
British Standard Specification for

Worm gearing

Part 2. Metric units

Roues à vis sans fin — Spécifications
Partie 2. Unités métriques

Schneckenradverzahnung
Teil 2. Metrische Einheiten

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Foreword

This Part of this British Standard has been prepared under the direction of the Mechanical Engineering Standards Committee and specifies worm gearing in metric units.

Part 1 of this standard specifies worm gearing in imperial units.

Compliance with a British Standard does not of itself confer immunity from legal obligations.

British Standard Specification for

Worm gearing

Part 2. Metric units

1. Scope

This Part of BS 721 specifies the requirements for worm gearing based on axial modules. Four classes of gear are specified, which are related to function and accuracy.

This Part applies to worm gearing comprising cylindrical involute helicoid worms and wormwheels conjugate thereto. It does not apply to pairs of cylindrical gears connecting non-parallel axes known as crossed helical gears.

Appendix A gives some typical wheel rim sections together with methods of attachment. Appendix B gives examples of calculations using the formulae given in the standard. Provision is made in appendix C for a method of application of existing hobs designated in terms of axial pitch.

NOTE 1. Worm gearing designed to this standard is not to be regarded as irreversible.

NOTE 2. The titles of the publications referred to in this standard are listed on the inside back cover.

2. Definitions

For the purposes of this British Standard, the definitions given in BS 2519 : Part 1 apply.

3. Symbols

For the purposes of this British Standard, the following symbols apply.

NOTE. Where applicable, the symbols are in accordance with BS 2519 : Part 2.

Symbol	Term
$h_{a,1}$	Worm thread addendum
$h_{f,1}$	Worm thread dedendum
c	Clearance
a	Centre distance
$d_{a,1}$	Worm tip diameter
d_1	Worm reference circle diameter
d_b	Base diameter of involute helicoid
$d_{f,1}$	Worm root diameter
d_2	Wormwheel reference circle diameter
$d_{a,2}$	Wormwheel tip diameter
$d_{f,2}$	Wormwheel root diameter
d_t	Wormwheel throat diameter
b_1	Length of worm
b_e	Wormwheel effective face width
$\bar{s}_c$	Constant chord thickness
$\bar{h}_c$	Constant chord height
x_2	Wormwheel addendum modification coefficient
K_A	Application factor

L	Length of arc (any selected)
$l_{f,2}$	Length of root of wormwheel teeth
p_z	Lead of worm threads
L_x	Axial distance between any two points at the same radial distance from the worm axis
m	Axial module
m_n	Normal module
M	Permissible torque on wormwheel
M_{act}	Actual torque on wormwheel
N	Speed of roadwheel (in r/min)
n_1	Speed of worm (in r/min)
n_2	Speed of wormwheel (in r/min)
p_n	Normal pitch
p_x	Axial pitch
q	Diameter factor
T	Working temperature of oil (in °C)
γ_g	Wormwheel gorge radius
$\sigma_{bm,1}$	Bending stress factor: worm
$\sigma_{bm,2}$	Bending stress factor: wormwheel
$\sigma_{cm,1}$	Surface stress factor: worm
$\sigma_{cm,2}$	Surface stress factor: wormwheel
z_1	Number of threads in worm
z_2	Number of teeth in wormwheel
H	Period (in h) in load cycle
H_b	Equivalent running time (in h) per cycle for strength
H_c	Equivalent running time (in h) per cycle for wear
w	Period in load cycle during which torque changes uniformly
$H_{b,w}$	Equivalent running time (in h) for period w for strength
$H_{c,w}$	Equivalent running time (in h) for period w for wear
H_{eb}	Total equivalent running time (in h) for strength
H_{ec}	Total equivalent running time (in h) for wear
v_s	Rubbing speed
$X_{b,1}$	Speed factor for strength: worm
$X_{b,2}$	Speed factor for strength: wormwheel
$X_{c,1}$	Speed factor for wear: worm
$X_{c,2}$	Speed factor for wear: wormwheel
Z	Zone factor
γ	Lead angle of worm thread
γ_b	Base lead angle of worm thread
α_n	Normal pressure angle

ϕ	Angle of friction
θ	The smaller of the two angles formed by radii through two points on a worm thread surface projected onto a transverse plane of the worm
G	Gradient factor*
M_e	Maximum engine torque*
R_t	Rolling radius of tyre*
R_1	Gearbox ratio for lowest road speed*
R_2	Rear axle ratio*
R_g	Gear ratio*
W_1	Load on driving axle*
W	Gross laden weight of vehicle*
μ	Coefficient of friction*

NOTE. In some mathematical progressions, symbols such as $n_{1,1}$ appear (e.g. see 9.3.1) where the first subscript refers to the worm and the second to the order in the progression.

