Technical Data

Note

Along with the union of ISO standard from JIS standard, most of JIS standard (including Technical report) is being revised and replaced.

In due time JIS standard and JGMA standard (Japan Gear Manufactures Association) for the gears shall be revised to a new edition. However JIS and JGMA standards are not complete and some standards have since been abolished when we started the new edition of KG catalogue. However old JIS and JGMA standards are essential reference of gears for KG-new catalogue.

Therefore we had adopted the latest JIS and JGMA standard in our new edition KG catalogue. However if we found inexplicability and nonexistence sentences, we use both the old and new standards for our new edition KG catalogue.

With respect to the new edition of ISO, JIS and JGMA standards, some parts of new edition KG catalogue are unable to adopt the latest revised ISO, JIS and JGMA standards. We seek your understanding for our latest edition of KG catalogue.

Chapter 1 Basic Knowledge of gear

1.1 Gear history

There is no literature reference concerning the origin of Gears. Who was the first person to use it or pioneered gear development.
Ancient people probably fabricated gears by making notches or projection at external of wooden disk for farm work or to ladle water.
We are able to find literature on gears to its origin when the author, Aristotle (Before Christ 384-322) had written 「Subject on Machine」 about 2,300 years ago.

Approximately 500 years ago, Leonardo da Vinci (1422-1519) had left several gear sketches. He drew almost all of the variety of gear currently used. (Refer to Fig. 1)
Changing with the times, present demand for precision and high strength of the gears are popular. The general public today has a fairly wide usage of the gear. The demand for high quality is as follows:

(1) Changing the module can provide a wide range of transfer power.
(2) Rotation and angle can be transmitted securely.
(3) Changing number of teeth can obtain more flexible gear ratios.
(4) Different varieties of axes position such as Parallel, Crossed, Non-parallel and Non-intersecting can be used for different designs.
(5) Conversion of rectilinear from the rotary motion or vice versa are simply.

The gear also has many other features.

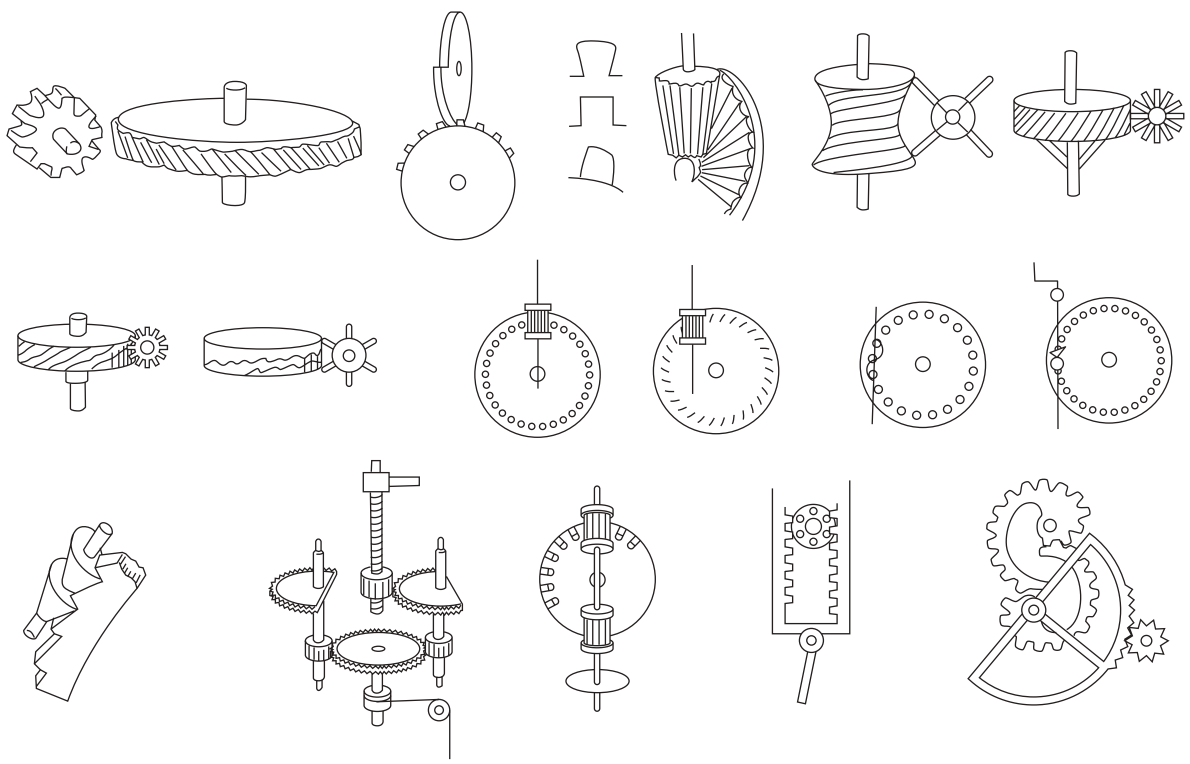


Fig. 1 Sketches of the gears by Leonardo da Vinci

1.2 Types of Gear

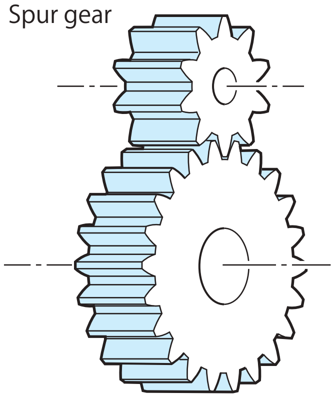
The gear has been classified through a lot of methods. The classification that is most popular is through the different position of the gear axis. Another way is through method of manufacturing, material and Tooth profile.
This time we introduce the methods through different position of gear axis and manufacturing.

Classifications by the position of gear axis

(1) Parallel axis gear (The teeth are parallel to axis)

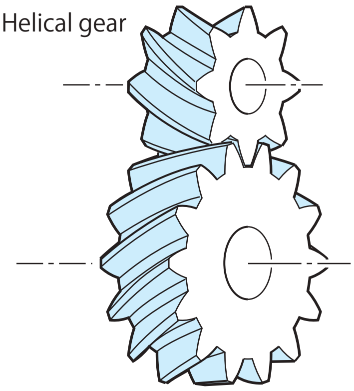
a) Spur gear
This is Cylindrical gear. Tooth trace is parallel to axis. This is a highly demanded gear, which is easy to manufacture and to assemble.

Spur gear



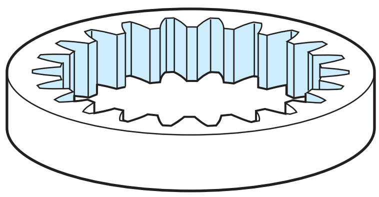
b) Helical gear
This is a Cylindrical gear. Tooth trace has helix curve. Helical gear provides more strength, less oscillation and lower noise level compared with Spur gears. However Helical gear provides a thrust load to axis direction. If changed rotating direction, thrust load is reversed.

Helical gear



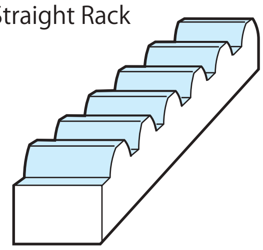
c) Internal gear
This is a cylindrical gear ring with teeth formed at the inner diameter. The most popular demand for Internal gears is used for mechanism of planetary gear train. There are two types of Tooth trace, one is parallel and the other is helix to axis. However gear with parallel axis has higher demand.

Internal gear



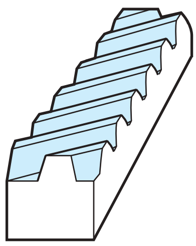
d) Straight Rack
It is thought that radius of spur gear grew infinite to become a straight line. It can be matched with Spur gear to convert between the rectilinear motion and the rotary motion.

Straight Rack

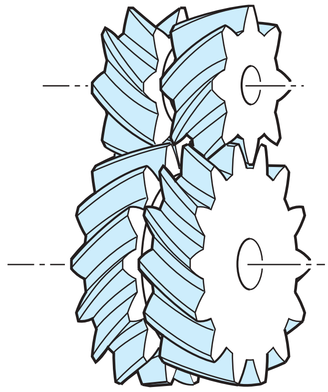


e) Helical Rack
It is thought that radius of Helical gear grew infinite to become a straight line and Tooth trace is also straight line. It can be matched with Helical gear to convert between the rectilinear motion and the rotary motion.

Helical Rack



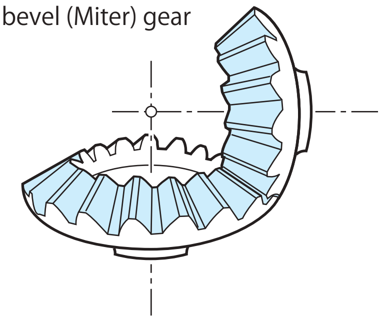
f) Double helical gear
The shape looks like two Helical gear joined together. Therefore this type of gear does not have thrust load to axis during operation.


Double helical gear

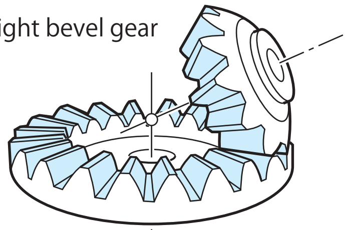
*Other types of gear designs used for parallel axis such as non-circular gear and Eccentricity gear are available but omitted this time.

(2) Intersecting axis gear

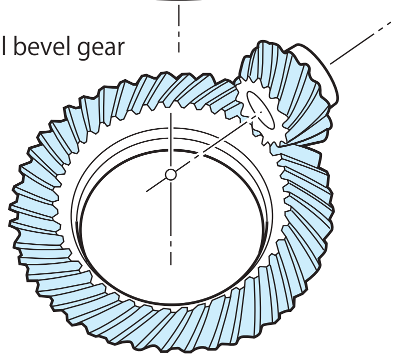
g) Straight bevel (Miter) gear
This is gear with Tooth trace, which is a straight line in the same direction as surface element of Pitch cone. Miter gear has shaft angle of 90° and gear ratio of 1:1.


Straight bevel (Miter) gear

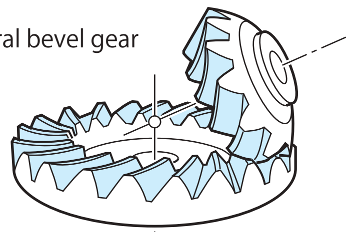
h) Angular straight bevel gear
Angular straight bevel gear which does not have shaft angle of 90°.


Angular straight bevel gear

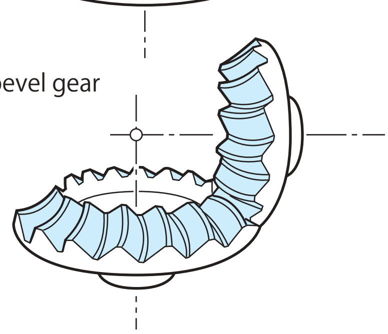
i) Spiral bevel gear
Tooth trace is described as a curve with spiral angle. Spiral bevel gear has advantage over Straight bevel gear for gear strength, oscillation and noise level. Disadvantage of Spiral bevel gear is axial thrust load. Therefore proper bearing location and firm support are needed.


Spiral bevel gear

j) Angular spiral bevel gear
Angular spiral bevel gear does not have shaft angle of 90°.

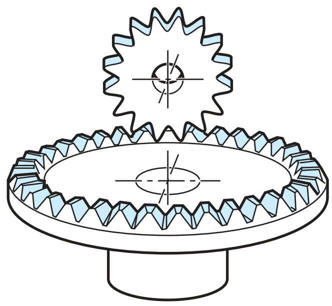

Angular spiral bevel gear

k) Zerol® bevel gear
This is Zerol® bevel gear, similar to Spiral bevel gear with zero spiral angle. The Tooth trace is described as a curve with spiral angle of zero degree. (Occasionally, spiral angle 10° or below are also called Zerol® bevel gear.)
The force of tooth action is the same as Straight bevel gear.
(Ⓡ mark is Gleason Works trademark)


Zerol® bevel gear

l) Face gear

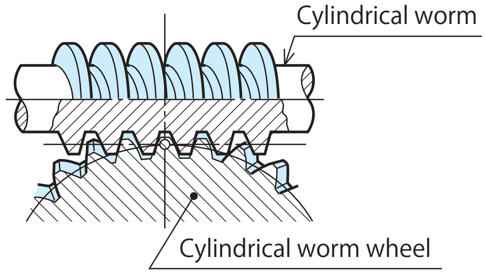
is Face gear. A Toothed disk gear, can be matched with Spur or Helical gear. There are two types of Face gear with shaft angle 90°, intersecting axis and Non intersecting axis.



Face gear

(3) Skew gear (Non Intersected Gear)

Cylindrical worm



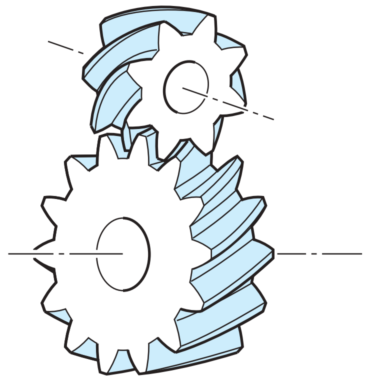
m) Cylindrical worm gear

This is a Worm gear pair consisting of Cylindrical worm gear and Worm wheel.

Meaning of Cylindrical worm gear is that the thread has one or more starts.

This Worm gear pair provides high speed reducing ratio and low noise level. Disadvantages are low efficiency and generation of heat.

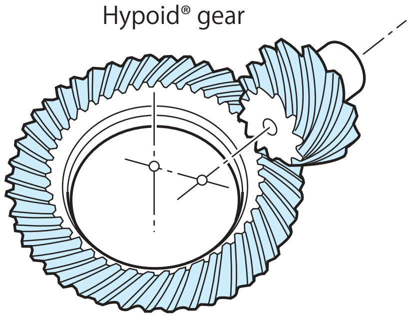
Crossed helical gear (Screw gear)



n) Crossed helical gear (Screw gear)

This is a Gear pair for transmission between Non-parallel and Non-intersecting axes of Cylindrical gears as pair of Helical or Spur gear. Due to spot contact in theory, design of transmission should be light load.

Hypoid® gear



o) Hypoid® gear

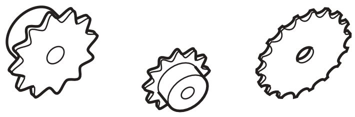
This is a gear for transmission between Non-parallel and Non-intersecting axis of conical gear. This gear is similar to Spiral bevel gear. Most popular usage is for Deferential gear for automotive.

(Ⓡ mark is Gleason Works trademark)

*Others including Enveloping worm, Spiroid® and Helicon gear types of gear designs are available for Non-parallel and Non-intersecting axis but omitted this time. (Ⓡ mark is ITW trademark)

(4) Machine elements compared with gear for similar shape and purpose of usage.

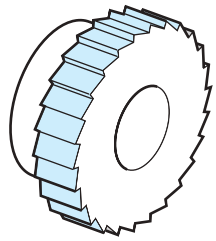
Sprockets



p) Sprockets

This is Sprocket wheel used for matching with bushed chain and Ladder chains. Usage is for transmitting a power over long distances between axes.

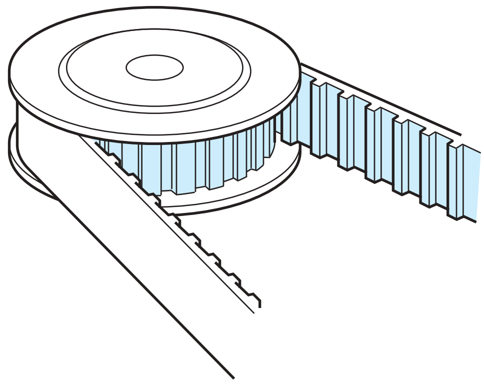
Ratchet gear



r) Ratchet gear

This is Ratchet gear, which looks like the teeth of saw formed at external wheel used for positioning (indexing) and preventing inversion.

Timing Pulley



s) Timing Pulley

This is Timing pulley using matching timing belt (belt with teeth). Usage is for transmitting power over long distance between axes.

Classifications by different manufacturing methods.

(1) Machined gear

Tooth of Spur and Helical gears are machined, method of manufacturing are hobbing machine, gear shaper, gear plainer and other machining. For mass production, method of broaching is used. As for Bevel gear process, dedicated cutting machine is usually used. However hobbing machine is rarely used.

(2) Shaving gear

This is a shaving gear that shaves out the minute finishing layer on the Tooth flank by shaving machine and its cutter.

(3) Ground gear

Grinding machine is used for making ground tooth flank. Classified as two methods of Form grinding and General grinding. Both processes use grinding wheel (Diamond and CBN). To obtain high quality gear, accurate lathing of the gear is necessary. The electrolytic ground gear has been developed recently but explanation will be omitted this time.

(4) Precision cold rolled processed gear

This is a gear done using plastic working (cold rolling) to form tooth by compression. KG Worm gears of module 0.5 to 2.0 are manufactured by using the Cold rolling forming method. The fabrication of Cold rolling-Worm gear is formed by rotating the hydraulic compression which causes the teeth to rise on both sides of the rolling tooth machine. The Tooth flank of Worm gear has glassy finish like mirror.

(5) Injection molded gear.

This is a gear formed by injecting molten plastic into a mold and applying pressure for a fixed duration.

(6) Sintering Gear

This is Sintering gear. Metallic powder is put into the mold before applying pressure and heat to gear mold to harden. Occasionally there are re-pressuring and re-sintering. Impregnation of oil, heat treatment and surface treatment can also be applied after the 1st sintering process. Sintering gear is suitable for mass production.

* On the other hand, we have the gears with pressed, forged or EMS (Electric Spark Machine and Wire Cut Sparking Machine) but omitted this time.

Refer to Table 1 for an estimate of System of accuracy depending on difference in gear processing methods used for economical production.

Table 1.
Comparison table of System of accuracy between different gear manufacturing methods (Confirmed by JIS B 1702-1995)

Fabrication and Distinctive heat treatment / System of Accuracy of JIS B1702 (old)		0	1	2	3	4	5	6	7	8
Non-Quenching gear	Hobbing					←	—	—	—	→
	Shaving		←	—	—	—	→			
Quenched gear	Hobbing							←	—	→
	Shaving				←	—	—	→		
Grinding		←	—	—	—	→				

1.3 Types of Tooth profiles curves

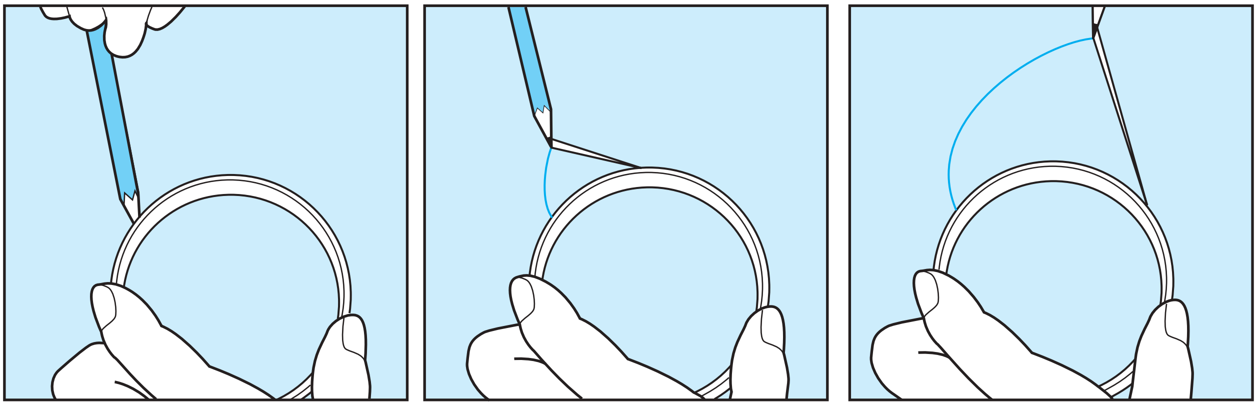
(1) Involute tooth profile

The Involute curve can be seen when unlacing the end of firm string from Cylinder. The end of firm string makes the curve.

Involute tooth profile uses part of Involute curve for Tooth profile. The Cylinder with string is called Base circle. (Refer to Fig. 2)

Involute tooth profile is as follows,

* Fabrication of accurate tooth profile, which is easily measured.
  (Also made into a cutting tool easily.)
* It matches with various gear ratio, any modified Invoute profile gear and compatible with other gear profiles.
* Can obtain the transferable correct engagement even if Centre distance has minor deviations. Due to these features of Involute tooth profile, it is widely used.



▲ The line drawn from the starting point to end of tensioned string after unlacing from the external of the cylinder is defined as an Involute curve.

