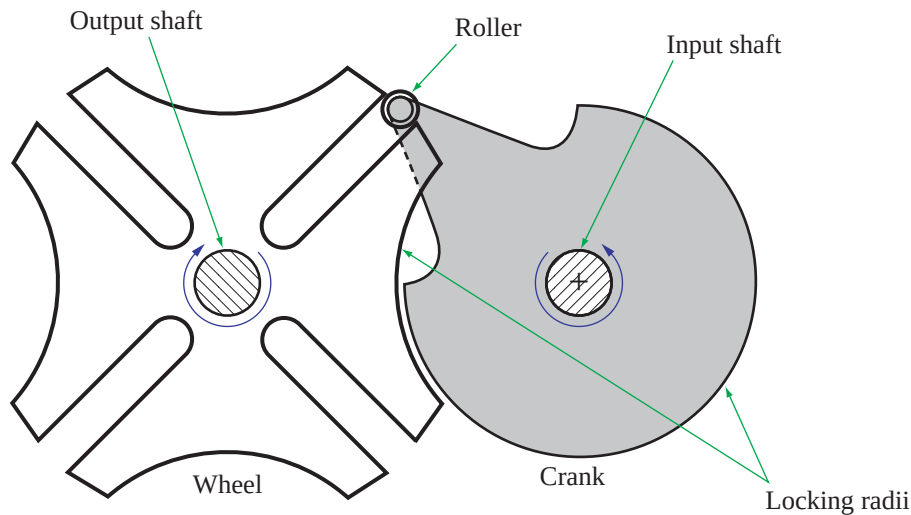
Typically, there are two design strategies by which a Geneva mechanism is analysed and designed. In the first, the centre distance between the crank and wheel shafts is specified and the dependent wheel and crank dimensions are evaluated. Alternatively, the dimensions of the wheel may be specified and the crank dimensions, together with the distance between the two shafts, become the dependent parameters to be determined.

Sections 3.2.1 and 3.3.1 consider the evaluation of the appropriate dependent geometric quantities for the external and internal geneva, respectively. Note that the two sets of equations involve identical parameters, albeit developed in a different order, for both the external and internal geneva mechanisms.

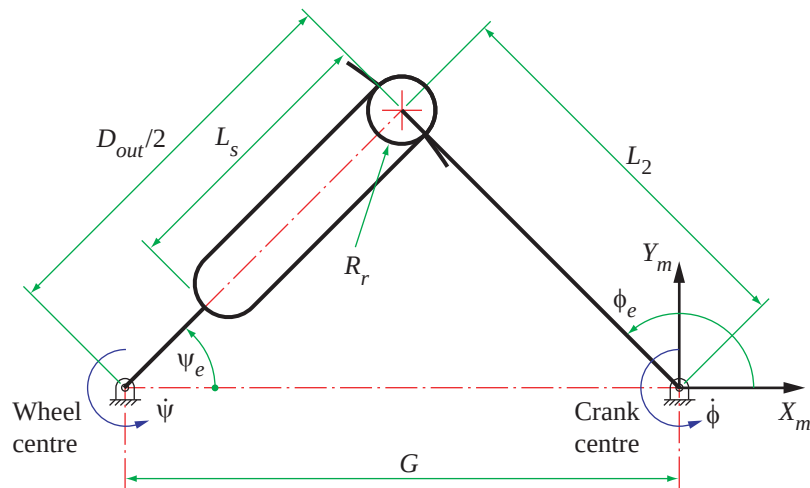
In the mathematical model of a well-designed Geneva mechanism, the drive pin (usually a roller) enters the slot in the output wheel along the centre line of the slot. In theory this avoids significant impact forces between the roller and the slot. A detailed description of the application of the theory and the assumptions used are given in ESDU ME3 (see Derivation 5).

3.2 External Geneva Mechanism

Sketch 3.1a shows a typical sub-system of an external Geneva mechanism composed of a crank with a single roller and a wheel with four slots. The functional dimensions of an external Geneva mechanism are shown in Sketch 3.1b.