4. Form of worm threads and wormwheel teeth

4.1 Standard worm thread surfaces. The standard worm thread surfaces shall be involute helicoids with a normal pressure angle of 20° at the reference circle diameter $d_1 = qm$.

4.2 Basic rack. The basic rack tooth profile shall comply with BS 436 : Part 2 with the exception that variation of clearance shall be permitted within the limits $0.2m \cos \gamma$ to $0.25m \cos \gamma$ according to the method of manufacture (see figure 1).

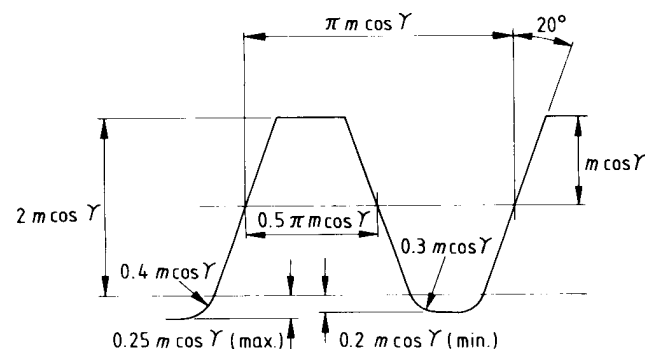


Figure 1. Normal section of basic rack (dimensioned in terms of axial module)

Sharp edges shall be removed from the tips of the worm threads.

NOTE. Certain worms with large lead angles would have undercut threads and/or too narrow thread crests unless modified. In such cases by withdrawal of the basic rack the root diameter and thread thickness are increased and the dedendum decreased but the tip diameter and reference addendum remain unaltered. Formulae for calculating the necessary dimensions are given in 6.3.

4.3 Thread thickness. The thread thickness shall be gauged on the constant chord thickness $\bar{s}_c$.

4.4 Wormwheel teeth and entry gaps. The form of the wormwheel teeth shall be as defined in 4.1.2.2 of BS 2519 : Part 1 : 1976.

NOTE. An 'entry gap' giving a light load contact similar to that shown in figure 2 is permitted.

4.5 Wormwheel tooth fillet. This shall be such as would be generated by a hob having a radius at its tip not less than $0.25m \cos \gamma$ or $0.08p_n$, and not exceeding $0.3m \cos \gamma$ or $0.095p_n$.

NOTE. Removal of sharp edges from the tips of the wormwheel teeth is optional.

4.6 Face width of wormwheel. The effective wormwheel face width, b_e , shall be equal to $2m\sqrt{(q+1)}$, or the actual face width, whichever is the lesser.

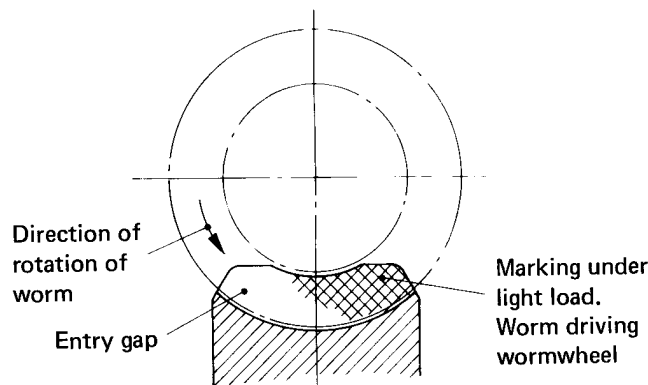


Figure 2. Entry gap in wormwheel teeth

5. Information on worm gear elements

5.1 Worm gear designations. A pair of worm gears shall be defined by the designation of the number of threads in the worm, number of teeth in the wormwheel, diameter factor and axial module, written in the brief form $z_1/z_2/q/m$.

NOTE 1. This information, together with the centre distance a , is sufficient to enable all detail dimensions to be calculated from the formulae and tables given in this standard. For example, a designation 3/44/8/9.8 at 255 mm centres signifies a 44 tooth wormwheel working at 255 mm centre distance with a 3 start worm of 9.8 mm axial module and a diameter factor 8.