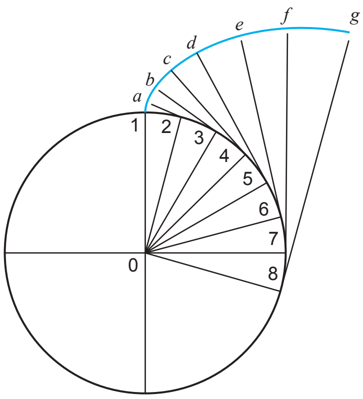


Fig. 2 Involute curve

(2) Cycloid tooth profile.

As shown in Fig 3, medium circle turns around the external of the Base circle. Trace the start (from point of contact) to end point of the medium circle; this is (a-b) Epicycloid. Do the same for the small circle in the internal of the Base circle; this is (a-b’) Hypocycloid. Part of b-b’ will be used for the Cycloid tooth profile.

To obtain the same wear off on the entire tooth, Cycloid tooth profile is used for gears for instruments and time-piece. However Cycloid tooth profile is rarely used for power transmission due to difficulty in production.

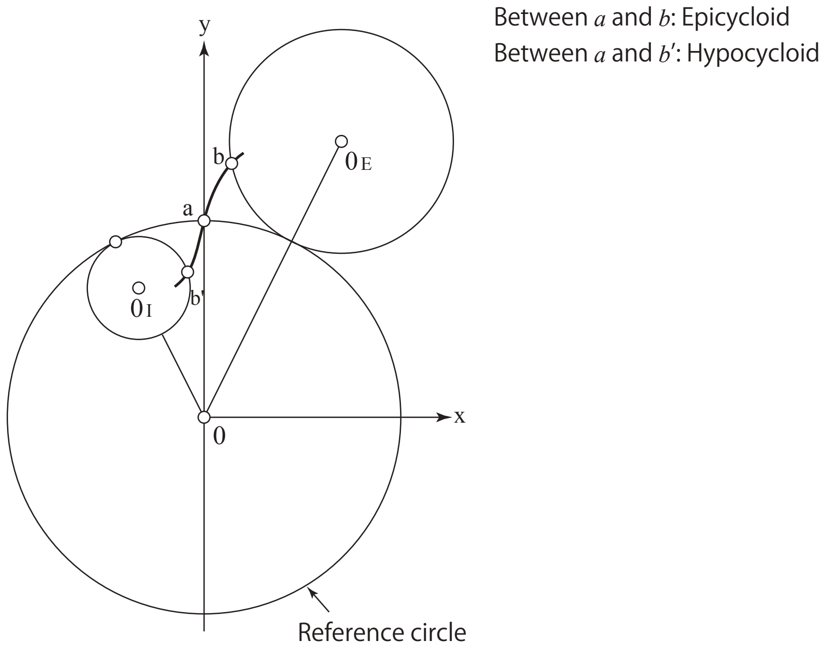


Fig. 3 Cycloid tooth profile

(3) Arc of circle profile

As for the Tooth profiles, they are classified into single arc and compound arc. It has an advantage for a slow wearing off due to uneven contact with arc. Generally Single arc and Compound arc tooth profiles are not generally used because Tooth profile is not efficient due to difficulty in production as compared with Involute tooth profile.

This is WN gear (Wildhaber · Novikov) remarkably different theory of production compared with Tooth profile of Arc circle in the past.

The Tooth profile of WN gear does not abide by the science of mechanism in the past as it uses spot contact. However a design change to the Helical gear enables it to transfer a point of load to Tooth trace direction realizing a transferable uniform rotated motion.

Depending on the purpose of usage, the Tooth profile of WN is superior over Involute. However this is for very special cases only. The Tooth profile of WN Helical gear is difficult to produce as compared with Involute tooth profile (Refer to Fig. 4).

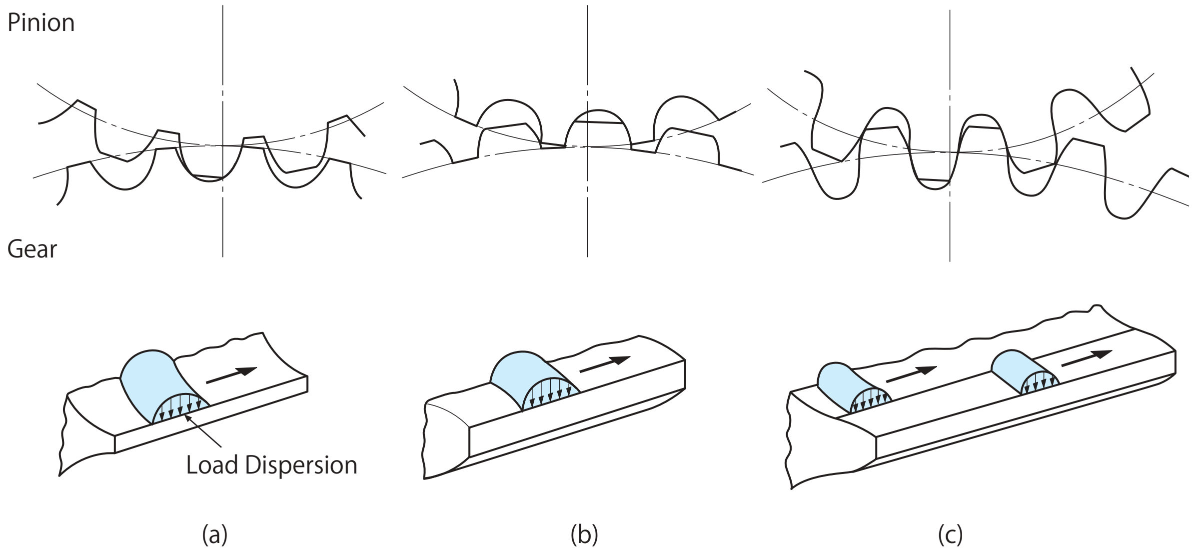


Fig. 4 Types of Tooth profile for Novikov Gear

(a) Arc circle of Pinion is convex
Arc circle of Gear is concave

(b) Arc circle of Pinion is concave
Arc circle of Gear is convex

(c) Arc circle of both pinion and gear are convex at the Addendum flank.
Arc circle of both pinion and gear are concave at the Dedendum flank.

1.4 Terminology of each part of the gear

Terminologies of gears are defined in **JIS B 0102:1999** Vocabulary of gear terms-Related to geometry.
The vocabularies for gear numerical formula and gear drawings are defined in **JIS B 0121:1999** International gear notation - Symbols for geometrical data.
Comparison table 2 for common gear terms of **JIS B 0102:1993**(old) and **JIS B 0102:1999** are confirmed. The names are changed but meanings are retained.

Table 2. Comparison between new and old for gear terms.

JIS B 0102:1999	JIS B 0102: 1993 confirmed
Reference circle (1)	Reference pitch circle
Reference diameter	Reference pitch diameter
Tooth depth	Tooth depth
Tooth thickness	Circular tooth thickness
Working depth	Working depth
Standard basic rack	Basic Rack (2)
Datum line of Rack	Pitch line for rack
Virtual cylindrical gear of Bevel gear	Virtual spur gear for Bevel gear
Pitch angle	Pitch angle
Tip angle	Tip cone angle
Root angle	Root angle
Spiral angle for Bevel gear	(Bevel gear) Spiral angle
Locating distance for Bevel gear	(Bevel gear) Location distance
Centre distance modification coefficient	Coefficient of increment Centre distance

Note (1) Pitch diameter is stipulated in **JIS B0102:1999**. Reference circle is classified with Pitch circle. Pitch circle is diameter of geometrical circle for gear described by moment of relative motion of axis with mating gear.

Note (2) Definition of Basic rack is "imaginary rack with Standard basic rack" under the Normal section in **JIS 0102:1999**

In addition, the gears terms have been updated but not outlined.

Standard basic rack tooth profile

Rack tooth profile is stipulated in **JIS B 0102:1999**, Standard tooth profile dimension in Involute tooth profile group. Therefore the gear and dimensions of tool are established while compatibility is kept.
The details of Standard basic rack tooth profile are shown in Fig. 5 and Table 3. According to the **JIS B1701-1:1999 Involute tooth profile Article 1: Standard basic rack tooth profile** and **it's recommended attached supplement articles adds Tooth profile and Usage of Basic rack** for reference, which is ommitted here.

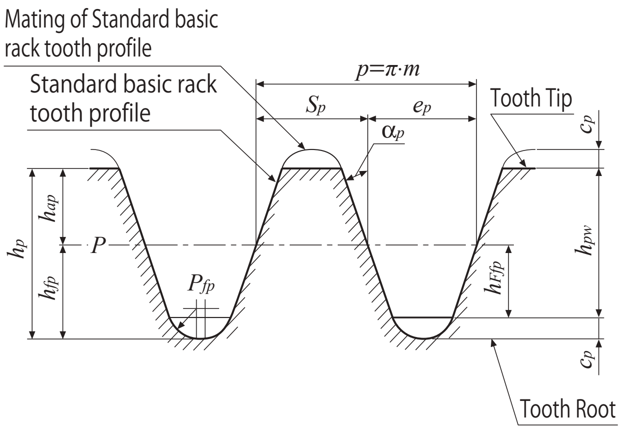


Fig. 5 Standard basic rack tooth profile and Mating of Standard basic rack tooth profile

Table 3. Dimensions of Standard basic rack

Vocabulary	Dimension of Standard basic rack
α_p	20°
h_{ap}	1.00mm
C_p	0.25m
h_{fp}	1.25m
P_{fp}	0.38m

Gear terms and Vocabularies for Involute gear

(JIS B 0102:1993 confirmed and extracted from JIS B 0121:1999)

Fig. 6 indicates the names (gear terms) for parts of Tooth profile.

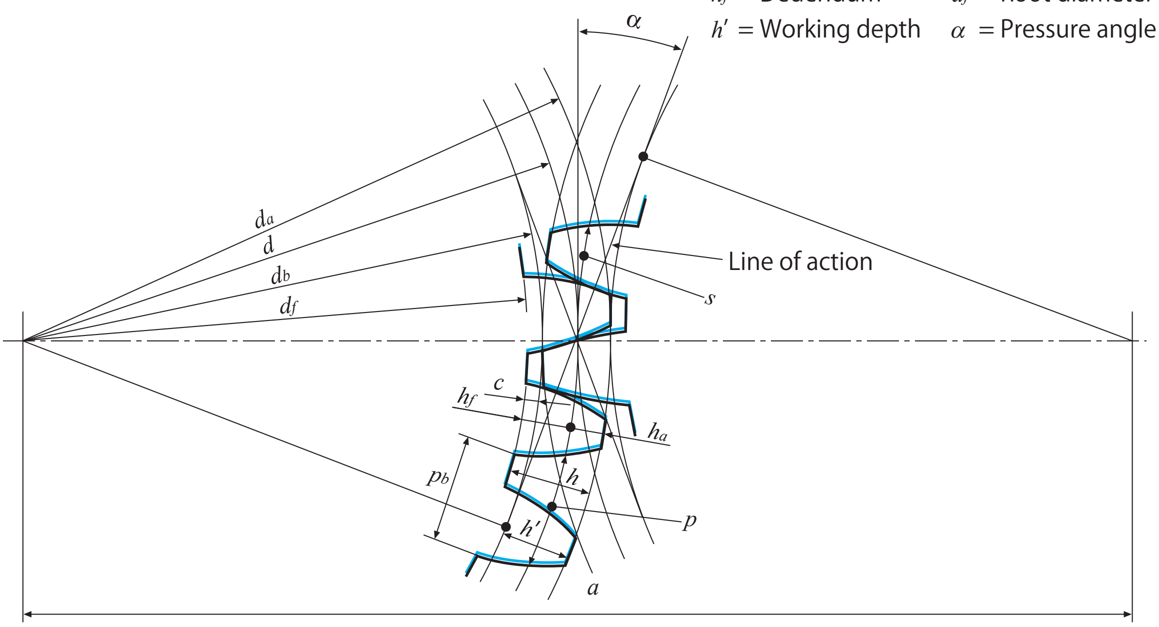
Standard is a defined term of an applicable limited word from Reference surface of gear, defined in **JIS B 0102:1999**. Normally "Standard" and "Working" are distinguished. When it is not necessary to classify between Standard and Working, it is common knowledge that the word "Standard" can be omitted.

Centre distance—a	Centre distance is defined as the shortest distance between axes of Parallel gear pair or pair of crossed gear. Reference centre distance is defined in **JIS B 0121:1999** which is not outlined here.
Circular pitch* — p	Circular pitch is the distance of Pitch between adjacent teeth as measured on the Reference circle or Reference line.
Base pitch* — p_b	Base pitch is perpendicular line to Pitch between any section of Tooth profile in Involute gear.
Tooth depth — h	Tooth depth is radial distance between Tip and Root circle.
Addendum — h_a	Addendum is radial distance between Tip and Reference circle.
Dedendum — h_f	Dedendum is radial distance between Root and Reference circle.
Working depth — h'.	Working depth is distance along the centre line between Tip surface of two engaging gears.
Bottom clearance — c	Bottom clearance is distance along the centre line between Tip surface of a Gear and Root surface of its Mating gear.
Tooth thickness — s.	This is length of Arc on Reference circle between the two profiles of a tooth.
Tip diameter — d_a	This is diameter of Tip circle.
Reference diameter — d.	This is diameter of Reference circle.
Root diameter — d_f.	This is diameter of Root circle.
Transverse line of action	This is normal line common to two Transverse profiles at their point of contact. For Involute gear pairs, the lines of action are also common tangents to their Base circles.
Pressure angle —α.	Angle drawn when centre connection line and profile crosses pitch point upon the reference circle.

The term of ∗-mark is not define in **JIS B 0121:1999**. The Pressure angle is supplemented due to insufficient description in this JIS. An Addendum and Dedendum of Worm wheel is defined for classification of " Reference" and " Mating", which omitted here.

Fig. 6 Name of the each parts of Tooth profile

a	= Centre distance	c	= Bottom clearance
p	= Circular pitch	s	= Tooth thickness
p_b	= Base pitch	d_a	= Tip (Out side) diameter
h	= Tooth depth	d	= Reference diameter
h_a	= Addendum	d_b	= Base diameter
h_f	= Dedendum	d_f	= Root diameter
h'	= Working depth	α	= Pressure angle



1.5 Fundamental dimensions for various sizes of Tooth profile

There are three types of formulas to calculate various sizes of Tooth profile.

1. Module m

Reference pitch divided by π is module, which defines the size of tooth in metric gear. If value of Reference diameter d(mm) divided by Number of teeth z increases, tooth capacity increases proportionately.

$$\text{Module } m = \frac{\text{Reference diameter } d}{\text{No. of teeth } z} \ \text{(mm)}$$

Tip (Outside) diameter is defined as d_a, calculation formula is $m = \frac{d_a}{z+2}$. Refer to Fig. 7 for a full-scale drawing.

2. Diametral pitch P or DP

Diametral pitch is size of tooth expressed in teeth per inch of pitch diameter. Formula of calculation is given as Number of teeth z divided by Reference diameter d (inch). Capacity of tooth profile increases and decreases inversely proportional to the numerical sum.

$$DP = \frac{\text{Number of teeth } z}{\text{Reference diameter } d \text{ (inch)}} \ \text{(An absolute number)}$$

Tip (Outside) diameter defined as d_a,

Calculation formula of $DP = \frac{z+2}{d_a \text{ (in)}}$

There is a relationship between module and Diametral pitch. (Comparison between module and Diametral pitch)

$$DP = \frac{25.4}{m} \qquad m = \frac{25.4}{DP} \ \text{(mm)}$$

3. Circular pitch CP

This is length of centre distance between adjacent teeth divided by arc circle of pitch circle. Calculated by circumference of pitch circle divided by number of teeth.

$$CP = \frac{\text{Circumference of Pitch circle } (\pi \times d)}{\text{Number of teeth } z} \ \text{(mm)}$$

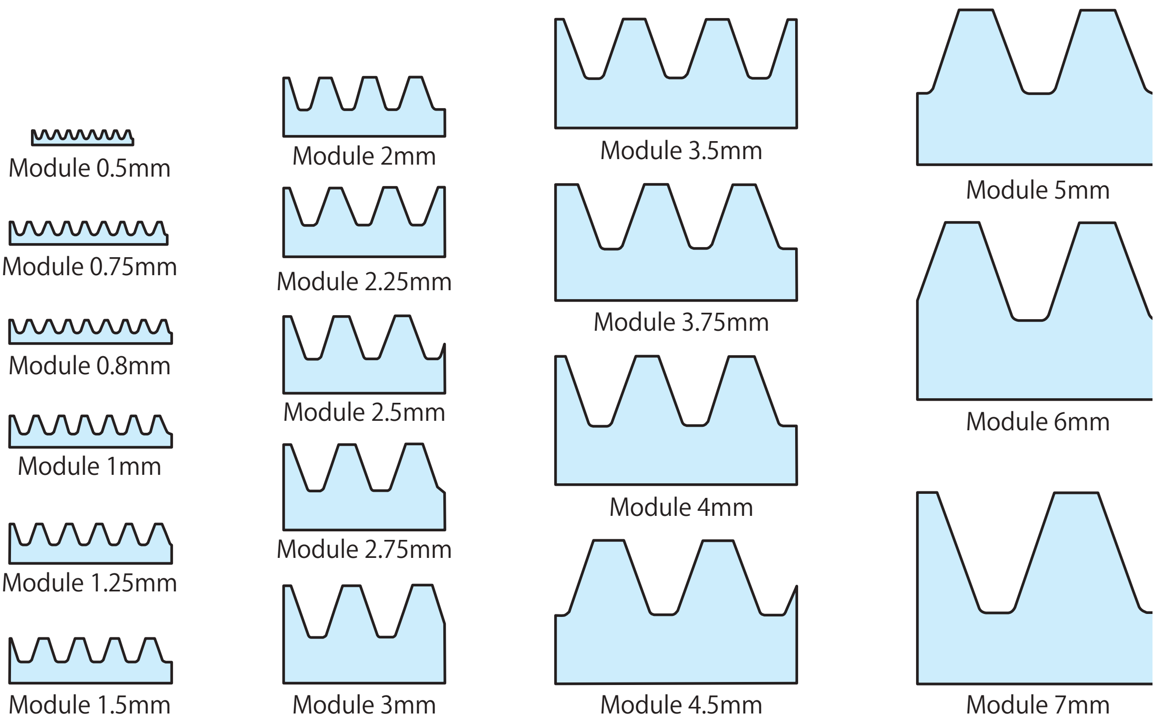


Fig. 7 Full-scale drawing of module

Note that π is ratio of the circumference of a circle to its diameter as π=3.14159 .....

Where Tip(outside) diameter da, calculation of $CP = \frac{\pi \times da}{z+2}$ (mm)

The 3 categories for size of Tooth profile mentioned above are widely used. In particular, Circular pitch CP is used to control traveling distance and positioning.

The standardization of module is shown by the following classification. Introduced in **JIS B 1701-2**: 1999 Cylindrical gear- Involute tooth profile and Article 2-Module and Appendix of the same standard (stipulation). Also shown below is classification not stipulated for Involute tooth profile cylindrical gear below module 1 in **ISO 54**.

Table 4. Standard value for module of Cylindrical gear.

Unit : mm

I	II	I	II	I	II	I	II
0.1		1			5.5	25	
	0.15		1.125	6			28
0.2		1.25			(6.5)	32	
	0.25		1.375		7		36
0.3		1.5		8		40	
	0.35		1.75		9		45
0.4		2		10		50	
	0.45		2.25		11		
0.5		2.5		12			
	0.55		2.75		14		
0.6		3		16			
	0.7		3.5		18		
	0.75	4		20			
0.8			4.5		22		
	0.9	5					

It is advisable to select column-*I* of module (priority selection) as far as possible.

It is not advisable to select the module 6.5 as seen in column-II.

The standardization of module for Bevel gear is shown by the following classification. Introduced in **JIS B 1706-2**: 1999 Straight bevel gear- Article 2-Module and Diametral pitch and Appendix of the same standard (stipulation). Also shown below is classification not stipulated for Straight bevel gear below module 1 in **ISO 678**. However the Diametral pitch is omitted here.

Table 5. Standard value for module of straight bevel gear.

Unit : mm

I	II	I	II	I	II
0.3		1			3.5
	0.35		1.125	4	
0.4		1.25			4.5
	0.45		1.375	5	
0.5		1.5			5.5
	0.55		1.75	6	
0.6		2			(6.5)
	0.7		2.25		7
	0.75	2.5		8	
0.8			2.75		9
	0.9	3		10	

It is advisable to select column-I of module (priority selection) as far as possible.

It is not advisable to select the module 6.5 as seen in column-II.

Table 6. Comparison tables between module and Diametral pitch.