Sketch 3.1a External Geneva mechanism



Sketch 3.1b Functional dimensions of an External Geneva mechanism

3.2.1 Geometric quantities

It is assumed that the roller (or pin) engages with the wheel slot when the slot centre line is perpendicular to the line joining the crankshaft axis to the centre of the roller (or pin). This simple rule ensures that the wheel velocity is zero at that point of engagement and so, with the correct clearance specified between the roller (or pin) and the slot (see Section 5.4h), impact does not occur.

In determining the basic dimensions of the mechanism, the choice of design strategy (see Section 3.1) fixes the set of equations required to specify the relationship between the number of slots, N , the centre distance, G , the crank length, L_2 , and the outer diameter of the output wheel, D_{out} .

For an external Geneva, in which the number of slots and the centre distance are specified, the crank length is given by

$$L_2 = G \sin\left(\frac{\pi}{N}\right) \quad (3.1)$$

and the outer diameter of the wheel is

$$D_{out} = 2\sqrt{G^2 - L_2^2} \quad (3.2)$$

Alternatively, for an external Geneva in which the number of slots and the outer diameter of the wheel are specified, the crank length is found from

$$L_2 = \left(\frac{D_{out}}{2}\right) \tan\left(\frac{\pi}{N}\right) \quad (3.3)$$

and the centre distance, between the crank shaft axis and the wheel axis, is

$$G = \sqrt{\frac{(D_{out})^2}{4} + L_2^2} \quad (3.4)$$

With the dimensions given by Equations (3.1) and (3.2) or (3.3) and (3.4), together with the radius of the roller (or pin), R_r , it is possible to calculate the minimum length of the slot as

$$(L_s)_{min} = L_2 + R_r - \left(G - \frac{D_{out}}{2}\right) \quad (3.5)$$

the upper limit on the size of the locking radius as

$$(R_L)_{max} = L_2 - R_r \quad (3.6)$$

the lower limit on the size of the locking radius as

$$(R_L)_{min} = G - \frac{D_{out}}{2} \quad (3.7)$$

and the thickness of the wheel tip (distance between the slot and the locking radius) as

$$L_t = L_2 - R_r - R_L \quad (3.8)$$

3.2.2 Wheel motion

It is assumed that the leading dimensions of the mechanism have been calculated using the equations presented in Section 3.2.1 and that the crank rotates with angular velocity, ϕ . During the periods when the wheel is locked the angle between the X_m axis and the centre line of the slot is given by

$$\psi_l = \frac{\pi}{N} \quad (3.9)$$

and the velocity and acceleration of the wheel are both zero.

During the period when the wheel is moving, the angle between the X_m axis and the centreline of the slot is given by

$$\psi = \arctan \left[\frac{\sin \phi}{\cos \phi + \frac{G}{L_2}} \right] \quad (3.10)$$

and the displacement of the output wheel, from its angle at the start of the motion, is

$$\psi_L - \psi = \frac{\pi}{N} - \arctan \left[\frac{\sin \phi}{\cos \phi + \frac{G}{L_2}} \right] . \quad (3.11)$$

Also, during this motion period, the (radial) distance of the roller (or pin) centre from the wheel shaft axis is given by

$$l_4 = \sqrt{L_2^2 + G^2 + 2L_2G \cos \phi} , \quad (3.12)$$

the linear velocity of the roller (or pin) along the slot is

$$\dot{l}_4 = \frac{-GL_2 \sin \phi \dot{\phi}}{l_4} \quad (3.13)$$

and the angular velocity of the output wheel is

$$\dot{\psi} = \frac{L_2 \cos(\psi - \phi) \dot{\phi}}{l_4} , \quad (3.14)$$

while the linear acceleration of the roller (or pin) along the slot is

$$\ddot{l}_4 = L_2 [\sin(\psi - \phi) \ddot{\phi} - \cos(\psi - \phi) \dot{\phi}^2] + l_4 \dot{\psi}^2 \quad (3.15)$$

and the angular acceleration of the output wheel is

$$\ddot{\psi} = \frac{L_2}{l_4} [\cos(\psi - \phi) \ddot{\phi} + \sin(\psi - \phi) (\dot{\phi}^2 - 2\dot{\psi} \dot{\phi})] . \quad (3.16)$$

3.2.3 Forces and torques

In this section it is assumed that the crank, wheel and roller (or pin) are stiff enough to be considered rigid and that once the roller (or pin) has engaged with the slot these two parts remain in contact until the wheel is locked for the next dwell period. The roller (or pin) moves from one side of the slot to the other when the direction of the force at that point reverses. This motion is assumed to take place instantaneously and without impact. It is also assumed that all the forces and the torques are due only to inertia and any externally-applied torque exerted on the wheel. For a full derivation of the force equations, see Section 6 in ESDU ME3 (Derivation 5).