NOTE 2. In order that existing hobs designated in terms of axial pitch may be employed in a manner consistent with the requirements of this standard, the method of application of such hobs should be as given in appendix C.

5.2 Determination of z_1 and z_2 . For industrial gears, z_1 shall be taken as the nearest whole number to the value given by the following expression:

$$\frac{7 + 2.4\sqrt{a}}{R_g}$$

z_2 shall then be taken as the next whole number below the value $R_g z_1$, except in the case of single thread worms when $z_2 = R_g$. The value of z_2 shall be not less than 17.

For vehicle axle transmission gears, z_1 shall be taken as the nearest whole number to the value given by:

$$\frac{7 + 1.6\sqrt{a}}{R_g}$$

5.3 Diameter factor q

5.3.1 Choice of worm diameter. Choice of worm diameter is limited by considerations of strength and efficiency. Since the efficiency of the gear increases as the worm diameter is reduced, the worm diameter shall be as small as possible consistent with adequate strength at the root section of the worm. The worm reference circle diameter, d_1 , is given by the product of the axial module, m , and the diameter factor q . For a given value of q (which shall be selected from the range given in 5.3.2) the reference circle diameter is therefore $d_1 = qm$.

NOTE. The worm reference circle diameter, d_1 , is equal to qm and is thus similar to the reference diameter of a spur gear having q teeth and m module. Similarly, the centre distance of a pair of worm gears (unless extended or closed) is equal to that of a pair of spur gears

*These symbols apply only to vehicle axle transmission gears (e.g. see 9.5).

having q and z_2 teeth, i.e. $a = 0.5m(q + z_2)$. The calculation of the worm tip diameter $d_{a,1}$ is also analogous to that of a spur gear with q teeth and m module and is found from $d_{a,1} = m(q + 2)$.

5.3.2 Choice of diameter factor q . For the purposes of this standard, values for q shall be selected from the following range of values:

6, 6.5, 7, 7.5, 8, 8.5, 9, 9.5, 10, 11, 12, 13, 14, 17, 20

NOTE. This ensures geometric similarity between all worms having the same values of z_1 and q . From any given combination of z_1 and q , all angular dimensions of a worm can be calculated and all linear dimensions will be proportional to the selected modules.

5.3.3 Determination of diameter factor q . A preliminary selection of the value of diameter factor q shall be made from figure 3, which may be used for any worm rotational speed. The value of q selected shall be as small as possible, except where low reversed efficiency is desired (see clause 11). For worm rotational speeds below 300 r/min, when some reduction of efficiency may be accepted in order to increase the worm bending strength, a preliminary selection of the value of q shall be made from figure 4. These methods of selecting values of q permit preparation of preliminary designs that shall be checked for acceptability of worm bending stress, and, if necessary, other values of q shall be chosen to maintain worm bending stress within specified limits, taking into account the bearing span.

For vehicle axle transmission gears, the value of q shall be selected from figure 3.

NOTE. The larger worm diameter factors from 12 upwards are seldom required except when the worm is bored. In such cases it is recommended that the thickness of metal below the thread be approximately equal to the depth of the thread.

5.4 Lead angle of worm thread. The lead angle, γ , of the worm thread at the reference cylinder shall be determined from $\tan \gamma = z_1/q$.

6. Formulae for calculation of dimensions

6.1 Limiting values of centre distance for given values of z_1, z_2, q and m . The limiting values of centre distance a for given values of z_1, z_2, q and m shall be as follows:

$$a_{\max} = 0.5m(z_2 + q + x_{2,\max})$$

where $x_{2,\max}$ is as given in figure 5.

$$a_{\min} = 0.5m(z_2 + q - x_{2,\min})$$

where $x_{2,\min}$ is as given in figure 6.

6.2 Limiting values of axial module for given values of a, z_1, z_2 and q . The limiting values of axial module m for given values of a, z_1, z_2 and q shall be as follows:

$$m_{\max} = \frac{2a}{z_2 + q - 2x_{2,\min}}$$

$$m_{\min} = \frac{2a}{z_2 + q + 2x_{2,\max}}$$

***Clearance.** The clearance is given maximum and minimum values (see 4.2). For new designs based on the module system, the clearance should be made to the minimum value given. If it is desired to use tools made to the depths given in BS 436 : Part 2 for cutting the worm threads, the maximum clearance may be adopted when the cutting is a finishing process, but if the cutting of the worm is used as a pre-grinding process, the clearance may be reduced to not less than the stated minimum.