Unit : mm

Module	9	8.467	8	7.257	7	6.35	6	5.08	5	4.233	4
Diametral pitch	2.822	3	3.175	3.5	3.629	4	4.233	5	5.08	6	6.35
Tooth depth	20.25	19.05	18.00	16.33	15.75	14.29	13.50	11.43	11.25	9.52	9.00
Pitch	28.27	26.60	25.13	22.80	21.99	19.95	18.85	15.96	15.71	13.30	12.57

Module	3.629	3.5	3.175	3	2.822	2.54	2.5	2.309	2.25	2.117	2
Diametral pitch	7	7.257	8	8.47	9	10	10.16	11	11.289	12	12.70
Tooth depth	8.17	7.88	7.14	6.75	6.35	5.72	5.63	5.20	5.06	4.76	4.50
Pitch	11.40	11.00	9.98	9.43	8.87	7.98	7.85	7.25	7.07	6.65	6.28

Module	1.814	1.75	1.588	1.5	1.411	1.27	1.25	1	0.8	0.75	0.5
Diametral pitch	14	14.514	16	16.933	18	20	20.32	25.4	31.75	33.867	50.8
Tooth depth	4.08	3.94	3.57	3.38	3.17	2.86	2.81	2.25	1.80	1.69	1.13
Pitch	5.70	5.50	4.99	4.71	4.43	3.99	3.93	3.14	2.51	2.36	1.57

Note that Tooth depth is calculated with Bottom clearance as C = 0.25 mm.

1.6 Features of common gears

Chapter 1.2 covered briefly on types of gear. The main gear features are explained here.

Helical gear

Helical gear has characteristics of transferability of larger load, less vibration and lower noise compared with Spur gear.
However, thrust load (axial direction) occurs due to helix angle. It is therefore necessary to design thrust bearings. (Refer to chapter 2 for thrust load)
When using parallel axis, engage with right and left hand of Helical gears at the same angle.
Using Screw gear for Non-parallel and Non-intersecting axis is called Crossed helical gear.
Roll up a piece of right-angled triangle paper as seen in Fig. 8. The straight lines of slope of the right-angled triangle become a thread curve (helix). Helical gear adopts this curved line as Tooth trace curve.
This right angled triangle unrolls to draw layers of Helical gear to become Fig. 9.

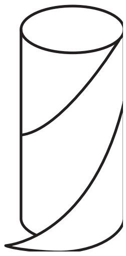


Fig. 8 Helix

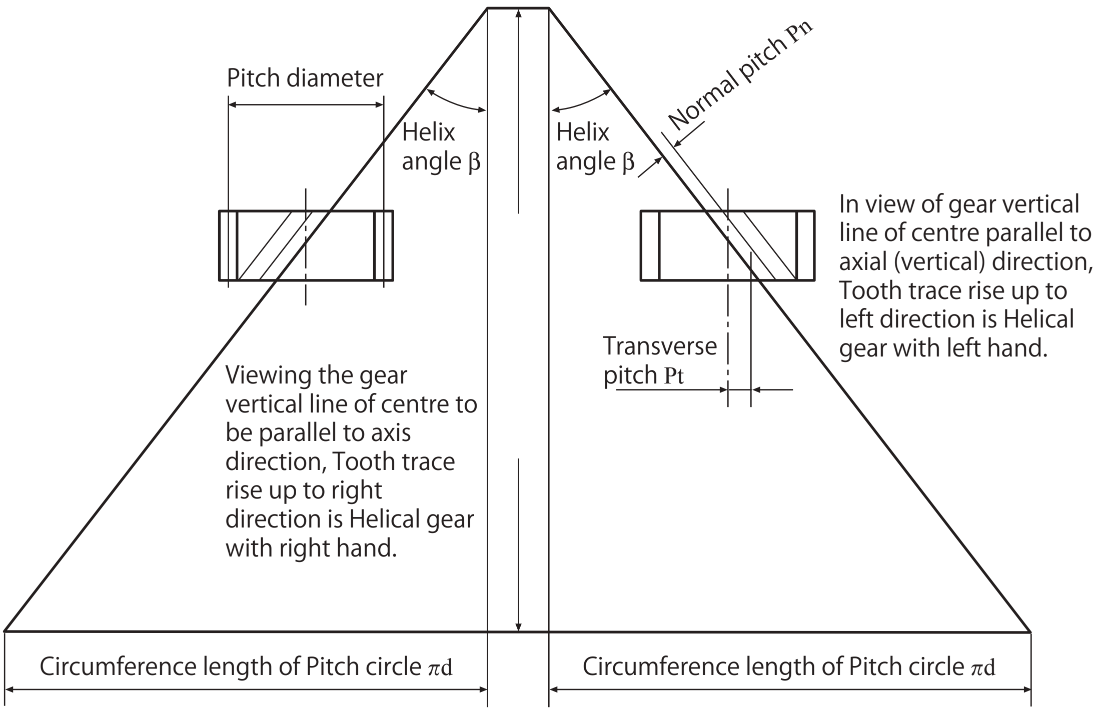


Fig. 9 Helical gear (right and left hands)

There are types of Normal and Axial for Helical gear. Standard tooth profile for Normal type of Helical gear is a section of the Tooth profile perpendicular to Tooth trace of Helical rack which is obtained by setting the Reference pitch radius to infinity.

Standard tooth profile for Axis type of Helical gear is perpendicular to gear axis.

Fig. 10 shows both of the Reference sections.

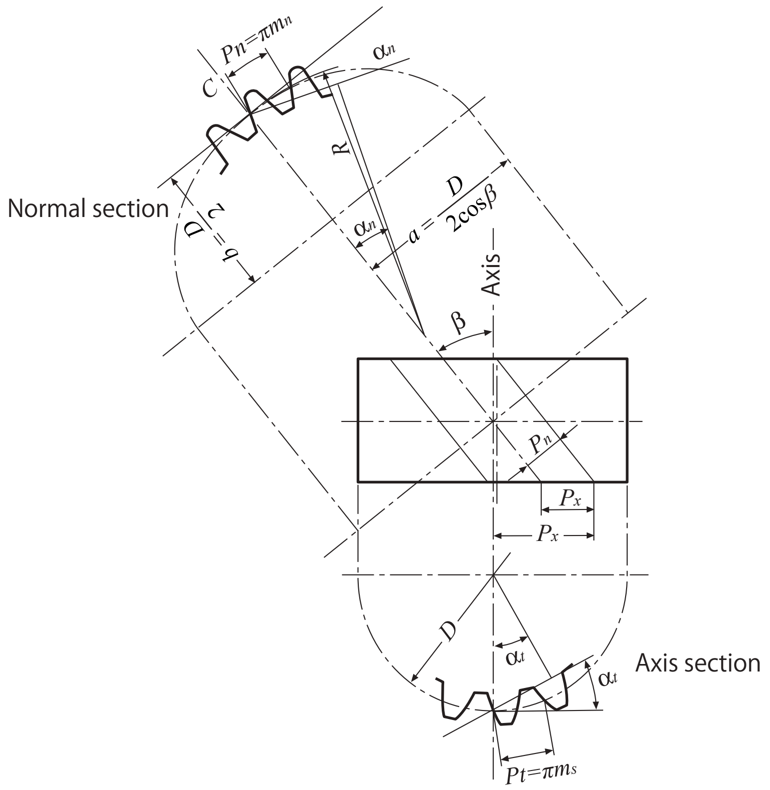


Fig. 10 Types of Normal and Axial

Regarding to manufacturing method, the same hob cutter and grinding wheel can be used to fabricate any helix angle gear. As long as the Normal type of Helical gear has same Normal module m_n (called module of hob cutter) and same Normal pressure angle a_n (called pressure angle of hob). Even if helix angle β was changed

Therefore, stock of hob cutters (tools) and manufacturing cost can be saved making Normal type of Helical gear economic and widely used generally.

However, calculating Centre distance for Normal type of Helical gear, it is necessary to adjust the helix angle to obtain an integer number for Centre distance due to the $\text{cosine}\,\beta$ in the denominator.

Regarding the manufacturing method, the hob cutter (tool) and grinding wheel must be changed to fabricate Axial type of Helical gear if helical angle β is changed. Therefore mass production for this type of gear is very limited.

Since calculation for Helical gear is the same as Spur gear, integer number for centre distance is easily obtained.

Note (1) Adopted old gear terms.

At the above Fig. 10, under perpendicular section to Tooth trace, the Pitch diameter becomes oval. Half of length of oval with major and minor axis is used for calculation below.

$$a=\frac{D}{2\cos\beta} \qquad b=\frac{D}{2}$$

Formula for the radius of curvature R of oval at the C-point is as follows,

$$R=\frac{a^2}{b}=\frac{D}{2\cos^2\beta}$$

Therefore assuming this is a Spur gear with Radius of pitch circle R, it is commonly called (1)Virtual spur gear for Helical gear.

The relation between (1)Virtual number of teeth of Spur gear z_v and actual number of teeth z of Helical gear is as follows.

$$z_v=\frac{z}{\cos^3\beta}$$

(1) The Virtual number of teeth of Spur gear becomes the standard for strength calculation of Helical gear, calculation of profile shifted gear and selection of hob cutter.

(Reference)

The Crossed helical gear (Screw gear) is simply a type of Helical gear. The Parallel helical gear has the same helix angle with opposite helix hand. Where as the Crossed helical gear (Screw gear) is engaged with Non-parallel and Non-intersecting axis with any optional helix angle.

Method of correct engagement, Normal module m_n and Normal pressure angle α_n must be the same.

When two non-profile shifted gears are engaged, each Reference cylinder helix angle are indicated as β_1 and β_2,

Where helix direction of both gears are the same, the formula for shaft angle Σ is ,

$$\Sigma = \beta_1 + \beta_2$$

Where helix direction of both gears are different, formula for shaft angle Σ is,

$$\Sigma = \beta_1 - \beta_2 \quad \text{or} \quad \Sigma = \beta_2 - \beta_1$$

Therefore these become the relation of Shaving cutter and Machined gear.

In Theory, Crossed helical gear (Screw gear) has spot contact and can only take small loads.

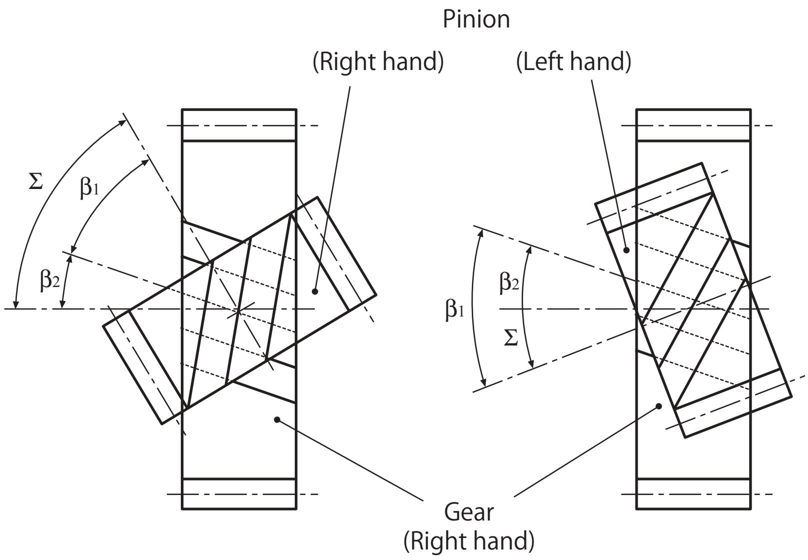


Fig. 11 Engagement of Crossed helical gear (Screw gear)

Bevel gear

This is Bevel gear, that is formed by making gear teeth to the Reference surface of a coned friction wheel. Usage of this conical gear is to transfer power to Crossed or Angular axis of gear. This Reference surface is called Pitch cone of Bevel gear.
Classification by shape of tooth trace that straight Tooth trace to axis direction is called Straight bevel gear. Spiral Tooth trace to axis direction is called Spiral bevel gear.
In Fig. 12, Spur gear with Radius of pitch circle R_{v1} and R_{v2} of Back cone is thought to be Tooth profile of Bevel gear.
This Spur gear is an incomplete circle. The incomplete circle Spur gear after being completed is called a Virtual spur gear [(1)], which is equivalent to a Bevel gear.

The relationship between Virtual number of teeth of Spur gear z_v and Actual number of teeth z of Bevel gear is as follow.

$$z_v = \frac{z}{\cos\delta} \qquad (\delta\text{: Pitch angle})$$

The Virtual number of teeth of Spur gear is standard for strength calculation of Bevel gear and selection of hob cutter.
This is Crown gear where Pitch surface of Bevel gear is changed into a flat surface and perpendicular to axial direction. Using high gear ratio and using creative motion of Bevel gear for examination from imaginary Tooth profile of Crown gear.
Bevel gear with shaft angle of 90° and gear ratio 1:1 is commonly called Miter gear.

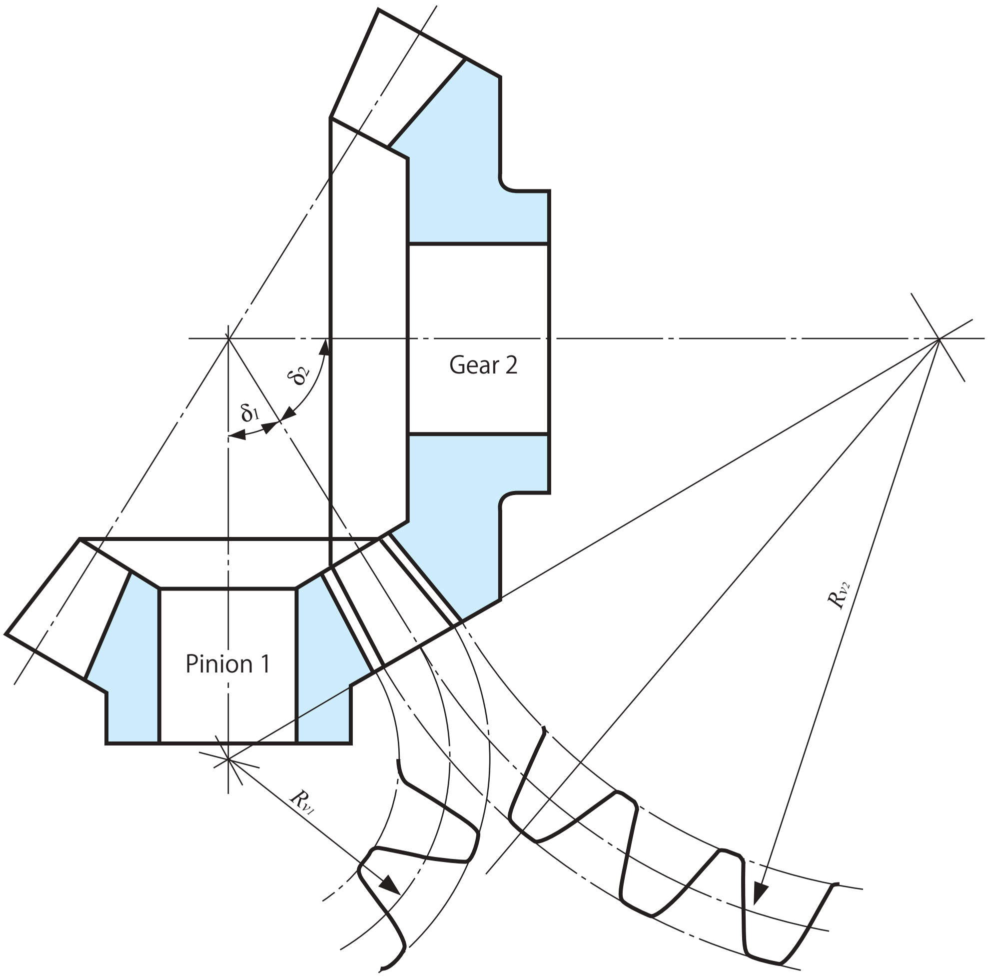


Fig. 12 Virtual spur gear [(1)] for Bevel gear

Note(1) Adopted old gear terms

(1) Straight bevel gear

Straight bevel gear has a straight Tooth trace. Standard straight bevel gear and Gleason system straight bevel gear are common types.
A Standard Straight bevel gear is equivalent to a Standard spur gear due to Virtual spur gear. Undercut occurs in small Number of teeth.
However, Gleason system for Straight bevel gear provides fewer problems of Undercut in small Number of teeth of Pinion because Gleason system for Straight bevel gear is designed to become profile shifted gear between Pinion and Gear. Refer to the below Table 7 for comparison table of the features between Standard system and Gleason system.

Table 7. Comparison table for the features between Gleason system and Standard system

	Gleason system	Standard system
Cause of Undercut	There are fewer problems in Pinion due to positive Rack shift (Gear is negative Rack shift)	It is designed without rack shift and Undercut occurs easily.
Balance of strength for Pinion and Gear	Maintains excellent balance by Rack shift	Unbalanced without Rack shift
Bottom clearance	There is no Tip interference at Toe due to Parallel bottom clearance.	Occur Tip interference at Toe easily due to not Parallel bottom clearance

∗ Miter gear is designed without Rack shift.

Coniflex® gear has Tooth trace with Crowning to Straight bevel gear as named by Gleason company.
Due to above features and Crowning, Gleason Straight bevel gear provides much lesser single contacts and assembly problems as compared to other methods.
Table 8 classifies the minimum Number of teeth to prevent Undercut for Gleason Straight bevel gear.

Table 8. Classifies the minimum Number of teeth to prevent Undercut for Gleason Straight bevel gear.

α=20°		α=14.5°	
Number of teeth of Pinion	Number of teeth of Gear	Number of teeth of Pinion	Number of teeth of Gear
z_1	z_2	z_1	z_2
13	30	24	57
14	20	25	40
15	17	26	35
16	16	27	31
		28	29
		29	29

(2) Spiral bevel gear

Fig. 13, angle between Tooth trace and Pitch cone surface element in Bevel gear with curved Tooth trace is called Spiral angle.
Mean spiral angle βm is spiral angle at centre of Facewidth. Unless otherwise specified, this Mean spiral angle is commonly called spiral angle.
For Gleason system of Spiral bevel gear, Standard spiral angle is 35° with arc of Tooth trace. Gleason system cutter performs to produce Crowning at Tooth trace automatically.
In general, Shaft angle is 90° and matches with left and right hand gears.
Refer to Fig. 14 for Right and Left hand of Spiral gear.
To prevent thrust force to axis direction, due to curved tooth, thrust bearing is necessary. (Refer to the thrust force in Chapter 2)
Shown in Table 9. Comparison table for minimum Number of teeth to prevent Undercut for Gleason Spiral bevel gear.

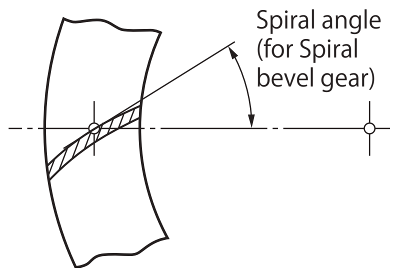


Fig. 13 Spiral angle at centre of Facewidth.

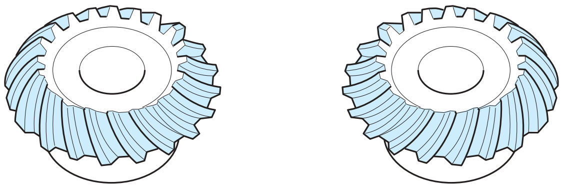


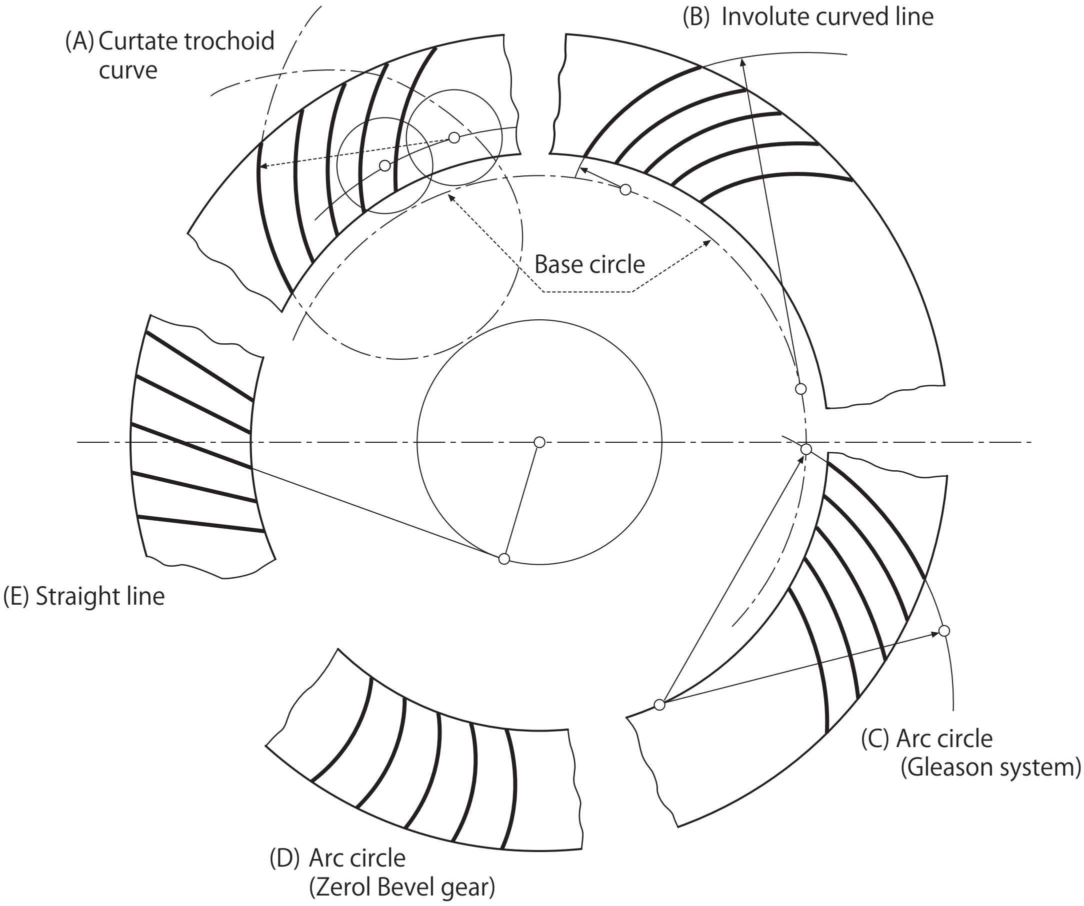
Fig. 14 Spiral bevel gear with Left and Right hand.