When the wheel is in the locked position, there is no contact between the roller (or pin) and the slot. Consequently, the components of force at the crank shaft bearing during the dwell period are given by

$$P_{x21} = m_2 r_2 [\dot{\phi}^2 \cos(\phi + \gamma_2) + \ddot{\phi} \sin(\phi + \gamma_2)] \quad (3.17a)$$

and
$$P_{y21} = m_2 r_2 [\dot{\phi}^2 \sin(\phi + \gamma_2) - \ddot{\phi} \cos(\phi + \gamma_2)] . \quad (3.17b)$$

When the wheel is in motion, the contact force between the roller (or pin) and the slot is given by

$$P_C = \frac{(T_{LOAD} - I_4 \ddot{\psi}) f_w}{l_4} \quad (3.18)$$

where f_w is a service factor, that can be used to account for unknown loads due to shock and vibration.

The input torque (required at the crank shaft to drive the mechanism) is given by

$$T_2 = - \frac{(T_{LOAD} - I_4 \ddot{\psi}) \dot{\psi}}{\dot{\phi}} . \quad (3.19)$$

The components of force at the crank shaft bearing during the motion period are given by

$$P_{x21} = m_2 r_2 [\dot{\phi}^2 \cos(\phi + \gamma_2) + \ddot{\phi} \sin(\phi + \gamma_2)] - P_C \sin \psi \quad (3.20a)$$

and
$$P_{y21} = m_2 r_2 [\dot{\phi}^2 \sin(\phi + \gamma_2) - \ddot{\phi} \cos(\phi + \gamma_2)] + P_C \cos \psi . \quad (3.20b)$$

The components of force at the wheel shaft bearing during the motion period are given by

$$P_{x41} = m_2 r_2 [\dot{\psi}^2 \cos(\phi + \gamma_4) + \ddot{\psi} \sin(\phi + \gamma_4)] + P_C \sin \psi \quad (3.21a)$$

and
$$P_{y41} = m_2 r_2 [\dot{\psi}^2 \sin(\phi + \gamma_4) - \ddot{\psi} \cos(\phi + \gamma_4)] - P_C \cos(\psi_m) . \quad (3.21b)$$

The components of shaking force acting on the frame throughout the motion are given by

$$F_{sx} = P_{x41} + P_{x21} \quad (3.22a)$$

and
$$F_{sy} = P_{y41} + P_{y21} . \quad (3.22b)$$

The overall magnitudes of the forces evaluated in this section are given by

$$\overline{P}_C = |P_C|, \quad (3.23)$$

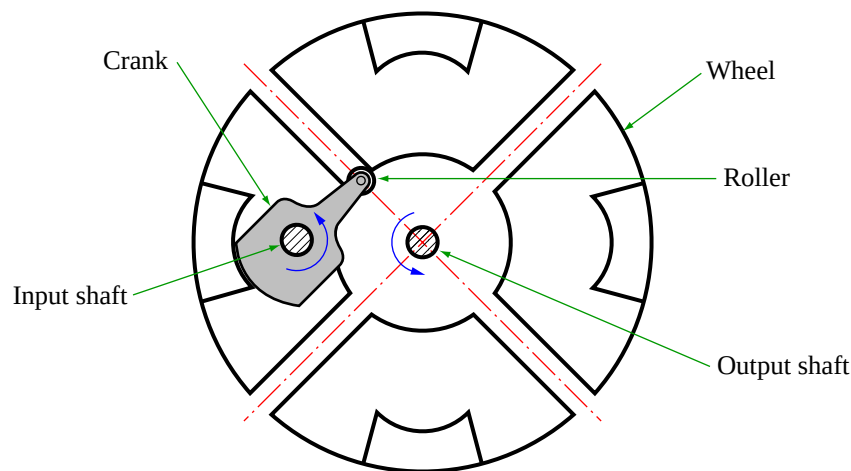
$$\overline{P}_{21} = [(P_{x21})^2 + (P_{y21})^2]^{1/2}, \quad (3.24)$$