†**Depth reduction.** Depth reduction is necessary on all worms on which the wormwheel throat would, from the above formulae, lie within the base circle of the worm. Those worms with a standard q value equal to or less than that given below for a given number of threads require a depth reduction.

$$z_1 = 3 \quad 4 \quad 5 \quad 6 \quad 7$$

$$q = 6.5 \quad 7 \quad 7.5 \quad 8 \quad 8$$

In such cases of depth reduction the formulae marked † become

$$h_{f,1} = 0.5(d_1 - d_b) + c$$

$$\bar{s}_c = 2.0298m \cos \gamma - 0.3214(d_1 - d_b)$$

6.3 Formulae for detail dimensions of worm and wheel

6.3.1 Worm. Worm dimensions shall be calculated from the following formulae.

$$\begin{aligned} \tan \gamma &= z_1/q \\ d_1 &= qm \\ h_{a,1} &= m \\ {}^*c_{\min} &= 0.2m \cos \gamma \\ {}^*c_{\max} &= 0.25m \cos \gamma \\ th_{f,1,\min} &= m(2.2 \cos \gamma - 1) \\ th_{f,1,\max} &= m(2.25 \cos \gamma - 1) \\ d_{a,1} &= d_1 + 2h_{a,1} = m(q + 2) \\ d_{f,1} &= d_1 - 2h_{f,1} \\ p_z &= \pi z_1 m \\ \alpha_n &= 20^\circ \\ \cos \gamma_b &= \cos \gamma \cos \alpha_n \\ d_b &= \frac{p_z}{\pi \tan \gamma_b} \\ b_1 &= 14m \cos \gamma \\ t\bar{s}_c &= m(0.7444 \cos \gamma + 0.6428) \\ \bar{h}_c &= m - \frac{\bar{s}_c}{2} \tan 20^\circ \end{aligned}$$

6.3.2 Wheel. Detail dimensions of wheels shall be calculated from the following formulae.

$$\begin{aligned} d_2 &= 2a - d_1 \\ d_t &= 2a - (d_{f,1} + 2c) \\ d_{f,2} &= 2a - (d_{a,1} + 2c) \\ d_{a,2,\min} &= d_t + 0.4m \\ d_{a,2,\max} &= d_t + m \\ \gamma_g &= 0.5(d_{f,1} + 2c) \\ {}^*c_{\min} &= 0.2m \cos \gamma \\ {}^*c_{\max} &= 0.25m \cos \gamma \\ b_e &= 2m \sqrt{(q + 1)} \\ m &= \frac{p_x}{\pi} \\ m_n &= m \cos \gamma \\ p_x &= \pi m \\ p_n &= p_x \cos \gamma \\ l_{f,2} &= (d_{a,1} + 2c) \sin^{-1} \left[\frac{b_e}{d_{a,1} + 2c} \right] \end{aligned}$$

where the angle is in radians.

NOTE. Some typical wormwheel rim sections and methods of attachment of rim to hub are shown in appendix A.

7. Accuracy of manufacture

7.1 Tolerances applicable to worms

7.1.1 Run out of worm tip diameter with reference to datum. This shall not exceed the appropriate value for grade 1T7 of BS 4500 : Part 1.

7.1.2 Worm pitch tolerances. The tolerances for pitch accuracy shall not exceed the values shown in table 1.

Table 1. Worm pitch tolerance

Class	Pitch tolerance
	mm
A	$0.0016 \sqrt{L_x} + 0.00008 \sqrt{(\theta d_1)} + 0.005$
B	$0.0025 \sqrt{L_x} + 0.000126 \sqrt{(\theta d_1)} + 0.008$
C	$0.004 \sqrt{L_x} + 0.0002 \sqrt{(\theta d_1)} + 0.013$
D	$0.006 \sqrt{L_x} + 0.00032 \sqrt{(\theta d_1)} + 0.020$

NOTE 1.

d_1 is the reference circle diameter of the worm (in mm);

L_x is the axial distance between any two points at the same radial distance from the worm axis (in mm);

θ is the smaller of the two angles formed by radii through two points on a worm thread surface projected onto a transverse plane of the worm.

NOTE 2. The tolerances cover all types of deviation in helix and spacing of worm threads, e.g. those usually known as 'transverse pitch tolerance', 'axial pitch tolerance', and 'undulations'.