Table 9. Comparison table shows the minimum Number of teeth to prevent Undercut for Gleason Straight spiral bevel gear.

α=20°		α=16°		α=14.5°	
Number of teeth of Pinion	Number of teeth of Gear	Number of teeth of Pinion	Number of teeth of Gear	Number of teeth of Pinion	Number of teeth of Gear
z_1	z_2	z_1	z_2	z_1	z_2
12	26	16	59	19	70
13	22	17	45	20	60
14	20	18	36	21	42
15	19	19	31	22	40
16	18	20	29	23	36
17	17	21	27	24	33
		22	26	25	32
		23	25	26	30
		24	24	27	29
				28	28

(Reference)

These are types of Crown gear (similar relation between Rack and Spur gear) with curved line Tooth trace. Spiral bevel gears have following types shown in Fig. 15.



*Gleason system of large size Spiral bevel gear is close to rectilinear tooth but there is a slight spiral Tooth trace by modified rolling.

Fig. 15 Types of Spiral bevel gear (curved line of Tooth trace for Crown gear)

Worm gear pair

This is a Worm gear pair used as one pair of Threaded worm gear engaged with Worm wheel. It is commonly used in high speed reducing ratio.
Due to the low efficiency character, it is an important point to use proper lubricant oil to prevent heat generation.
To prevent thrust force to axis direction due to curve tooth, thrust bearing is necessary. (Refer to thrust force in Chapter 2)
Helix angle of Tooth for Worm gear is called Lead angle. Helix angle for Worm Wheel is called Helix angle same for Helical gear.
Generally Worm gear and Worm wheel match with Non-parallel and Non-intersecting axis. Shaft angle is 90°. For example, right lead angle of Worm gear matches with right helix angle of Worm wheel.
Worm gear with 2 or more number of threads are commonly called Multi-threaded worm gear.
Use suitable lead angle from Worm gear to fabricate the Helix angle for Multi thread Worm wheel.
Regarding the engagement for KG-Worm gear and KG-Worm wheel, refer to Table 10.

Table 10. The Engagement of KG-Worm and KG-Worm wheel (Assembled Gear pair should have same module.)

	Worm gear	Worm wheel
Symbol for Direction of thread and Number of thread.	R1 (Right hand /Single thread) R2 (Right hand/Double thread) L1 (Left hand/Single thread) L2 (Left hand/Double thread)	R1 (Right hand, fabricate helix angle by single thread of Worm gear) R2 (Right hand, fabricate helix angle by double thread of Worm gear) L1 (Left hand, fabricate helix angle by single thread of Worm gear) L2 (Left hand, fabricate helix angle by double thread of Worm gear)

There are the types of Normal and Axis worm gear pair, same as Helical gear. Normal type of Worm gear pair has come into wide use generally because it is economical.
When calculating Centre distance for Normal type of Worm gear pair, fraction appears due to $\tan\gamma$. In case of small lead angles, adjust Worm wheel by method of Negative Rack Shift to designated centre distance.

1.7 Backlash

Summary of the backlash is **"play"** or **"clearance"** between one pair of gear.

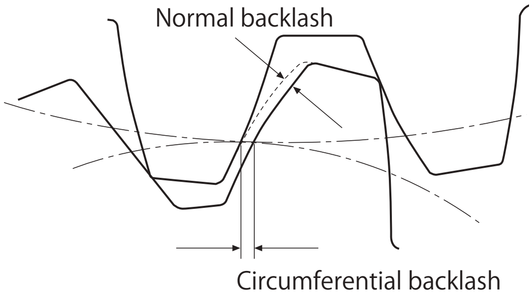


Fig. 17 Backlash

Great care is taken to produce the gear with zero deviation. However we are unable to completely eliminate deviation from manufacturing and surface heat treatment. A gear always has innate deviations of Tooth profile, Pitch, Runout, Tooth thickness and Helix by manufacturing process.
The Gearbox has innate deviation from manufacturing process. For example, shorter Centre distance compared with designed dimension, insufficient parallelism of axis or inaccurate right angle.
When starts the operation for gearbox, generation of heat from the load causes the gearbox to deform. Continuous operation increases the temperature of gearbox and thermal expansion of each part. As a result, swelling of the teeth causes oscillation, noise, sand burning and damages the tooth or bearing.

Proper backlash from the **"Play in the gears"** are necessary to absorb the deviations of noise and oscillation in order to have smooth rotation.
When assembling the gears, please provide the proper backlash between flanks.
Methods that provide the proper backlash to the gears are as follows;

1) Method to shift centre distance away. (Locating distance for Bevel gear)
This method does not provide the modification to the Tooth thickness, as it does not decrease Tooth thickness. This method simply shifts Centre distance away to obtain proper backlash to flanks.

2) Method of deeper cut during gear cutting process.
This method provides a deeper cut to reduce Tooth thickness when manufacturing the gear. Proper backlash is obtained if the gears are assembled with designated Centre distance.

Backlash for KG-STOCK GEARS

KG STOCK GEARS has been using method 2) from previous page. This method gives a proper backlash when assembled with designated centre distance of gearbox without adjustments.
Refer to the below references 11 to 13 for amount of backlash when assembling a pair of KG STOCK GEARS with designated centre distance.

Table 11. Amount of backlash for KG Spur gear (engagement of one pair with same material)

Module (m)	Materials	Amount of backlash (mm)
Range below m=0.9 is 0.02 - 0.06		
Range from m=0.9 to m=3.0	D, SU, BS	$0.06 \times m - 0.12 \times m$
	S	$0.04 \times m - 0.10 \times m$
	SCM	$0.04 \times m - 0.08 \times m$
Range from m=3 to m=5	S	$0.06 \times m - 0.12 \times m$

D: Polyacetal, SU: Stainless steel, S: Carbon steel, BS: Brass
SCM: Chromium molybdenum steel (Ground spur gear)

Table 13. Backlash of Bevel gear (one pair of gear engagement)

Module (m)	Backlash (mm)	
	SCM, S, SU, BS	D
Range below m=0.9	0.02-0.08	0.03-0.10
Range from m=0.9 to m=2.0	0.05-0.12	0.05-0.16
Range from m=2 to m=4	0.06-0.15	-
Range from m=4 to m=6	0.08-0.20	-
Range from m=6 to m=7	0.10-0.22	-

SU: Stainless steel, S: Carbon steel, SCM: Chromium molybdenum steel, D: Polyacetal

Table 12. Amount of backlash of Worm gear pair range from m 1.0 and above (one pair of engagement)

Centre distance	Amount of backlash (mm)
Range below m=0.8 is 0.06 - 0.15	
Below 50	0.08 – 0.20
Range from 50 to 150	0.15 – 0.30
Range from 150 to 300	0.30 – 0.50

Range below m=0.8, is 0.06 - 0.15(mm)

Measurement of the backlash

(1) Spur and Helical gears

There are a number of methods to measure the backlash for Spur and Helical gears.
Introduced are two (2) methods of measurement as follows;

a) Circumferential backlash j_t

Assemble one pair of gear with designated centre distance, fix one side of gear, put an indicator (Dial gauge) to Pitch circle of Mating gear and turn gear to the left and right to measure the amount of backlash. For Helical gear, measure backlash on the Pitch circumference at right angle section to axis.
In JIS, this is called the Circumferential backlash. Circumferential backlash for **Spur and Helical gears is stipulated in JIS B 1703.**

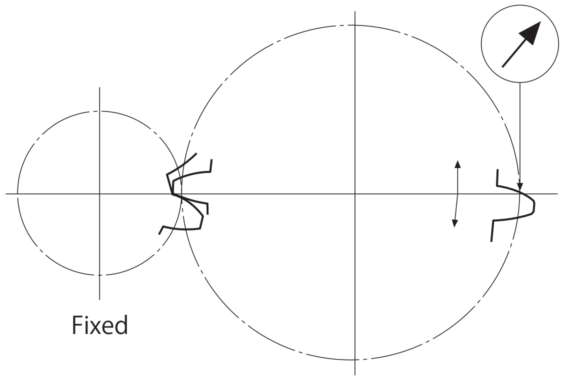


Fig. 18 Measurement of Circumferential backlash

b) Backlash j_n **in perpendicular direction to flank.**

Method of placinging indicatior perpendicularly to flank then follow same procedure in a).
In addition, another method is by putting a soft metal, eg. lead, between Flanks to measure the flattened metal thickness by a micrometer. This method of measurement may show different results compared with the method of simply using indicator to Flank because it is under the influence of play from bearing or other part's tolerance deviation. This method is called Normal backlash in JIS.

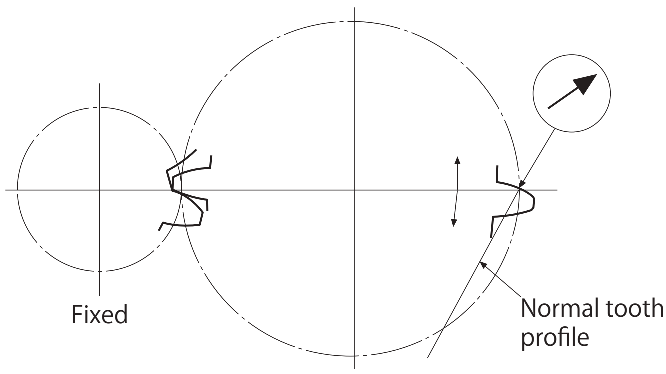


Fig. 19 Measurement of Normal backlash

For Spur gear with Pressure angle α, it has the following relationship between j_t and j_n .

$$j_n = j_t \cos\alpha \qquad j_t = j_n / \cos\alpha$$

When α is 20°, $\cos 20° = 0.93969$, j_t and j_n have similar value.

For Helical gear, an indicator is placed perpendicularly to the helixes of tooth for measurement. When Normal pressure angle is α_n and a helix angle is β,the relationship between j_t and j_n are as follows.

$$j_n = j_t \cos\alpha_n \cos\beta \qquad j_t = j_n / \cos\alpha_n \cos\beta$$

To measure backlash for Crossed helical gear pair (Screw gear) with indicator, fix either Pinion or Gear. When using either Pinion or Gear with Non-parallel and Non-intersecting axis, the reading on the indicator depends on which is chosen to be fixed. Usually Pinion is fixed and indicator is placed to flank of gear.

(2) Bevel gear

To Measure the backlash for Bevel gear pair, there are two (2) types of measurements. Circumferential backlash j_t and normal backlash j_n, which is the same for Spur and Helical gears.

Fix the pinion and put an indicator to outer gear to measure.

Normal pressure angle α_n and centre (mean) gear tooth of helix angle β_m of Spiral bevel gear have the following relationship between j_t and j_n.

$$j_n = j_t \cos\alpha_n \cos\beta_m \qquad j_t = j_n / \cos\alpha_n \cos\beta_m$$

(The above calculation formula is for Spiral bevel gear. For Straight bevel gear, it is cosine β_m =1)

Circumferential backlash for **Bevel gear pair is stipulated in JIS B 1705.**

In addition to this, there is another method to assemble the Bevel gear with a designated Locating distance. Fix a gear and move the Pinion in axis direction. Measure the amount of movement with an indicator.

Bevel gear has the following relationship between Circumferential backlash j_t and Locating direction jx.

$j_x = j_t / 2\tan\alpha_n \sin\delta_1$ Straight bevel gear

$j_x = j_{tt} / 2\tan d_t \sin\delta_1$ Spiral bevel gear

Hereby

j_{tt} : Circumferential backlash at Transverse plane

$j_{tt} = j_t /$cosine α_t

α_t: Transverse pressure angle $\alpha_t = \tan^{-1}(\tan\alpha_n / \cos\beta)$

For example, Straight bevel gear with Pressure angle 20° and gear ratio 1:1. Assuming that Circumferential backlash j_t is 1.0mm therefore backlash of Locating direction is 1.94mm. Which means it can measure minute backlash to about twice the accuracy.

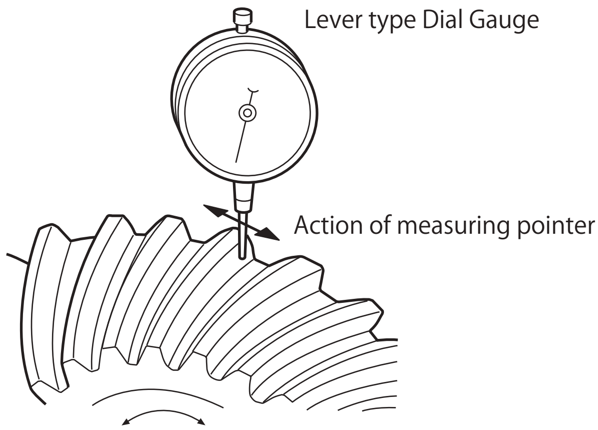


Fig. 20 Measurement method of backlash for the Bevel gear (Circumference direction)

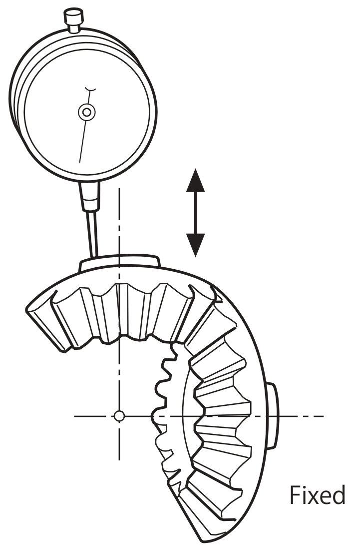


Fig. 21 Move the pinion in axis direction to measure the backlash.

(3) Backlash of Worm gear pair

Generally the Worm gear is fixed and indicator is placed to flank of Worm wheel for backlash measurement. This is the same method for both Spur and Helical gears pair.

Shown in Table 22, value for KG-Worm gear pair with assembled designated centre distance. Due to undefined backlash for Worm gear in JIS currently.

When using worm gear pair for accurate locating and positioning, it is necessary to keep backlash to a minimum. Providing large backlash for power transmission is recommend due to expansion caused by generation of heat. Even though the backlash may be larger, performance of worm gear pair will almost be the same.

Racing angle of Worm gear caused by backlash become a crucial problem occasionally.

Below is the explanation of the calculation formula for racing angle of Worm gear instead of backlash of Worm wheel.

Place an indicator to flank of Worm Wheel as show in Fig. 22 to measure circumferential backlash.

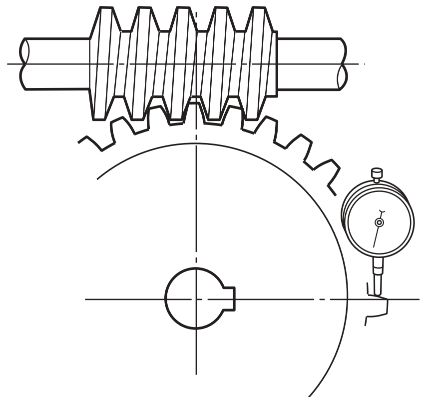


Fig. 22 Method of measurement for Worm gear pair (Circumference direction)

For example,
Module is 2.0,
Gear ratio 1：30,
Reference diameter of Worm gear is 31.0 mm,
Lead angle of Worm gear is 3°42′′ ,
Lead of Worm gear is 6.2963,
Measurement amount of Circumferential backlash is 0.2 mm.
Calculation formula is as follows.

(Lead) : (360°) = (Measured circumferential backlash) : (Racing angle of Worm gear) therefore,

$$\text{Racing angle of Worm gear} = \frac{360° \times \text{Circumferential backlash}}{\text{Lead}} = 360° \times 0.2 / 6.2963$$

$$= 11°27'$$

Worm gear provides the racing of 11°27′ .

(Lead of Worm gear : It is the distance of a point on the flank as it moves forward in axis direction when the Worm gear turns one revolution.)

1.8 Rack shift of the gear

Undercut

When Number of teeth is below minimum as shown in Fig. 23, part of dedendum is no longer an Involute curve but will look like a shape scooped out by cutter tool.

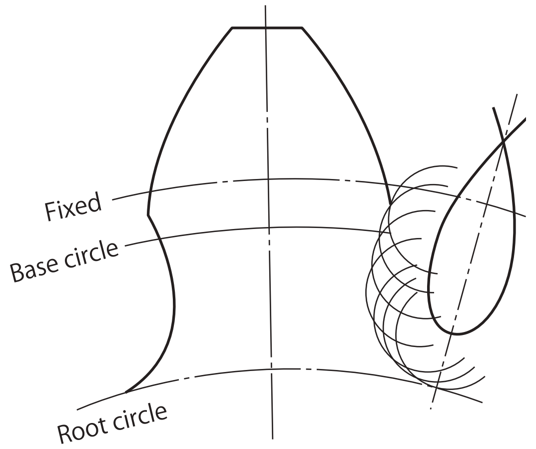
Refer to drawing, when Involute curve shows the scooped out shape condition from Base circle (Tooth tip side), it is called **Undercut**.

Gear with undercut has low strength of Dedendum and provides bad influence to gear contact due to shortened Involute curve.

Calculation formula for minimum number of teeth (z) to prevent undercut is as follows,

$$z = \frac{2}{\sin^2 \alpha_0} \qquad (\alpha_0\text{: Cutter pressure angle})$$

Condition of Undercut generally appears when Number of teeth is 17 or less and pressure angle of gear is 20°. According to DIN standard, minimum Number of teeth is 14 accepting slight Undercut which may cause no serious influence.



(The Trochoid curve line on the right hand side is the centre locus of roundness of cutter of rack tool. (radius of roundness γ_f=0.375m=7.5)

Fig. 23 Undercut

Profile Shifted Gear

(1) The Summary of Profile Shifted Gear

Using a rack tool (for example, hob cutter) to fabricate Profile Shifted Gear is to achieve the following purposes.

1) Prevent condition of Undercut for gear with less than minimum Number of teeth.

2) When there is deviation or failure for centre distance, fabricate a modified gear to correct the fault centre distance.

(3) Adjust distribution of Tooth thickness for gear pair to achieve equal gear strength.

4) Adjust to suitable contact ratio to lessen gear noise level and/or trapping of pump gear.

5) Take into consideration the wear of flank to adjust Specific sliding. (Another theory states that Specific sliding and wear are not proportional.)

It is possible to adjusting gear by item 2) to control helix angle of Helical gear. However it is necessary to provide thrust bearing in axis direction to countermeasure force (thrust force) occurring in Helical gear.

When design multi engagement between axes with different gear ratio, items 2) is also useful (for example, speed reducer).

Generally, Positive profile shift (+) is the method of gear fabrication where Reference pitch line of Rack type cutter shifts x-times of module toward outer radius direction from Reference pitch. The Negative profile shift (-) is that Reference pitch line of Rack type cutter shifts x-times of module towards inner radius direction from Reference pitch. $x.m$ is commonly called the **Amount of rack shift** where x is called **Rack shift coefficient**. (Please refer to Fig. 24).