$$\overline{P}_{41} = [(P_{x41})^2 + (P_{y41})^2]^{1/2} \quad (3.25)$$

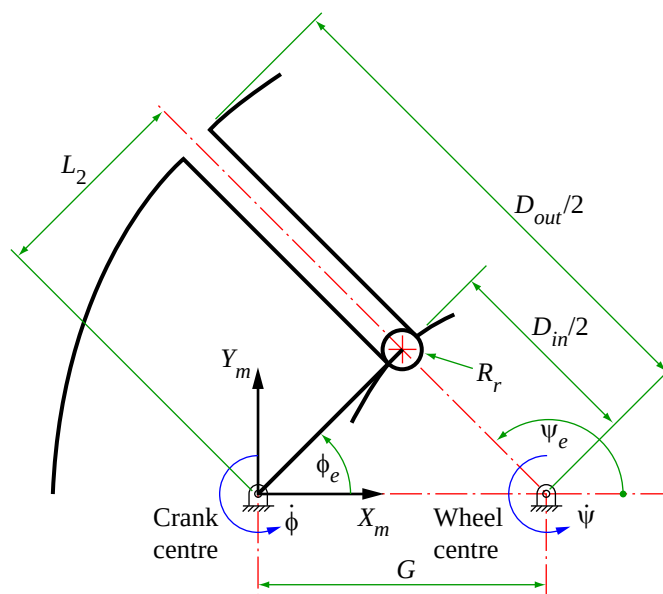
and
$$\overline{F}_s = [(F_{sx})^2 + (F_{sy})^2]^{1/2}. \quad (3.26)$$

3.3 Internal Geneva Mechanism

Sketch 3.2a shows a typical sub-system of an internal Geneva mechanism composed of a crank with a single roller and a wheel with four slots. The functional dimensions which affect the motion of an internal Geneva mechanism are shown in Sketch 3.2b.



Sketch 3.2a Internal Geneva mechanism



Sketch 3.2b Functional dimensions of an internal Geneva mechanism

3.3.1 Geometric quantities

It is assumed that the roller (or pin) engages with the wheel slot when the slot centre line is perpendicular to the line joining the crankshaft axis to the centre of the roller (or pin). This simple rule ensures that the wheel velocity is zero at that point of engagement and so, with the correct clearance specified (see Section 5.4h) between the roller (or pin) and the slot, impact does not occur.

In determining the basic dimensions of the mechanism, the choice of design strategy (see Section 3.1) fixes the set of equations required to specify the relationship between the number of slots, N , the centre distance, G , the crank length, L_2 , and the inner diameter of the output wheel, D_{in} .

For an internal Geneva, in which the number of slots and the centre distance are specified, the crank length is given by

$$L_2 = G \sin\left(\frac{\pi}{N}\right) \quad (3.27)$$

and the inner diameter of the wheel is

$$D_{in} = 2\sqrt{G^2 - L_2^2} \quad (3.28)$$

Alternatively, for an internal Geneva in which the number of slots and the inner diameter of the wheel are specified, the crank length is found from

$$L_2 = \left(\frac{D_{in}}{2}\right) \tan\left(\frac{\pi}{N}\right) \quad (3.29)$$

and the centre distance, between the crank shaft axis and the wheel axis, is

$$G = \sqrt{\frac{(D_{in})^2}{4} + L_2^2} \quad (3.30)$$

With the dimensions given by Equations (3.27) and (3.28) or (3.29) and (3.30), together with the radius of the roller (or pin), R_r , it is possible to calculate the minimum outer diameter of the wheel which is

$$(D_{out})_{min} = 2[G + L_2 + R_r] \quad (3.31)$$

and the upper limit on the size of the locking radius is

$$(R_L)_{max} = L_2 + R_r \quad (3.32)$$

3.3.2 Wheel motion

It is assumed that the leading dimensions of the mechanism have been calculated using the equations presented in Section 3.3.1 and that the crank rotates with angular velocity, ϕ . During the periods when the wheel is locked the angle between the X_m axis and the centre line of the slot is given by

$$\psi_l = \pi - \frac{\pi}{N} \quad (3.33)$$

and the velocity and acceleration of the wheel for these periods are zero.