For transverse pitch deviation the first term becomes zero.

For axial pitch deviation the second term is zero. For checking undulation height or normal pitch deviations, all three terms are required.

NOTE 3. For hobbing machine table gears, reference should be made to BS 1498 and BS 4656 : Part 19.

7.1.3 Profile tolerance. The profile deviation when measured normal to the thread surface shall not exceed the values shown in table 2.

Table 2. Worm profile tolerance

Class	Tolerance
	mm
A	$0.00025m + 0.003$
B	$0.0004m + 0.004$
C	$0.0010m + 0.006$
D	$0.0025m + 0.015$

The deviation shall be measured as a departure from the straight line generator tangential to the base cylinder coincident with the actual profile at mid-working depth.

7.1.4 Worm thread thickness tolerance. The tolerances on worm thread thickness shall be as given in table 3.

Table 3. Worm thread thickness tolerance

Class	Tolerance
	mm
A	$+0.000$ $-(0.0025m + 0.020)$
B	$+0.000$ $-(0.0035m + 0.033)$
C	$+0.000$ $-(0.0063m + 0.051)$
D	$+0.000$ $-(0.010m + 0.081)$

7.2 Wormwheel pitch tolerances. The adjacent and the cumulative pitch deviation measured in a plane normal to the gear axis and on each set of flanks independently shall not exceed the values shown in table 4.

Table 4. Wormwheel pitch tolerance

Class	Tolerance
	mm
A	$0.0010 \sqrt{L} + 0.0025$
B	$0.0025 \sqrt{L} + 0.0064$
C	$0.0066 \sqrt{L} + 0.0160$
D	$0.0161 \sqrt{L} + 0.0406$

NOTE. L is any selected length of arc (in mm) including an arc of one pitch, but is not greater than $\pi d_2/2$.

8. Normal backlash

To allow for expansion of the wormwheel at working temperature, backlash has to be present at workshop temperature. Backlash values based on a temperature of 20 °C shall be within the limits given in table 5. These values shall be achieved by adjusting the wormwheel tooth thickness.

9. Load capacity

9.1 Determination of load capacity. The permissible torque for a pair of worm gears is limited either by consideration of surface stress (conveniently referred to as 'wear') or bending stress (referred to as 'strength'), in both the worm threads and wormwheel teeth. Consequently the load capacity of a pair of gears shall be determined using four calculations, concerned with wear and strength of worm and wormwheel (see 9.2.1 and 9.2.2). The permissible torque on the wormwheel shall be the least of the four values.

Table 5. Normal backlash

Grade	Minimum	Maximum
	mm	mm
1	$0.00025m(z_2 + 7) + 0.013$	$0.00025m(z_2 + 20) + 0.05$
2	$0.00025m(z_2 + 7) + 0.013$	$0.00025m(z_2 + 40) + 0.05$
3	$0.00025m(z_2 + 15) + 0.025$	$0.00025m(z_2 + 50) + 0.06$
4	$0.00025m(z_2 + 15) + 0.038$	$0.00025m(z_2 + 65) + 0.10$
5	$0.00025m(z_2 + 15) + 0.038$	$0.00025m(z_2 + 80) + 0.15$

NOTE 1. Any of the grades of backlash may be associated with any class of gear at the discretion of the designer, e.g.:

- grade 1 would be suitable where backlash tolerances relate to critical requirements, such as timing gears and some control mechanisms;
- grade 2 is applicable to precision drives of machine tools;
- grades 3, 4 and 5 are suitable for industrial applications, grade 5 providing adequate backlash for working temperatures of the order of 120 °C.

Axle transmission gears would normally be grade 4 or 5.

NOTE 2. For a worm meshing with a bronze wheel in a cast iron or steel case, the backlash (in mm) between worm and wormwheel is reduced by approximately

$$\frac{3.42m(z_2 + 7)(T - 20)}{10^6}$$

Where bronze or aluminium cases are used, the reduction in backlash will be less than the above.

9.2 Steady load conditions

9.2.1 Permissible load for wear. For the normal rating, the permissible torque, M , (in N·m) on the wormwheel shall be limited by wear to the lower of the following values:

- (a) $0.00191X_{c,1}\sigma_{cm,1}Zd_2^{1.8}m$
- (b) $0.00191X_{c,2}\sigma_{cm,2}Zd_2^{1.8}m$

where

d_2 is the reference circle diameter of the wormwheel (in mm);

m is the axial module (in mm);

$X_{c,1}$ and $X_{c,2}$ are the speed factors for wear on the worm and wormwheel respectively;

$\sigma_{cm,1}$ and $\sigma_{cm,2}$ are the surface stress factors on the worm and wormwheel respectively;

Z is the zone factor as given in table 6.