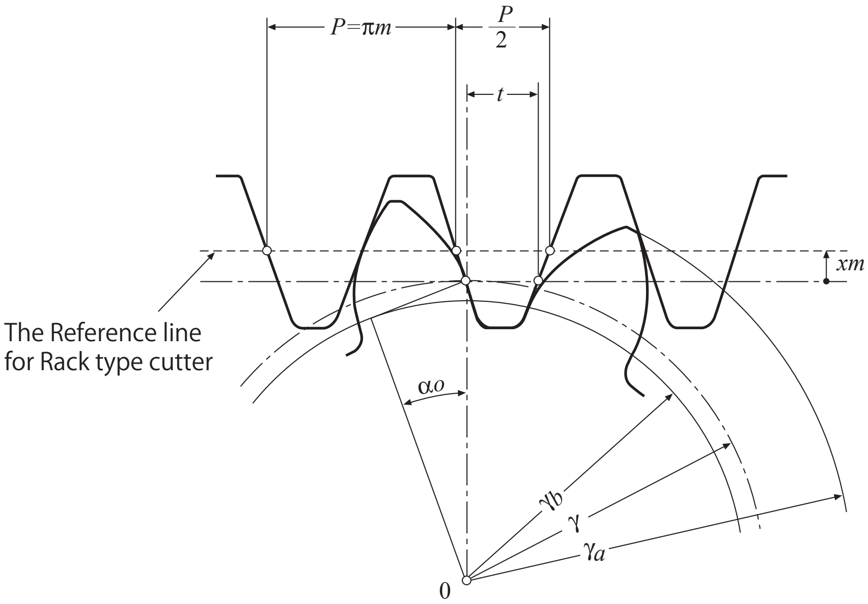
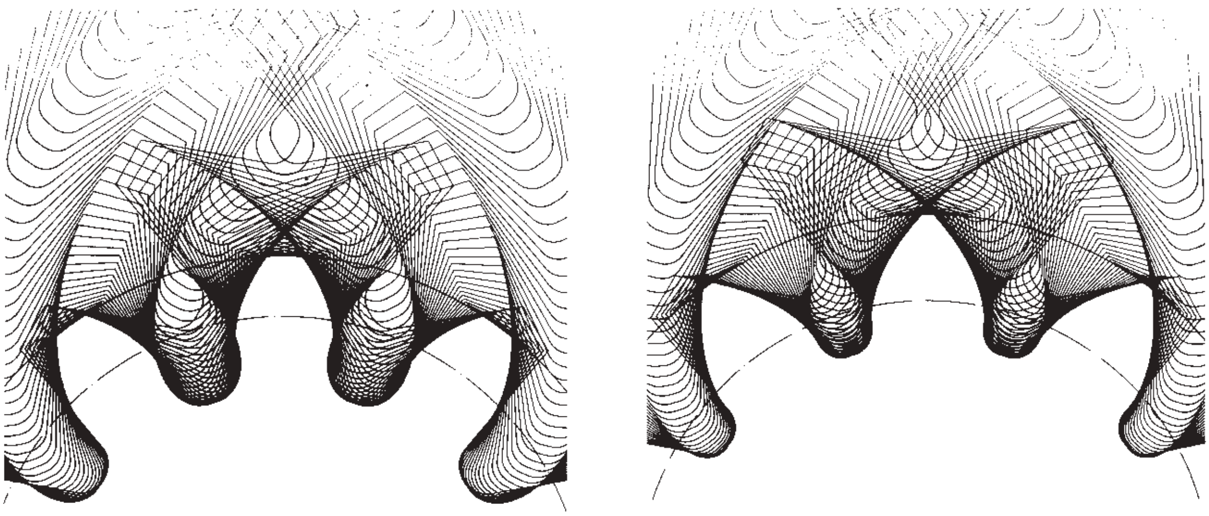


Fig. 24 Rack shift for Spur gear



Spur gear with full depth tooth
(Rack shift coefficient x=0)

Positive (+) profile shifted gear
(Rack shift coefficient x=0.5)

Fig. 25 Profile shifted gear (Examples of Positive and Negative profile shifted gear, Number of teeth is 12z)

Note: (1) Adopted the old Standard term.

Limitation of Pointed tooth tip

When increase the positive amount of Rack shift, area of top land is gets narrower and soon, Tooth profile becomes sharp.

A sharp pointed Tooth profile has insufficient tooth depth, thus Tooth tip of Mating gear may interfere with Root of tooth causing proper assembly and smooth gear rotation to be impossible. Therefore Rack shift of Top land exceeding zero is not advisable.

To calculate Top land 's' of Spur gear by the following formula,

$$s = m(z+2+2x) \bullet \left\{ \left(\frac{\pi}{2} + 2x\tan\alpha_0\right) \bullet \frac{1}{z} - (\mathrm{inv}\,\alpha_a - \mathrm{inv}\,\alpha_0) \right\}$$

For easy reference, please refer to Table 14 for area of formed gear with Pressure angle 20°.

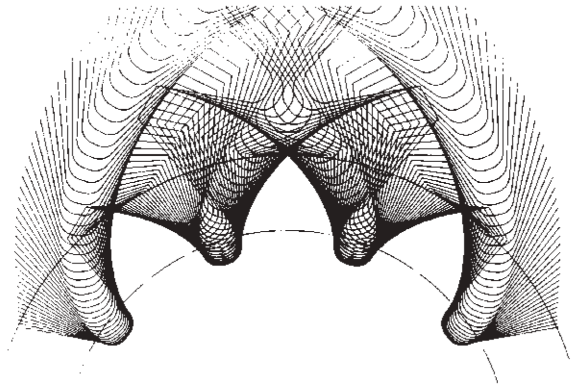


Fig. 26 Pointed tooth tip

Calculation for Rack shift coefficient.

(1) Rack shift coefficient to prevent Undercut.

Undercut is sure to occur when Number of teeth is 17 or below with Pressure angle 20°. Prevent Undercut using theoretical Rack shift coefficient by following calculation formula.

$$x = \frac{17-z}{17}$$ (z: Practical number of teeth)

Practical number of teeth 14 z is available to use for DIN standard, calculation formula of DIN is defined as follows.

$$x = \frac{14-z}{17}$$ (z: Practical number of teeth)

Theoretical Rack shift coefficient for Spur gear with Number of teeth 10z with Pressure angle 20° is by following formula

$$x = \frac{17-10}{17} = 0.412$$

(Please check for occurrence of sharp pointed tooth top tip using Table 14.)

Practical rack shift coefficient is obtained by following calculation.

$$x = \frac{14-10}{17} = 0.235$$

(2) Rack shift coefficient to adjust Centre distance

Below is the explanation using examples.

For example, calculate Rack shift coefficient for adjustable gear with Centre distance of 80.5mm (Proper distance is 80.0mm) with: Gear: Spur gear,

Pressure angle:20°,

Module: 2.0mm,

Number of teeth for Pinion: 20z,

Number of teeth for Gear: 60z,

Centre distance modification coefficient

$$y = (a' - a)/m$$
$$= (80.5 - 80)/2$$
$$= 0.25$$

$$y = \frac{z_1+z_2}{2}\left(\frac{\cos\alpha_0}{\cos\alpha_w} - 1\right)$$ therefore

$$\cos\alpha_w = \frac{\cos\alpha_0}{\frac{2 \bullet y}{z_1+z_2} + 1} = \frac{\cos 20°}{\frac{2 \bullet 0.25}{20+60} + 1}$$
$$= 0.933856$$
$$\alpha_w = 20.955894°$$
$$\mathrm{inv}\,\alpha_w = \tan\alpha_w - \alpha_w$$
$$= \tan 20.955894° - 20.955894° \bullet \pi/180$$
$$= 0.0172317$$

$$\mathrm{inv}\,\alpha_w = 2 \bullet \tan\alpha_0 \bullet \left(\frac{x_1+x_2}{z_1+z_2}\right) + \mathrm{inv}\,\alpha_0$$ therefore

Sum of Rack shift coefficient

$$x_1 + x_2 = \left(\frac{\mathrm{inv}\,\alpha_w - \mathrm{inv}\,\alpha_0}{2 \bullet \tan\alpha_0}\right) \bullet (z_1+z_2)$$
$$= \frac{0.0172317 - 0.0149044}{2 \bullet \tan 20°} = 0.2557$$

a' : Actual centre distance (mm)
a : Proper centre distance (mm)
z_1 : Number of teeth for Pinion
z_2 : Number of teeth for Gear
α_0 : Pressure angle of Cutter (°)
α_w :pressure angle (°)
y : Centre distance increment coefficient
x_1 : Rack shift coefficient for Pinion
x_2 : Rack shift coefficient for Gear
$\mathrm{inv}\,\alpha_0$: Functional involute for Cutter pressure angle

$$\mathrm{inv}\,\alpha_0 = \tan\alpha_0 - \alpha_0$$
$$\mathrm{inv}\,20° = 0.0149044$$

(The last α_0 is in Radian Unit)

You may provide the sum (0.2557) of this Rack shift coefficient to Pinion only or can divide between Gear and Pinion.

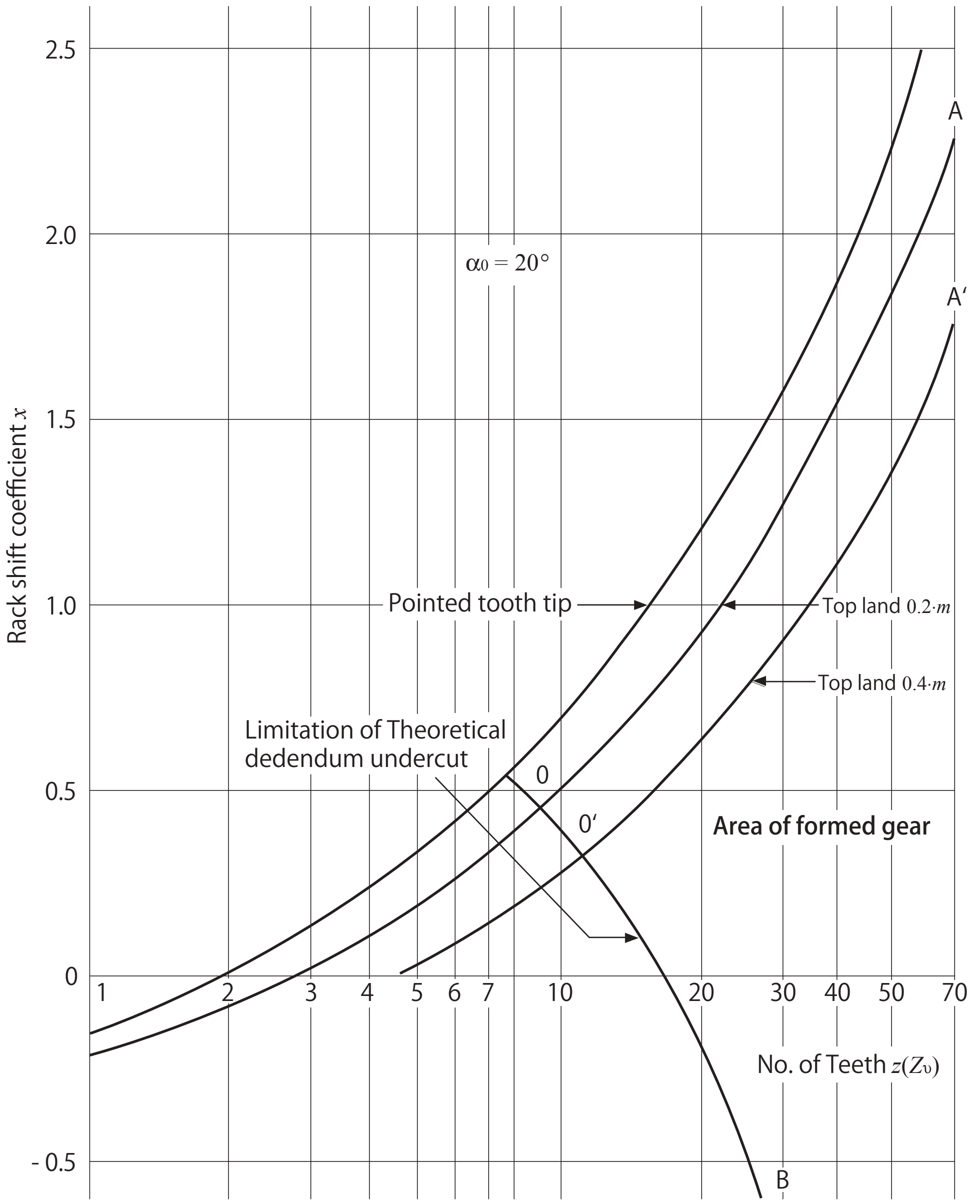
(3) Guidelines for determining Rack shift coefficient.

Rack shift to positive side is mainly designed for Pinion. It is necessary to check that the calculated Rack shift coefficient does not cause pointed tooth tip.

If design causes pointed tooth tip, reduce amount of Rack shift coefficient to Pinion and offset amount to Gear.

As for Rack shift to negative side, it is necessary to check for Undercut. If Undercut should occur, offset the Negative rack shift coefficient to mating gear.

Refer to Table 14 to shown the area of formed gear with pressure angle 20°.



Curved line A : Top land changed to 0.2・m by Rack shift coefficient and Number of teeth.

Curved line A' : Top land changed to 0.4・m by Rack shift coefficient and Number of teeth.

Curved line B : Rack shift coefficient and Number of teeth for Limitation of Theoretical dedendum undercut.

* For Helical gear, use horizontal axis in chart for Virtual number of teeth of spur gear Zv.

$$Zv = Z / \cos^3\beta$$

Table 14. Area of formed gear (pressure angle 20°)

Note (1) Adopted old gear terms.

The features of Tooth profile 05

Tooth profile of KG STOCK GEARS (Number of teeth from 8z to 11z) has been adopted by type 05 in DIN standard.

Tooth profile type 05 has its Rack shift coefficient fixed to plus (+) 0.5. Adjust Addendum by shortening coefficient x module ($\kappa.m$) to fabricate smaller Outside diameter, as the Bottom clearance have a tendency to be narrow.

The calculation of Rack shift for Number of teeth ranging from 8z to 11z for KG STOCK GEARS is as follows,

Calculation formula for Working pressure angle α_w is as follows:

$$\mathrm{inv}\,\alpha_w = 2\tan\alpha\left(\frac{x_1+x_2}{z_1+z_2}\right)+\mathrm{inv}\,\alpha$$

Hereby

z_1=No. of teeth for Pinion

z_2= No. of teeth for Gear

z_1=Rack shift coefficient for Pinion

x_2=Rack shift coefficient for Gear

α_0= Pressure angle (Cutter pressure angle)

inv= Involute function

$\mathrm{inv}\alpha=\tan\alpha-\alpha$

(Refer to page 164-167 for the Involute function table)

Centre distance modification coefficient y is as follows:

$$y=\frac{z_1+z_2}{2}\left(\frac{\cos\alpha}{\cos\alpha_w}-1\right)$$

Centre distance a_x is following formula:

$$a_x=\left(\frac{z_1+z_2}{2}+y\right)m$$

Hereby

m=module

Working pitch diameter d'_1 and d'_2 is by following formula:

$$d'_1=2\,a_x\left(\frac{z_1}{z_1+z_2}\right)$$

$$d'_2=2\,a_x\left(\frac{z_2}{z_1+z_2}\right)$$

Reference diameter d_1 and d_2 is by following formula:

$$d_1=z_1m$$

$$d_1=z_2m$$

Tip (Outside) diameter d_{ax} is following formula:

$$d_{ax}=2m\left(\frac{z+3}{2}-\kappa\right)$$

Hereby

κ=Truncation coefficient

$$\kappa m=\left[x_1+x_2-\frac{z_1+z_2}{2}\left(\frac{\cos\alpha}{\cos\alpha_w}-1\right)\right]m$$

When Addendum of cutter is module 1.25, Bottom clearance (minimum amount) is module 0.21.

The Centre distance for number of teeth 8z and 8z is as follows,

(Rack shift coefficient x=0.5)

$$a_x/m=8.7788\text{mm}$$

The centre distance for number of teeth 10z and 10z is as follows.

(Rack shift coefficient x=0.5)

$$a_x/m=10.8043\text{mm}$$

The above calculations are for module 1.0. Example for module is 2.0 with number of teeth 8z and 8z are engaged, centre distance ax based on above a_x/m = 8.778mm is as follows:

$$a_x=8.7788\times 2$$
$$=17.5576\text{mm}$$

Mating gear with other Number of teeth of KG STOCK GEARS is available.

With regards to the tooth profile of type 05 for the Rack Shift Coefficient quoted by Gear Industry Volume No.54, "German Gear Standard" (DIN 3994 and 3995)

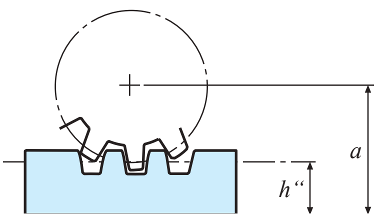
The Centre distance between KG Rack shifted spur gear and KG STOCK GEARS

Usage of below comparison table: Where module is 1.0, calculate the centre distance a_x multiply by module.

No. of teeth

No. of teeth \ No. of teeth	8	9	10	11
8	8.779	9.286	9.792	10.298
9	9.286	9.792	10.299	10.804
10	9.792	10.299	10.804	11.310
11	10.299	10.804	11.310	11.815
12	10.437	10.939	11.441	11.943
13	10.939	11.441	11.943	12.445
14	11.441	11.953	12.445	12.946
15	11.943	12.445	12.946	13.448
16	12.445	12.946	13.448	13.949
17	12.946	13.448	13.949	14.451
18	13.448	13.949	14.451	14.952
19	13.949	14.451	14.952	15.453
20	14.451	14.952	15.453	15.954
21	14.952	15.453	15.954	16.455
22	15.453	15.954	16.455	16.956
23	15.954	16.455	16.956	17.457
24	16.455	16.956	17.457	17.958
25	16.956	17.457	17.958	18.459
26	17.457	17.958	18.459	18.960
27	17.958	18.459	18.960	19.461
28	18.459	18.960	19.461	19.962
29	18.960	19.461	19.962	20.463
30	19.461	19.962	20.463	20.963
32	20.463	20.963	21.464	21.965
34	21.464	21.965	22.465	22.966
35	21.965	22.465	22.966	23.467
36	22.465	22.966	23.467	23.967
38	23.467	23.967	24.468	24.968
40	24.468	24.968	25.469	25.969
42	25.469	25.969	26.470	26.970
44	26.470	26.970	27.471	27.971

No. of teeth \ No. of teeth	8	9	10	11
45	26.970	27.471	27.971	28.472
46	27.471	27.971	28.472	28.972
48	28.472	28.972	29.473	29.973
50	29.473	29.973	30.473	30.974
52	30.473	30.974	31.474	31.974
54	31.474	31.974	32.475	32.975
55	31.974	32.475	32.975	33.475
56	32.475	32.975	33.475	33.976
58	33.475	33.976	34.476	34.976
60	34.476	34.976	35.477	35.977
62	35.477	35.977	36.477	36.977
64	36.477	36.977	37.478	37.978
65	36.977	37.478	37.978	38.478
66	37.478	37.978	38.478	38.979
68	38.478	38.979	39.479	39.979
70	39.479	39.979	40.879	40.979
72	40.479	40.979	41.480	41.980
75	41.980	42.480	42.980	43.480
80	44.481	44.981	45.481	45.981
84	49.482	46.982	47.482	47.982
85	46.982	47.482	47.982	48.482
90	49.483	49.983	50.483	50.983
95	51.983	52.483	52.984	53.484
96	52.483	52.984	53.484	53.984
100	54.484	54.984	55.484	55.985
105	56.985	57.485	57.985	58.485
108	58.485	58.985	59.485	59.985
110	59.485	59.985	60.485	60.986
112	60.485	60.986	61.486	61.986
115	61.986	62.486	62.986	63.486
120	64.486	64.987	65.487	65.987



Centre distance between KG-Rack shifted spur gear and KG-Rack

$$a = h'' + \frac{m \times z}{2} + xm$$

Hereby

- a : Centre Distance (Distance from Datum of Rack to Centre of KG-Spur gear)
- h'' : Datum line of Rack (Refer to page 259)
- m : Module
- x : Rack shift coefficient
- z : Number of teeth

(Module 1.0 and above
For Number of teeth 8 to 11, x =0.5
For Number of teeth 12 and above, x=0)

1.9 Contact ratio and Specific sliding

Contact ratio

(1) Theory of Contact ratio

Actual engaging teeth at working area are lesser than number of teeth manufactured on circumference.

Contact ratio describes working condition and is an element that influences gear oscillation, noise, strength, rotation and others.

It is generally believed that large Contact ratio is better. Below is the explanation using engagement between Spur gears as example.

Refer to the Fig. 27 for Involute cylindrical gear describes the engagement on the tangential line $\overline{I_1I_2}$ of Base circle for both gears. This line is commonly called **Contact line** or **Line of action**.

Actual engagement on this Contact line is from range A_1 to A_2 of both Tip circles.

On the assumption that pinion is the driving gear. Firstly start contact between Dedendum of Pinion and tooth tip of gear at A_1 to engage.