NOTE 1. The speed factor for wear X_c , corresponding to the combination of rotational speed and rubbing speed, is as given in figure 7. The rubbing speed v_s (in m/s) is given by:

$$v_s = 0.0000524d_1n_1 \sec \gamma$$

and

$$v_s = 0.0000524mn_1\sqrt{(z_1^2 + q^2)}$$

where

d_1 is the reference circle diameter of the worm (in mm);

m is the axial module (in mm);

γ is the lead angle of worm thread;

n_1 is the speed of the worm (in r/min);

z_1 is the number of threads in the worm;

q is the diameter factor.

NOTE 2. The surface stress factor σ_{cm} , corresponding to the combination of materials used, is as given in table 7.

9.2.2 Permissible load for strength. For the normal rating, the permissible torque, M , (in N·m) on the wormwheel shall be limited by strength to the lower of the following values:

- (a) $0.0018X_{b,1}\sigma_{bm,1}ml_{f,2}d_2 \cos \gamma$
- (b) $0.0018X_{b,2}\sigma_{bm,2}ml_{f,2}d_2 \cos \gamma$

where

$X_{b,1}$ and $X_{b,2}$ are the speed factors for strength on the worm and wormwheel respectively;

$\sigma_{bm,1}$ and $\sigma_{bm,2}$ are the bending stress factors on the worm and wormwheel respectively;

$l_{f,2}$ is the length of root of the wormwheel teeth (in mm);

d_2 is the wormwheel reference circle diameter (in mm);

m is the axial module (in mm);

γ is the lead angle of worm thread.

NOTE 1. The speed factor X_b , corresponding to the rotational speed only, is as given in figure 8.

NOTE 2. The bending stress factor, σ_{bm} , corresponding to the material used, is as given in table 7.

NOTE 3. The continuous rating of worm gearing operating in still air may be limited by considerations of temperature rise rather than by resistance to bending or surface stresses. Where a working oil temperature of 95 °C is exceeded, the maximum permissible torque on the wormwheel may be less than that indicated in 9.2.1 and 9.2.2.

9.2.3 Power. Subject to the provisions of clause 10, the normal power rating (in kW) of the wormwheel is given by:

$$\frac{Mn_2}{9550}$$

where

M is the smallest of the four values obtained in accordance with 9.2.1 and 9.2.2.

9.2.4 Expected life. The permissible power or wormwheel torque obtained in accordance with 9.2.1, 9.2.2 and 9.2.3 shall apply to an expected total running life of 26 000 h. Where worm gears are designed on the basis of any other life, the permissible power or wormwheel torque shall be multiplied by the following expressions:

(a) for wear:

$$\left(\frac{27000}{1000 + H_{ec}} \right)^{1/3}$$

(b) for strength:

$$\left(\frac{26200}{200 + H_{eb}} \right)^{1/7}$$

where

H_{ec} and H_{eb} are the total equivalent running times (in h) for wear and strength respectively, corresponding to the expected life (see figure 9).

Unless otherwise specified, the normal rating (e.g. the continuous load capacity of the motor) shall apply continuously throughout the expected total running time.

9.2.5 Factor of safety. The factor of safety is given by:

$$\frac{\text{Tensile strength of worm or wormwheel material (whichever is the less)}}{\text{Working stress}}$$

where the working stress (in MN/m²) is given by

$$\frac{M_{act}}{0.0018 m l_{f,2} d_2 \cos \gamma}$$

9.3 Determination of equivalent running time for loading conditions other than a steady load

9.3.1 Cycles of different uniform loads at different speeds. If the load cycle on the wormwheel comprises a maximum torque M_1 acting for a period H_1 at a mean speed $n_{2,1}$ and smaller torques M_2, M_3 , etc. acting for periods H_2, H_3 , etc. at mean speeds $n_{2,2}, n_{2,3}$, etc., the equivalent running time per cycle for wear at torque M_1 and speed $n_{2,1}$ is given by the following equation:

$$H_c = H_1 + H_2 \frac{(n_{2,2})(M_2)^3}{(n_{2,1})(M_1)^3} + H_3 \frac{(n_{2,3})(M_3)^3}{(n_{2,1})(M_1)^3} + \dots \text{etc.}$$

where each term represents the equivalent running time for the corresponding part of the cycle.