As the gear rotates, point of contact passes through P-point (Pitch point), engaging with Dedendum of gear and Tooth tip of Pinion. After a short time, gears disengage at point A_2.

To perform gear rotation continuously, it is necessary for the next engaging pair of teeth to be engaged perfectly before disengaging the current pair.

In Fig. 27, $\overline{A_1A_2} = g_a$ is called **Length of path of contact**. Distance from point A_1 to P is called Length of approach path g_α, distance point P to A_2 is called **Length of recess path** g_β.

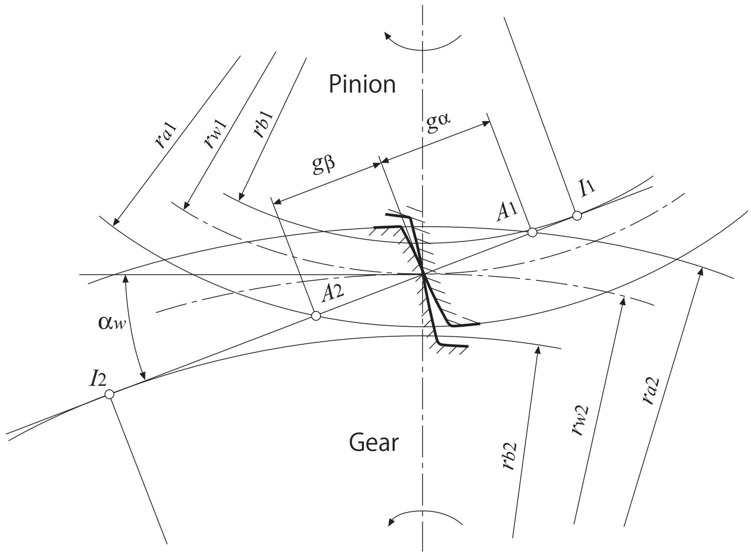


Fig. 27 Length of path of contact

Formula for Length of path of contact g is as follows.

$$g_{\alpha} = \overline{A_1P} = \overline{A_1I_2} - \overline{PI_2} = \sqrt{\gamma_{a2}^{\,2} - \gamma_{b2}^{\,2}} - \gamma_{w2} \bullet \sin\alpha_w$$

$$g_{\beta} = \overline{A_2P} = \overline{A_2I_1} - \overline{PI_1} = \sqrt{\gamma_{a1}^{\,2} - \gamma_{b1}^{\,2}} - \gamma_{w1} \bullet \sin\alpha_w$$

$$a_x = \gamma_{w1} + \gamma_{w2} \quad \text{Therefore}$$

$$g_a = g_{\alpha} + g_{\beta} = \sqrt{\gamma_{a2}^{\,2} - \gamma_{b2}^{\,2}} + \sqrt{\gamma_{a1}^{\,2} - \gamma_{b1}^{\,2}} - \alpha_x \bullet \sin\alpha_w$$

Hereby

γ_a :Tip radius

(The subscripts 1 and 2 indicate Pinion and Gear, respectively.)

γ_b : Base radius

α_w : Working pressure angle

α_x : Centre distance (Profile shifted gear)

Spacewidth on contact line is Base pitch ρ_b. **Contact ratio** is Length of path of contact divided by Base pitch. To maintain continuous rotation, Length of path of contact should be larger than Base pitch.

Therefore, formula of Contact ratio ε is as follows,

$$\varepsilon = \frac{\text{Length of path of contact}}{\text{Base Pitch}} = \frac{g_a}{\rho_b} \quad (\rho_b = \pi m \cos\alpha_0)$$

Contact ratio ε must be above 1.0

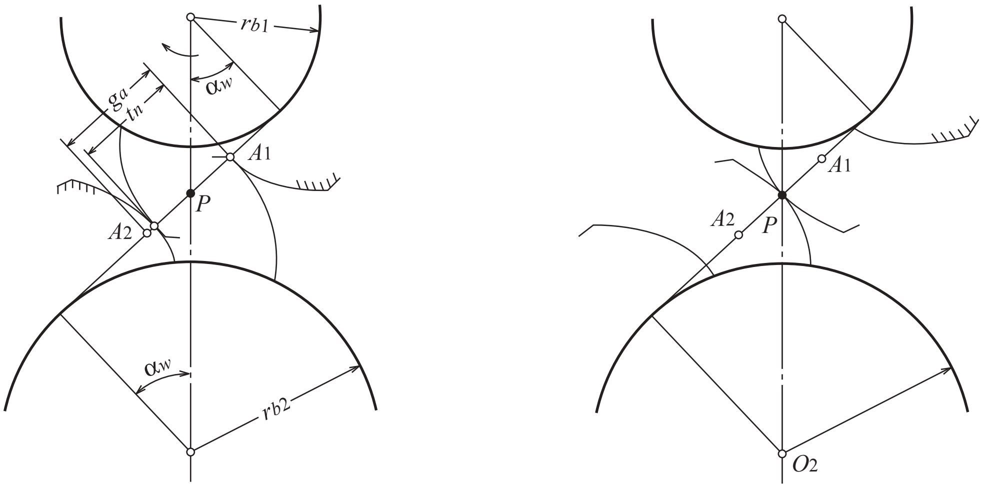


Fig. 28. Two teeth - contact and One tooth - contact.

For example, assume Contact ratio 1.487 for Spur gear pair engagement.

Look carefully at Fig. 28. In the beginning of engagement, engagement with two pairs of teeth. As two pairs rotate toward Pitch point, one pair of tooth is engaged.

When one pair of teeth continues rotating forward, two pairs of teeth engages. The cycle repeats.

Therefore, meaning of Contact ratio 1.487 is when two pairs of teeth will be engaged at 48.7% of Length on the path of contact with in the beginning and at the end. One pair of teeth will be engaged at the remaining 51.3%.

For gear with pressure angle 20°, repeating the same rotation when full load to one tooth and shared load to two teeth of gear.

Cause of oscillation and noise is due to the amount of deflection, which is different when engaging with one tooth or two teeth.

The value of Contact ratio depends on Pitch diameter, Pressure angle, Number of teeth, Rack shift coefficient and Tip diameter. Therefore refer to below.

1) Increase in Pressure angle will decrease Contact ratio.

2) Increase in sum (x_1+x_2) of Rack shift coefficient will decrease Contact ratio.

3) Full depth tooth gear with same Pressure angle and module will result in increase Contact ratio when Number of teeth is increased. On the other hand, when Number of teeth decreases and undercut occurs, Contact ratio will decrease extremely. Smaller Pressure angle will result in Contact ratio with a tendency to decrease.

4) When designing Full depth gear tooth (height of tooth is taller than full depth tooth), special tool is needed for the increased Tip diameter.

(2) Contact ratio of Spur gear

Refer to Table 15 for calculation formula for Contact ratio of Spur gear is as follows.

Assume the gear as a Rack, formula is $g_a=(h_{a2}-x_1m)/\sin\alpha_w$

Hereby

h_{a2} : Addendum of rack

x_1 : Rack shift coefficient of Spur gear

(3) Contact ratio for Helical gear

Contact ratio for Helical gear on the Transverse plane has the same calculation formula as Spur gear. Due to Helix tooth, value of Facewidth b divided by Normal pitch is added to Transverse contact ratio (This value is commonly called Overlap ratio).

Therefore,

The Transverse contact ratio ε_α + The Overlap ratio ε_β = The Total contact ratio ε_γ. Refer to Table 16, calculation formula of Contact ratio for Helical gear is as follows.

Table 15. Examples of Contact ratio for Spur gear

Common gear data: Module m=2.0, Cutter pressure angle α_0=20°

Gear 1	Gear 2		Contact ratio ε	Example
Spur gear $z_1=12$ $x_1=0.5$	Spur gear	$z_2=40$ $x_2=0$	$\varepsilon=\dfrac{\sqrt{r_{a1}^2-r_{b1}^2}+\sqrt{r_{a2}^2-r_{b2}^2}-a_x\sin\alpha_w}{\pi m\cos\alpha_0}$	ε=1.399
	Rack	$x_2=0$	$\varepsilon=\dfrac{\sqrt{r_{a1}^2-r_{b1}^2}+\dfrac{h_{a2}-x_1m}{\sin\alpha_0}-r_1\sin\alpha_0}{\pi m\cos\alpha_0}$	ε=1.475
	Internal gear	$z_2=100$ $x_2=0$	$\varepsilon=\dfrac{\sqrt{r_{a1}^2-r_{b1}^2}+\sqrt{r_{a2}^2-r_{b2}^2}+a_x\sin\alpha_w}{\pi m\cos\alpha_0}$	ε=1.515

Table 16. Contact ratio of Helical gear

Common gear data: Normal module mn=2.0, Helix angle β=15°, Cutter pressure angle α_0=20°, Facewidth b=20.0.

Gear 1	Gear 2	Contact ratio ε	Example
$z=20$ $x_{n1}=0$	$z=40$ $x_{n2}=0$	Transverse contact ratio $\varepsilon_\alpha=\dfrac{\sqrt{r_{a1}^2-r_{b1}^2}+\sqrt{r_{a2}^2-r_{b2}^2}-a_x\sin\alpha_{wt}}{\pi m_t\cos\alpha_t}$ Overlap ratio $\varepsilon_\beta=\dfrac{b\bullet\sin\beta}{\pi m_n}$ Total contact ratio $\varepsilon_\gamma=\varepsilon_\alpha+\varepsilon_\beta$	ε_α=1.561 ε_β=0.824 ε_γ=2.385

(4) Contact ratio for Bevel gear

Straight bevel gear uses the same calculation as Spur gear. To obtain Contact ratio, it assumes the formula of [(1)] Virtual spur gear upon the Back cone.
Due to Helix from tooth of Spiral bevel gear, overlap ratio is added to obtain the Transverse contact ratio from [(1)] Virtual spur gear for calculation.
Refer to Table 17 for calculation formula for Contact ratio of Bevel gear is as follows.

Table 17. Contact ratio for Bevel gear

Common gear data: Module m=2, Shaft angle Σ=90,
Face width b=13 (Spiral tooth) Pitch diameter d_1=36, d_2=72 Pitch angle δ_1=26° 33′ 54″ δ_2=63° 26′ 06″

Gear 1	Gear 2	Contact ratio ε	Example
z=18	z=36	Back cone distance $R_v = \dfrac{d}{2 \bullet \cos\delta}$	R_{v1}=20.125 R_{v2}=80.499
		Base radius of [(1)] Virtual spur gear (Straight tooth) $R_{vb} = R_v \bullet \cos\alpha_0$ (Spiral tooth) $R_{vb} = R_v \bullet \cos\alpha_t$	R_{vb1}=18.911 R_{vb2}=75.644 R_{vb1}=18.391 R_{vb2}=73.564
		Tip radius of [(1)] Virtual spur gear $R_{va} = R_v + h_a$	(Straight tooth) R_{va1}=22.815 R_{va2}=81.809 (まがり歯) R_{va1}=22.410 R_{va2}=81.614
		Contact ratio (Straight tooth) $\varepsilon = \dfrac{\sqrt{R_{va1}^2 - R_{vb1}^2} + \sqrt{R_{va2}^2 - R_{vb2}^2} - (R_{v1} + R_{v2})\sin\alpha_0}{\pi m \cos\alpha_0}$	ε=1.610
		Transverse contact ratio (Spiral tooth) $\varepsilon_\alpha = \dfrac{\sqrt{R_{va1}^2 - R_{vb1}^2} + \sqrt{R_{va2}^2 - R_{vb2}^2} - (R_{v1} + R_{v2})\sin\alpha_t}{\pi m \cos\alpha_t}$ Overlap ratio $\varepsilon_\beta = \dfrac{b\tan\beta_m}{\pi m} \bullet \dfrac{R_e}{R_e - 0.5b}$ Total contact ratio $\varepsilon_\gamma = \varepsilon_\alpha + \varepsilon_\beta$	ε_α=1.270 ε_β=1.728 ε_γ=2.998

Note that (1) Adopted the old standard term.

Theory for Specific sliding (for reference)

Specific sliding is shown as condition of sliding where engaged flanks slides to transfer the rotation except area of pitch point.

Refer to Fig. 29, when one pair of Tooth profile is in contact at C point, after minute moment, it will contact points of C_1 and C_2 respectively. Where $C\text{-}C_1=ds_1$ and $C\text{-}C_2=ds_2$, calculation formula for Specific sliding δ is as follows.

$$\delta_1 = \frac{ds_1 - ds_2}{ds_1} \qquad \delta_2 = \frac{ds_2 - ds_1}{ds_2}$$

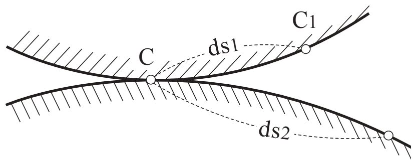


Fig. 29 Sliding

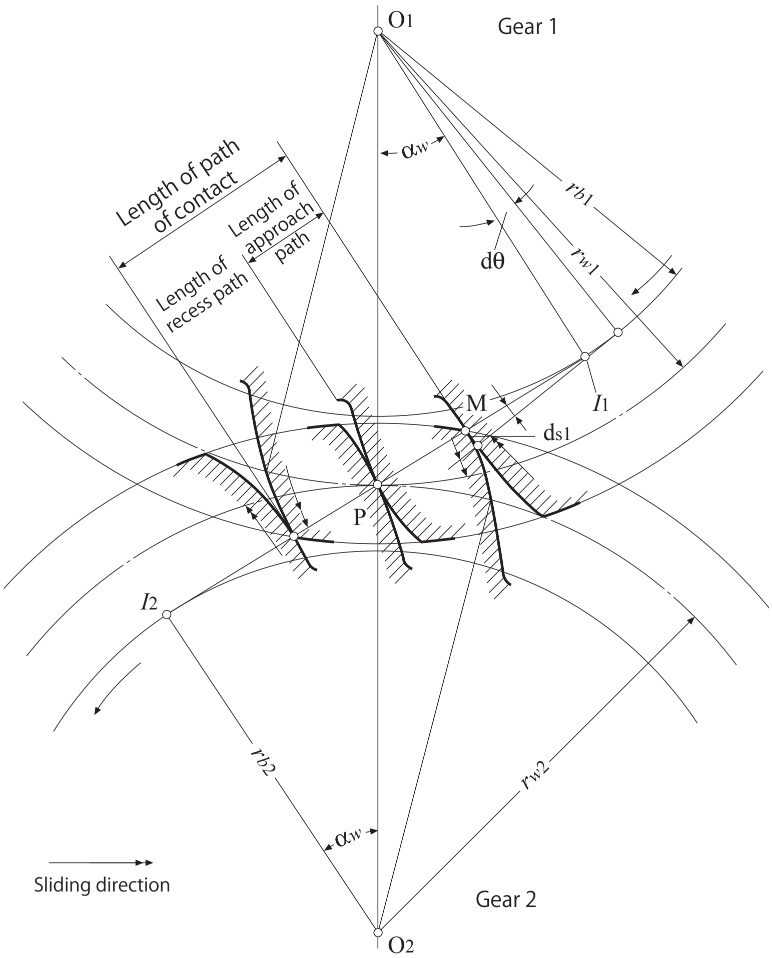


Fig. 30 Sliding direction of Flank for Involute tooth profile

Refer to Fig. 30, when Involute gear 1 makes $d\theta$ revolution as gear 2 makes $\gamma_1.d\theta/\gamma_2$ revolution.

When contact point upon Tooth profile has been shifted, length of ds_2 and ds_1 is by following formula,

$$ds_1 = (\overline{I_1M})d\theta \qquad ds_1 = (\overline{I_2M})\frac{\gamma_{w1}}{\gamma_{w2}}d\theta$$

When PM=L, calculation formula is as follows,

$$\overline{I_1M} = \overline{PI_1} - \overline{PM} = \gamma_{w1} \bullet \sin\alpha_w - L$$

$$\overline{I_2M} = \overline{PI_2} - \overline{PM} = \gamma_{w2} \bullet \sin\alpha_w + L$$

$$L = \sqrt{\gamma_{a2}{}^2 - \gamma_{b2}{}^2} - \gamma_{w2} \bullet \sin\alpha_w$$

Refer to Fig. 18. Specific sliding for each part of Tooth profile.

Table 18. Specific sliding for Involute gear

	Specific sliding of Addendum flank	Specific sliding of Dedendum flank
Gear 1	$\delta_{a1} = \dfrac{1 + \dfrac{\gamma_{w1}}{\gamma_{w2}}}{\dfrac{\gamma_{w1}}{L}\sin\alpha_w + 1}$	$\delta_{f1} = \dfrac{1 + \dfrac{\gamma_{w1}}{\gamma_{w2}}}{\dfrac{\gamma_{w1}}{L}\sin\alpha_w - 1}$
Gear 2	$\delta_{a2} = \dfrac{1 + \dfrac{\gamma_{w1}}{\gamma_{w2}}}{\dfrac{\gamma_{w1}}{L}\sin\alpha_w + \dfrac{\gamma_{w1}}{\gamma_{w2}}}$	$\delta_{f2} = \dfrac{1 + \dfrac{\gamma_{w1}}{\gamma_{w2}}}{\dfrac{\gamma_{w1}}{L}\sin\alpha_w - \dfrac{\gamma_{w1}}{\gamma_{w2}}}$

As for Involute gear, refer to Fig. 31 for sliding contact to all areas except area of intermeshing pitch point. The Specific sliding increases as teeth moves away from Pitch point

When Contact ratio increases for Involute tooth profile, condition of Specific sliding will have a tendency to decrease.

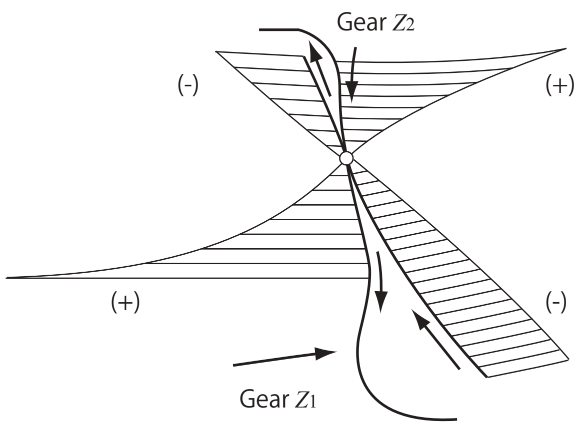


Fig. 31 Distribution of Specific sliding

1.10 Tooth profile modification

Tooth profile modification

Regarding Tooth profile modification, modify the tooth profile that is shifted from the involute to be (concave) near part of tooth tip or fillet of dedendum. Tooth profile modification is to prevent deflection of tooth caused by load, intereference of Tooth tip caused by Pitch deviation and adds to provide smooth gear rotation.
However, needlessly exceeding amount of Tooth profile modification is not advisable as it will result in deterioration of Contact ratio. Proper amount of Tooth profile modification is highly recommended.
Commonly, modify the fillet area of dedendum for driver gear and area of Tooth tip for driven gear.

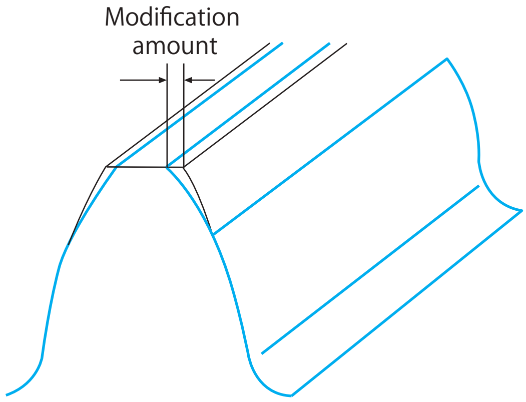


Fig. 32 Profile modification
(Part of Tooth tip of driven gear)

Modification of Tooth trace (Crowning and Relieving)

Refer to Fig. 33, regarding modification of Crowning. Reduce Tooth thickness from centre towards the end of Tooth trace gradually.
Refer to Fig. 34, regarding Relieving. Reduce Tooth thickness gradually at end of Tooth trace.
The purposes of both modifications are to prevent the stress concentration by single contact. Different points between Crowning and Relieving are that Crowning prevents stress concentration caused by single contact and Relieving simply relieves the end of Tooth trace to prevent single contact.
These methods are commonly called Tooth trace modification. Excessive amount of Tooth trace modification will result in deterioration of tooth contact. This excessive modification is not advisable.

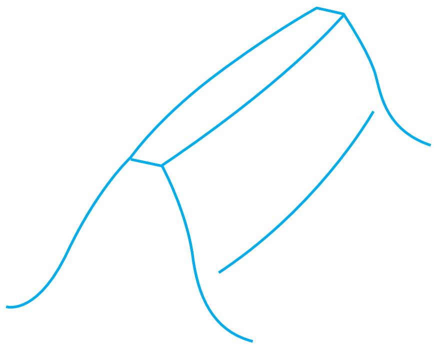


Fig. 33 Crowning

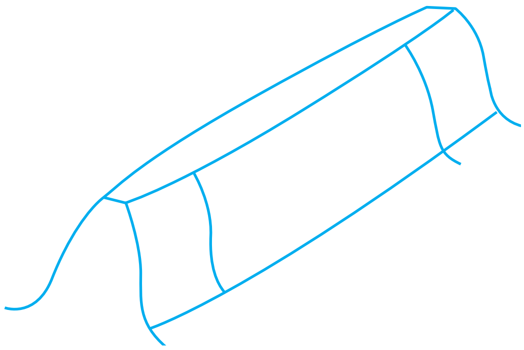


Fig. 34 Relieving

Topping and Semi Topping

When cutting the flanks by Topping hob cutter, the outside diameter of gear is also processed at the same time. Semi topping method is similar but outside diameter is chamfered by hob cutter.

Method of Topping uses an external micrometer to measure the outside diameter to control Tooth thickness if module is too small and unable to use method of Sector span.

Topping cutter is designed to obtain designated Tooth thickness when machined outer diameter of gear is in place. It has an effect to reduce off-centre deviation of outside diameter as the hob cutter processes the outside diameter of gear at same time.

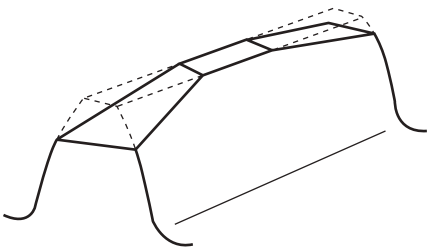
Semi topping prevents dent marks and burrs from occurring at Tooth tip. Semi topping has an effect to lower oscillation and noise, as smaller dent mark does not interfere with engagement. Excessive Semi topping will deteriorate Contact ratio and is not advisable.

Tooth profile adjustment

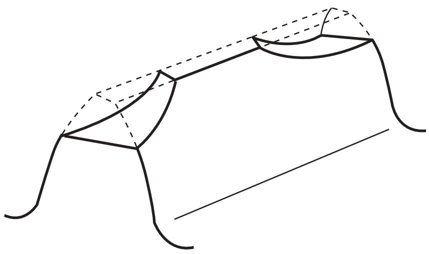
Process of hob cutter slightly cuts away both ends of fillet from outside diameter of gear in direction of Tooth trace. There are types of Straight line and Curved line profile adjustments, which can reduce the fluctuation of spring constant for gear. This has an effect to lower oscillation and noise to within expectation.

Professor Niemann has introduced other adjustments, where the outside diameter of gear is slightly (in other words, not as extreme as for Bevel gear) tapered or the outside diameter of gear is cut in an arc shape to make it a drum shaped body, which is not outlined here.

Another method is for outside diameter of Bevel gear to be slightly cut away to prevent interference at the toe.



Straight adjustment



Arc adjustment

Chapter 2 Precaution for usage

2.1 Precaution of usage for Helical gear

① To obtain ideal engagement for Crossed helical gear (Screw gear), provide both shaft angles to be 90° as accurately as possible.

② Provide the bearing that will completely support the thrust load when Helical gear is operated in the axial thrust direction.

③ Thrust load in Helical gear:
Helical gear is able to obtain a smoother engagement as compared to Spur gear. However, Helical gear produces thrust load in the axial direction due to Tooth trace is helix shape. Therefore the design of the shafts between driver gear (pinion) and driven gear (gear) should have bearing that will completely support against axial thrust load. (Refer to Fig. 1)

④ Load applied on Helical gear

(a) Tangential load

$$F = \frac{1.432H \times 10^6}{dn}$$

Hereby

H : Transfer power(PS)

n : Revolution per minute (rpm)

d : Pitch diameter (mm)

(b) Axial direction thrust

$$F_a = F\tan\beta \ \text{(kgf)}$$

Hereby

β : Helix angle

(c) Calculation for load to displace the axis

$$F_s = F\tan\alpha_t \ \text{(kgf)}$$

$$= \frac{F\tan\alpha_n}{\cos\beta}$$

Hereby

α_t : Transverse pressure angle

α_n : Normal pressure angle

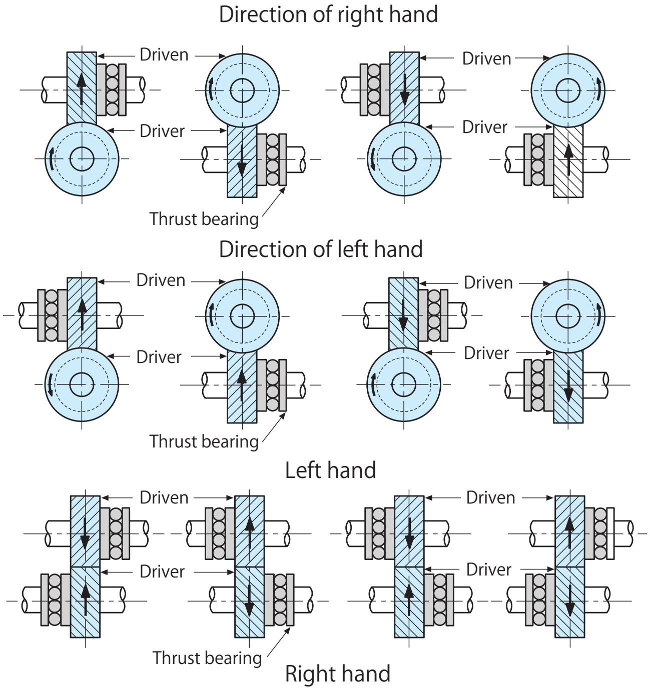


Fig. 1 Axial thrust load of Helical gear and location of bearings

(d) Normal load (Perpendicular to flank)

$$F_n = \frac{F}{\cos\beta\cos\alpha_n} \ \text{(kgf)}$$

Load applied to bearing: ① Tangential load-F is divided between two bearings in connected direction of gears, ② Load to displace the axis- F_s is divided between two bearings, perpendicular to ①, ③ Couple of force by axial direction thrust- F_a (in the direction perpendicular to tooth surface where F_a is applied)
Therefore the sum of 3 types of load vector acts to each bearing.
[Gear Design and Manufacture] written by Dr. Waguri Akira

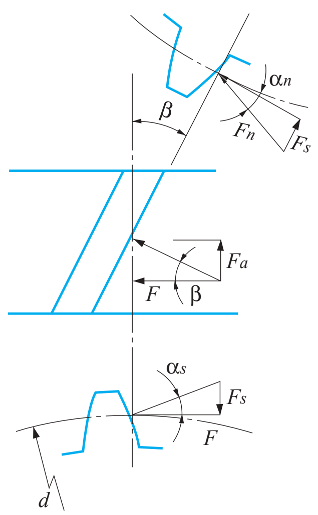


Fig. 2

2.2 Precaution of usage for Bevel gear

① To obtain ideal engagement of the Bevel gears, the correct shaft angle and proper backlash is necessary when assembling.

② For Bevel gear, it is important to note method of installation. Bearing for the shaft for Bevel gear is mainly on one side. Therefore shaft becomes defective due to deflection when load is applied. Single contact occurs and results in overhung condition.
The design of gear axes and bearings should be firm and provide bearing as close as possible to Bevel gear.
During assembly, shift the non-fixed Bevel gear up and down in axis direction to obtain proper tooth bearing. It is recommended to put shim at area of base surface for adjustment of tooth bearing.

③ We recommend that Machined straight bevel gears are suitable for circumferential speed (pitch diameter) less than 328m/min and Machined spiral bevel gears are suitable for circumferential speed (pitch diameter) more than 328m/min. The above-mentioned statement does not apply to Injection molded type of Bevel gears.
The Gleason Company in USA recommend that Machined spiral bevel gears are suitable for circumferential speed (pitch diameter) more than 5.5 m/s and above 1,000 revolution per minute and Ground spiral bevel gear are suitable for circumferential speed (pitch diameter) more than 40 m/s.

④ Spiral bevel gears are able to run smoothly in high speed environment providing a quiet operation due to fewer Number of teeth contacting with mated gear and wide Number of teeth on Pitch cone as compared to Straight bevel gear.
Spiral bevel gear has overlapping engagement on Pitch cone surface element between tooth to tooth and the load does not concentrate on one (1) Tooth tip. The advantages are extremely steady and compact design for usage at high speed.
The only disadvantage is axial thrust load, which is generated due to Spiral tooth trace. Therefore proper design of the bearing location with firm support is needed to be as close to the Spiral bevel gear as possible in order to minimize this Axial thrust load. (Refer to Fig. 3)

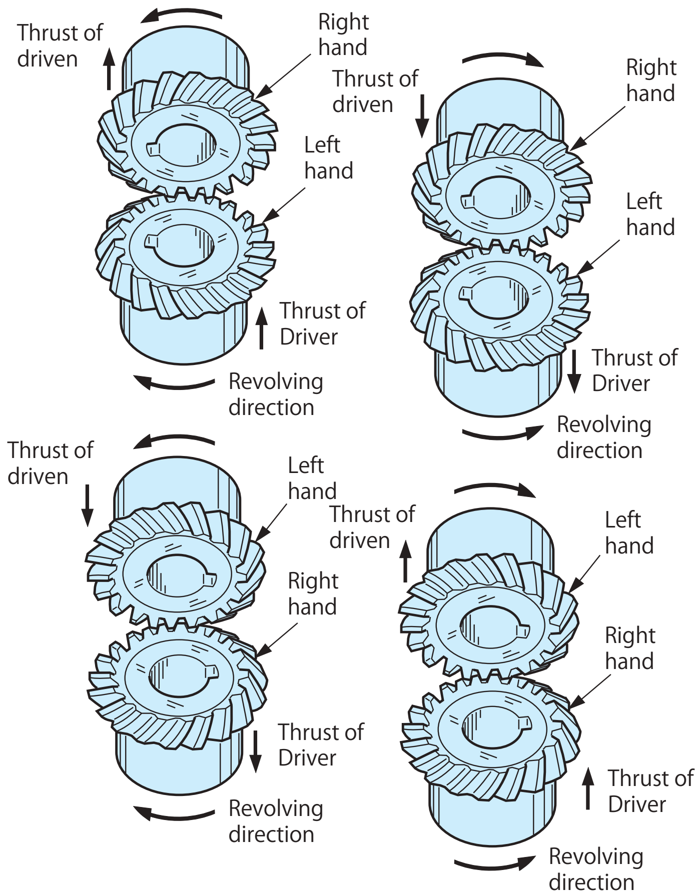


Fig. 3 Thrust load on Spiral bevel gear

⑤ The load applied to Straight bevel gear (Refer to Fig. 4)

(a) Tangential load

$$F = \frac{1.432H \times 10^6}{d_m n} \text{ (kgf)}$$

Hereby

H : Transfer power (PS)

n : Revolution per minute (rpm)

d_m : Mean pitch diameter (mm)

(b) Thrust in Axial direction

$$F_a = F\tan\alpha\sin\delta \text{ (kgf)}$$

Hereby

α : Pressure angle

δ : Pitch angle

(c) Calculation for load to displace the axis

$$F_s = F\tan\alpha\cos\delta \text{ (kgf)}$$

(d)Normal load

$$F_n = \frac{F}{\cos\alpha} \text{ (kgf)}$$

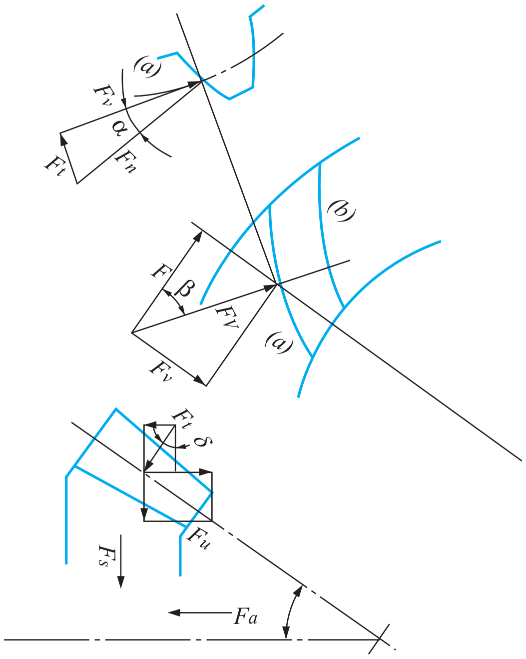


Fig. 5

⑥ The load applied to Spiral bevel gear. (Refer to Fig. 5)

(a)Tangential load

$$F = \frac{1.432H \times 10^6}{d_m n}$$

Hereby

H : Transfer power (PS)

n : Revolution per minute (rpm)

d_m : Mean pitch diameter (mm)

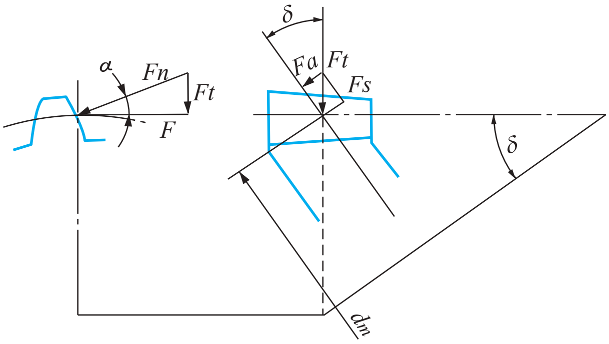


Fig. 4

(b) When convex side is driver

(b.1) Thrust in axial direction

Driving gear

$$F_a = F\{\tan\alpha\left(\frac{\sin\delta}{\cos\beta}\right) - \tan\beta\cos\delta\} \cdot 9.80665\,[\mathrm{N}]$$

Driven gear

$$F_a = F\{\tan\alpha\left(\frac{\sin\delta}{\cos\beta}\right) + \tan\beta\cos\delta\} \cdot 9.80665\,[\mathrm{N}]$$

Hereby

α :Pressure angle

δ :Pitch angle

β : Spiral angle

Refer to Fig. 5, when the condition is F_a>0, axial thrust direction is away from the top. The condition is F_a<0, axis thrust direction is towards the top.
Generally, pinion has smaller pitch angle δ due to F_a<0. Stable design to convex side is necessary.

(b.2) Calculation for load to displace the axis

$$F_s = F\{\tan\alpha\left(\frac{\cos\delta}{\cos\beta}\right) + \tan\beta\sin\delta\} \text{ (kgf)} \cdot 9.80665\,[\mathrm{N}]$$

(b.3) Normal load

$$F_n = \frac{F}{\cos\beta\cos\alpha}$$

(c) When concave side is driver.

When the load is applied to flank, F_u direction is opposite from drawing,

(c.1) Thrust to axial direction (direction away from the top)

Driving gear

$$F_a = F\{\tan\alpha\left(\frac{\sin\delta}{\cos\beta}\right) + \tan\beta\cos\delta\} \cdot 9.80665\,[\mathrm{N}]$$

Driven gear

$$F_a = F\{\tan\alpha\left(\frac{\sin\delta}{\cos\beta}\right) - \tan\beta\cos\delta\} \cdot 9.80665\,[\mathrm{N}]$$

(c.2) Calculation for load to displace the axis

$$F_s = F\{\tan\alpha\left(\frac{\cos\delta}{\cos\beta}\right) - \tan\beta\sin\delta\} \cdot 9.80665\,[\mathrm{N}]$$

2.3 Precaution of usage for Worm gear pair

① To obtain ideal engagement of Worm gear and Worm wheel's shafts, provide right angle (90°) correctly.

② Lubricant oil is indispensable to Worm gear and Worm wheel during operation due to high friction between flanks of Worm gear and Worm wheel.

③ Engagement of the same number of thread and hand of thread are indispensable to Worm gear and Worm wheel. (Engage Worm gear and Worm wheel with both having right hand and one thread)

④ The design of the axes between Worm gear and Worm wheel should be firm and provide bearing as close as possible to Worm gear pair.

⑤ Provide the bearing that will completely support the Worm gear pair as the axial thrust increases during operation. Refer to Fig. 6 for axial thrust direction.

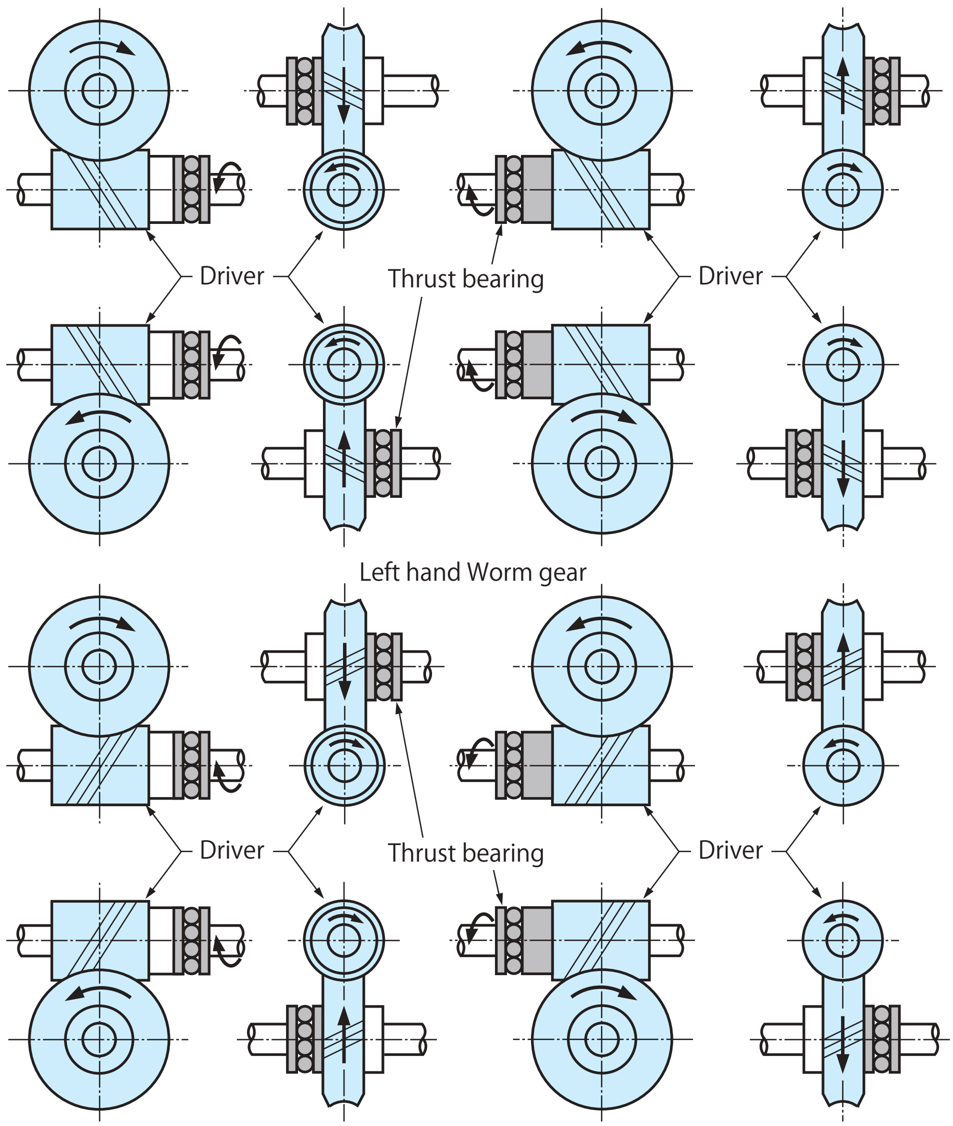


Fig. 6 Axial thrust load to Worm gear and location of bearings.

⑥ When assembling and warm up for Worm gear pair, design such that Tooth contact can be measured and assembly position can be adjusted.

⑦ Worm gear pair performs self-locking when lead angle is below 4°. Please separately design the safety device to stop the gear from inversing.