The total equivalent running time for wear at torque M_1 and wormwheel speed $n_{2,1}$ is then given by:

$$H_{ec} = H_c \times \text{number of complete cycles expected during the life of the gears.}$$

The equivalent running time per cycle for strength at torque M_1 and wormwheel speed $n_{2,1}$ is then given by the following equation:

$$H_b = H_1 + H_2 \left(\frac{n_{2,2}}{n_{2,1}} \right) \left(\frac{M_2}{M_1} \right)^7 + H_3 \left(\frac{n_{2,3}}{n_{2,1}} \right) \left(\frac{M_3}{M_1} \right)^7 + \dots \text{etc.}$$

where each term represents the equivalent running time for the corresponding part of the cycle.

The total equivalent running time for strength at torque M_1 and wormwheel speed $n_{2,1}$ is then given by:

$$H_{eb} = H_b \times \text{number of complete cycles expected during the life of the gears.}$$

9.3.2 Irregular variation of load (variable speed).

The equivalent running time for a period of the load cycle specified in 9.3.1 during which load and speed vary other

than uniformly shall be determined by subdividing it into periods during none of which the torque changes by more than $M_1/4$. The equivalent running time for wear and strength corresponding to each such period on the basis of the mean load and the mean speed during the period shall be calculated in accordance with 9.3.1.

9.3.3 Uniform variation of load (variable speed). If during any period w of the load cycle specified in 9.3.1 the torque changes uniformly from $M_{w,1}$ to $M_{w,2}$ at mean speed $n_{2,w}$, the expression for the equivalent running time for that period for wear at torque M_1 and speed $n_{2,1}$ is given by the expression:

$$H_{c,w} = \frac{H_w}{4} \left(\frac{n_{2,w}}{n_{2,1}} \right) \left(\frac{M_{w,1}}{M_1} + \frac{M_{w,2}}{M_1} \right) \times \left[\left(\frac{M_{w,1}}{M_1} \right)^2 + \left(\frac{M_{w,2}}{M_1} \right)^2 \right]$$

The equivalent running time for strength at torque M_1 and speed $n_{2,1}$ is given by:

$$H_{b,w} = \frac{H_w}{8} \left(\frac{n_{2,w}}{n_{2,1}} \right) \left(\frac{M_{w,1}}{M_1} + \frac{M_{w,2}}{M_1} \right) \times \left[\left(\frac{M_{w,1}}{M_1} \right)^2 + \left(\frac{M_{w,2}}{M_1} \right)^2 \right] \times \left[\left(\frac{M_{w,1}}{M_1} \right)^4 + \left(\frac{M_{w,2}}{M_1} \right)^4 \right]$$

NOTE. The requirements of 9.3.1, 9.3.2 and 9.3.3 will also be applicable if the speed is uniform, in which case $\frac{n_{2,2}}{n_{2,1}}$, or $\frac{n_{2,w}}{n_{2,1}}$, etc. become unity.

9.3.4 Load varying in a cycle that is repeated a whole number of times in one revolution of the wormwheel.

If a whole number of load cycles are completed in exactly one revolution of the wormwheel, the gears shall be designed to transmit the maximum torque continuously for an equivalent time equal to the expected running life of the gears.

9.4 Momentary overload capacity. A momentary overload shall be considered as one whose duration is too short to be defined with certainty but does not exceed 15 s.

Worm gears shall be capable of transmitting occasional momentary overloads (in N·m) not exceeding values calculated from the following expressions:

$$(a) \text{ for wear: } 0.00382 \sigma_{cm,2} Z d_2^{1.8} m$$

$$(b) \text{ for strength: } 0.004 \sigma_{bm,2} l_{f,2} d_2 m \cos \gamma$$

9.5 Vehicle axle transmission gears

9.5.1 The torque (in N·m) transmitted by an axle transmission wormwheel is given by the following expression:

$$M_e R_1 R_2$$

where

M_e is the maximum engine torque (in N·m);

R_1 is the gearbox ratio giving lowest road speed;

R_2 is the rear axle ratio.

9.5.2 The roadwheel torque (in N·m) necessary to cause wheel spin/skidding when the vehicle is fully laden is given by the following expression:

$$W_1 \mu R_t$$