⑧ Load applied to Worm gear pair (Refer to Fig. 7)
$F_1d_1/2$ is moment for driver of Worm gear. F_2 is revolving force for Worm wheel by $F_1d_1/2$. Formula is as follows,

$$F_1 = F_2\tan(\gamma+\rho) = \frac{4.5H\times10^6}{\pi d_1 n_1}\bullet 9.80665\ (\mathrm{N})$$

(a) F_2 is axial direction thrust for Worm gear.

$$F_2 = \frac{F_1}{\tan(\gamma+\rho)} = \frac{1.432H\times10^6}{\tan(\gamma+\rho)\times d_1 n_1}\bullet 9.80665\ (\mathrm{N})$$

F_1 is axial direction thrust for Worm wheel.

$$F_1 = \frac{1.432H\times10^6}{(d_1 n_1)}\bullet 9.80665\ (\mathrm{N})$$

Hereby

H : Net power applied to Worm gear (PS=horse power)

γ : Lead angle

$$\tan\rho = \frac{\mu}{\cos\alpha_n}$$

μ : Coefficient of friction on flank

α_n : Normal pressure angle

d_1 : Pitch diameter of Worm gear (mm)

n_1 : Revolution speed per minute for Worm gear

ρ : Apparent friction angle of flank

Note: If H_2PS is the power from Worm wheel and η_R is efficiency. Calculation is as follows.

$$H = \frac{H_2}{\eta_R}$$

(b) Calculation for load to displace the axis

$$F_s = \frac{F_1\tan\alpha_n\cos\rho}{\sin(\gamma+\rho)}\ (\mathrm{N})$$

Alternatively $= F_n\sin\alpha_n$

(c) Normal load

$$F_n = \frac{F_1\cos\rho}{\sin(\gamma+\rho)\cos\alpha_n}\ (\mathrm{N})$$

Reference literature: Dr. Waguri Akira "Gear Design and Manufacturer" 30th Machine Literary of Japan.

Basic formula of Worm gear pair

1. Sliding velocity υ_s(m/s)

$$\upsilon_s = \frac{\pi d_1 n_1}{60\times1000\times\cos\gamma}$$

Hereby

d_1 : Pitch diameter of Worm gear (mm)

n_1 : Revolution per minute for Worm gear (min^{-1})

γ : Reference pitch cylinder lead angle (°)

2. Torque and Efficiency (When the driver is from Worm gear)

$$T_2 = \frac{F_t d_2}{2000}\bullet 9.80665\ (\mathrm{N\bullet m})$$

Hereby

T_2 : Nominal torque of Worm wheel (N・m)

F_t : Nominal circular force of Worm wheel (N)

d_2 : Pitch circumferential diameter of Worm wheel (mm)

$$T_1 = \frac{T_2}{u\eta_R} = \frac{F_t d_2}{2000u\eta_R}$$

Hereby

T_1 : Nominal torque of Worm gear (N・m)

u : Gear ratio ($u=z_2/zw$)

η_R : Transfer efficiency of Worm gear pair when driver is from Worm gear.

$$\eta_R = \frac{\tan\gamma\left(1-\tan\gamma\frac{\mu}{\cos\alpha_n}\right)}{\tan\gamma+\frac{\mu}{\cos\alpha_n}}$$

Hereby

μ : Coefficient of friction

α_n : Normal reference pressure angle (°)

Note the efficiency of KG' s Worm gear pair is as follows.

Worm gear with single thread	45% - 55%
Worm gear with double thread	55% - 65%

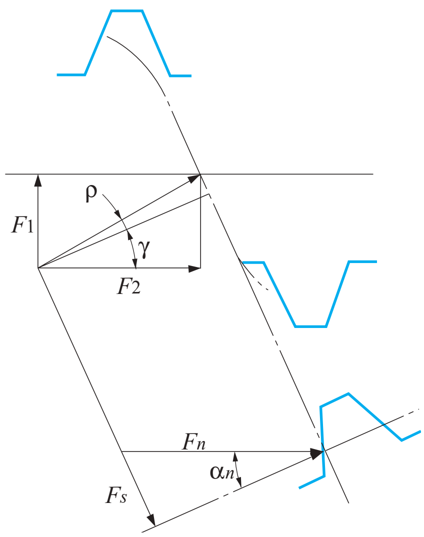


Fig. 7 Load applied to Worm gear pair.

2.4 Precaution of usage for Anti backlash spur gears

Function of Anti backlash spur gear

Backlash is a necessary function for gearing, however anti-backlash spur gear can remove backlash mechanically.

The principle of KG-Anti backlash spur gear is that of a time-honored method, KG-Anti backlash spur gear has springs that produce load. These springs generate larger torque than the axial torque applied to a pair of gearbox. Select Allowable torque based on calculation of load produced from the springs.

Mechanism of Anti backlash spur gear has built-in springs that pull each other between gear A and B to pinch the Mating gear like a scissor.

When rotated direction of the gear is reversible, the springs of Anti backlash spur gear can continue to maintain suitable torque by pinching Mating gear.

If interference occurs due to gear quality, Anti backlash spur gear with gear A and B absorbs the interference by stretching the spring mechanism while engaged between Anti backlash spur gear and Mating gear

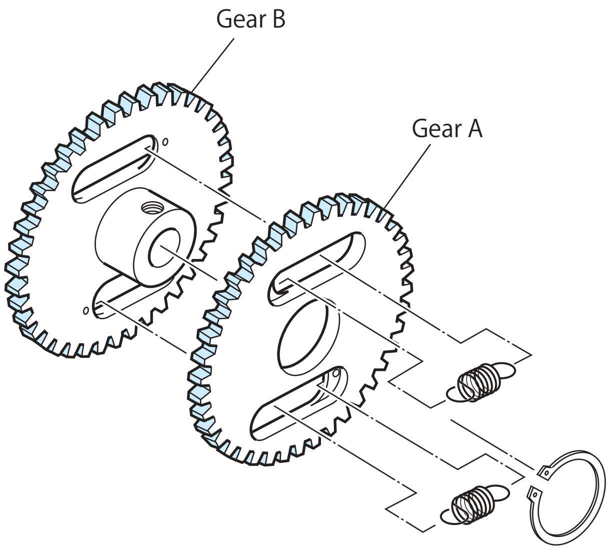


Fig. 8 Mechanism of Anti backlash spur gear.

Regarding the Mating gear for KG-Anti backlash spur gear

Mating gear for KG-Anti backlash spur gear is compatible with other makers. However, it is advisable to use KG-GROUND SPUR GEARS or KG-STOCK SPUR GEARS for best results.

Adjustment of zero point as n0

When built-in springs on the Anti backlash spur gear is in free condition (free condition- no tension to spring), positions of tooth tips between gear A and B do not match.

Method of adjusting n0. Firstly fix gear B, secondly rotate gear A in the engraved arrow direction until both gear have no tension from spring for types BS and BW and gradually rotate the installed spring towards tension direction. Zero point n0 is the first position of matched teeth between gears A and B with tension of spring.

Method of settlement of required Allowable transfer torque

1. Method of Shifting pitch (n)

Firstly, select a suitable NS or NSG series from KG-Anti backlash spur gears. Secondly, select the numerical value of shifting pitch n higher than your required torque from the Allowable transfer capability torque table.

For NSU series, there is a limitation of selection for shifting pitch in accordance with the Allowable transfer capability torque table. The allowed shifting is only two (n=2).

Please refer to Allowable transfer capability torque table for NSU series.

2. Method of settlement for Allowable transfer torque

For example to obtain the required Allowable torque for your existing required NSG part number NSG50S 60B+0808.

If the current torque speed is 15N per cm it is required to shift to n3 pitch before getting the next larger value.

3. In case where Allowable torque required is unattainable (n).

(1) In such a case where gear engagement operation cannot be obtained after shifting pitch (n) is set in accordance to [Method of settlement for Allowable transfer torque], please re-study the amount of desired torque. The actual torque applied to gear may sometimes vary from theoretical torque.

(2) If a suitable shifted pitch n0 cannot be selected from table of limitation of Anti backlash to function (N・cm) for your required Allowable torque after setting the shifted pitch n0 by [Method of settlement for Allowable transfer torque], please do not hesitate to call us for discussion.

Precaution for additional process to Anti backlash spur gear

Additional machining to Anti backlash spur gear is not advisable, as deformation will result in loss of anti backlash function.

Precaution for additional machining to KG- Anti backlash spur and Ground spur gears, dismantle gear A and B before additional machining. Note: remove snap ring at the hub of Anti backlash gear to dismantle. Note: remove snap ring at the hub of Anti backlash gear to dismantle.

Beware of dent marks when doing additional works or dismantling gear A and B.

Remove the burrs on the gear perfectly after additional machining.

As a precaution for re-assembly of gears after additional machining, ensure dust free condition between gear A and B.

Customized Anti backlash spur and ground spur gears.

Please provide us with the following details for making customized Anti backlash spur and ground spur gears.

1. Gear data and type of gear
2. Usage of maximum torque [N・m]
3. Usage of Revolution per minute [min^{-1}]
4. Material
5. Usage environment (Air, under water, vacuum and etc.)
6. Lubrication
7. Check surroundings for object that may cause interference to gear.

We look forward to receiving your gear drawing and above details for customized Anti backlash spur and ground spur gears.

2.5 Precaution of usage for B-BOX

Dismantlement of B-BOX is strictly prohibited.
Please do not use torque that exceeds Allowable transfer capability.
Avoid overhang load action to input and output shafts. If there is overhang and thrust loads to gear shafts of the HY-BOX, B-BOX (BS, BSH) and B-SET, it is necessary to design an extra preventable function. (Refer to Fig. 9)
Beware of shocks to shafts and body of BOX.

Installation precaution (For efficient use of B-BOX)

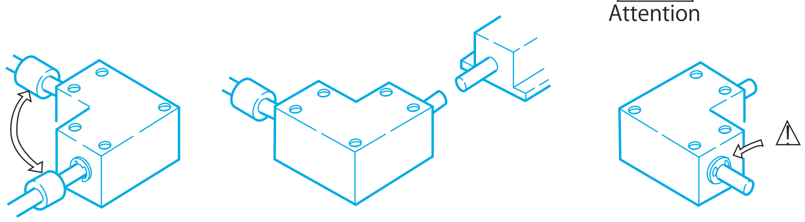
◇ To prevent damage to the gear shafts, gear shaft of B-BOX and mating shaft must be aligned at right angle before assembly.

◇ Before operation, it is necessary to confirm smooth rotation of shafts by hand.

◇ To prevent damage to the gear shafts, provide accurate parallelism and shaft center between gear shaft of B-BOX and mating shaft before assembly. (Accuracy of alignment ϕ0.05 or less is recommended)

◇ Beware of any waste objects being caught in the snap ring.

Attention



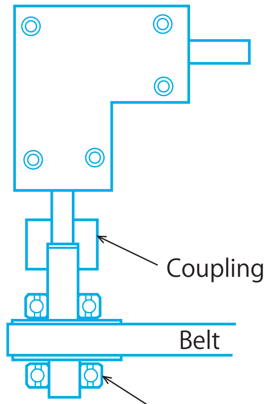


Fig. 9 Reference solution for overhang load

Beware of unusual sound during warm up test. If unusual sound is heard, stop operation and do not hesitate to contact us for solution.
Using flexible coupling will reduce misalignment between shaft of B-BOX and mating shaft.
KG-B-BOX series is not completely sealed. Please do not use in environment with water, oil and chemicals.
When gear is rotating at high-speed at ratio 2:1, power to output shaft, noise level and temperature will increase but power transfer will decrease. The opposite is true at low speed.

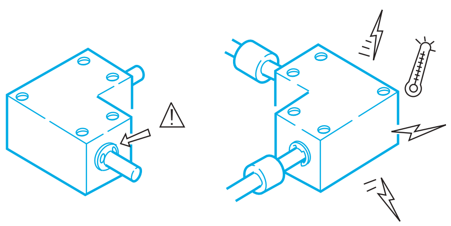
During operation (For safety purposes, please pay attention to the followings below)

◇ Do not touch the gearbox, shaft and key during operation.

◇ Beware of waste objects being caught in the snap ring at the back of body.

◇ Stop operation and check for faults if there are any problems such as unusual sound and high temperature occurring from gearbox. Do not start the machine until the faults has been cleared.

Attention



Precaution of additional works

(Take note to be careful to prevent loss of function when making additional machining works to B-BOX)

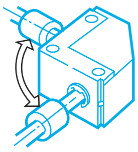
To avoid damage to B-BOX, please do not hesitate to contact us for more details.
Before additional machining, ensure that bearing portion is covered, so that waste objects will not contaminate it.
Beware of shaft deformation when doing additional machining works on the bearings.
Additional machining to body of B-BOX is strictly prohibited as it may damage the internal functions.

2.6 Precaution of usage for B-SET

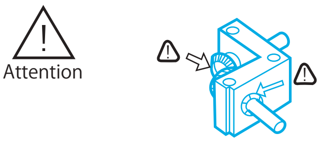
Please do not use torque that exceeds Allowable transfer capability.
Avoid overhang load action to input and output shafts. If there is overhang and thrust loads to gear shafts of the B-SET, it is necessary to design an extra preventable function. (Refer to Fig. 10)

Installation precaution (For efficient use of B-BOX)

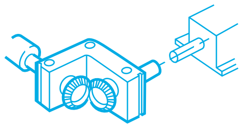
◇ To prevent damage to the gear shafts, gear shaft of B-SET and mating shaft must be aligned at right angle before assembly.
◇ Before operation, it is necessary to confirm smooth rotation of shafts by operating with hand.



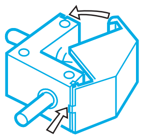
◇ Beware of fingers or waste objects getting caught in a snap ring.



◇ To prevent damage to the gear shafts, provide accurate parallelism and shaft centre between gear shaft of B-SET and mating shaft before assembly.

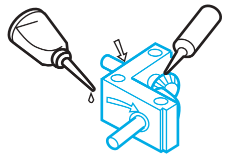


◇ Ensure that the body of B-SET is properly covered by plastic cover before starting the machine.
◇ Follow steps ① and ② for instruction to properly cover the plastic cover and handle the cover with care.

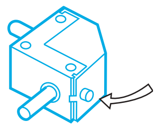


② Push the convex area of plastic cover into the concave groove of body perfectly.
① Set the convex area of plastic cover to overlap the concave groove properly.

◇ Apply grease to the Tooth and lubricant oil to bearings regularly. Beware of running out of oil.



◇ If there is a possibility that the plastic cover might come off during operation, use bolts to secure the plastic cover.

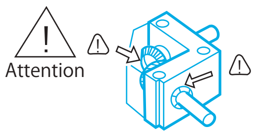


Beware of damaging the bearings and surrounding area when additional drilling is carried out on the B-SET.

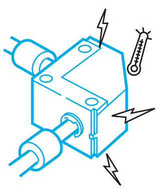
Before operation, test run with no load is recommended to check for faults or noise.
Use flexible coupling to reduce misalignment between shaft of B-SET and mating shaft.
Beware of dust and particles clogging the bearing and Toothing.
When the gear is rotating at high speed at ratio 2:1, noise level and temperature will increase but power transfer will decrease. The opposite is true at low speed.

During operation (For safety purposes, please pay attention to the followings below)

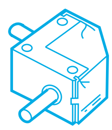
◇ Do not touch the gearbox during operation.
◇ Beware of waste objects being caught in the snap ring at the back of body.



◇ Stop operation and check for faults if there are any problems such as unusual sound and high temperature occurring from gearbox. Do not start the machine until the faults has been cleared.



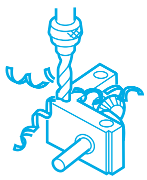
◇ Plastic cover is available for purchase as spare parts for maintenance use when time to be replaced due to aged deterioration.



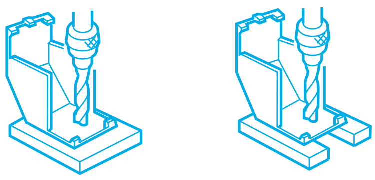
Precaution of additional works

(Take note to be careful to prevent loss of function when making additional machining works to B-SET)
To avoid damage to B-BOX, please do not hesitate to contact us if uncertain of details.

◇ Before additional machining, ensure that bearing and gear portions are covered, so that waste objects will not contaminate it.
◇ For additional machining to Drill hole diameter, do not exceed limitation of drill holes sizes. Refer to reference Figure from page 74.



◇ Put a support plate under the plastic cover when doing additional drilling works.
Put a support plate under the plastic cover.
Plastic cover may break.



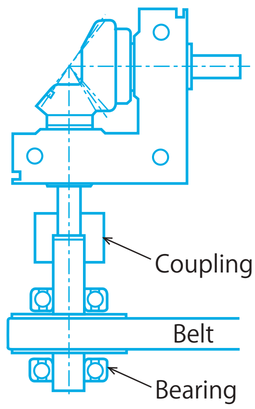
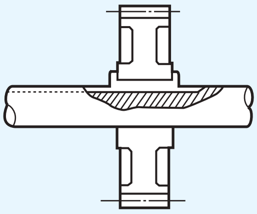
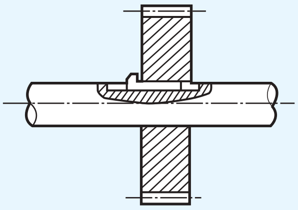
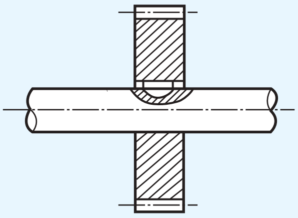
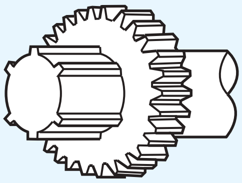
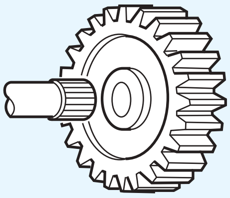
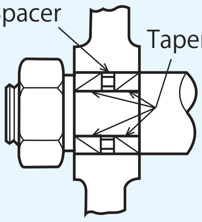
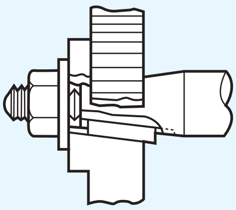


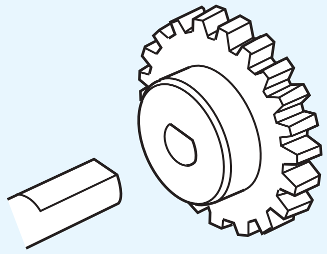
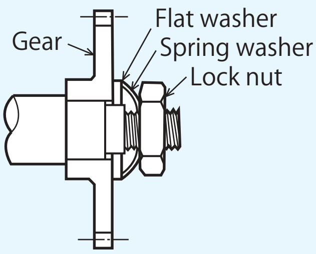
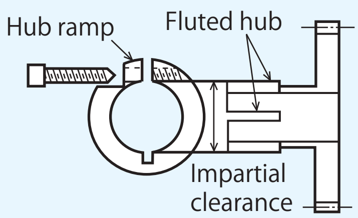
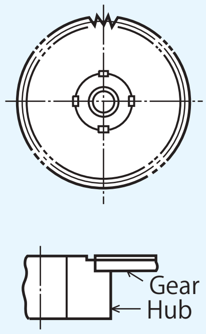
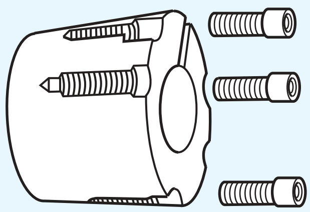
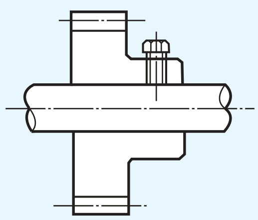
Fig. 10. Reference solution for overhang load

The last characters A, B in the catalogue number for the items indicate different shaft dimensions.
A: Types of standard shaft diameter
B: Types of thicker shaft diameter

2.7 Locking fixtures for gear shaft

Types of element (1)

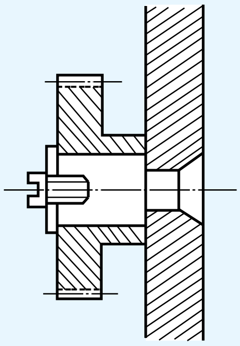
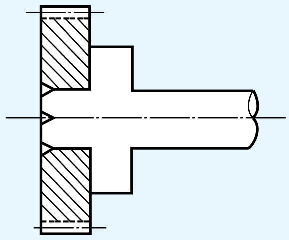
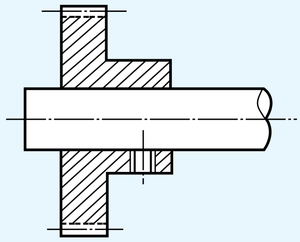
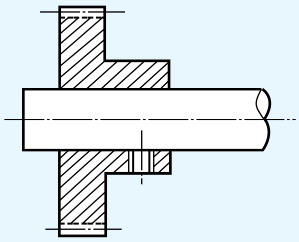
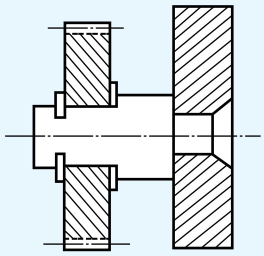
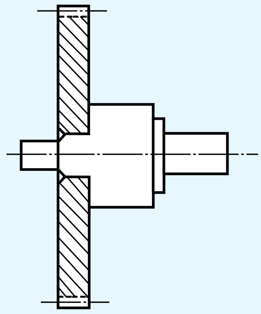
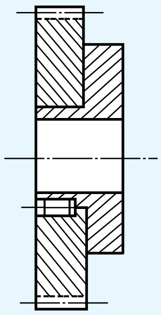
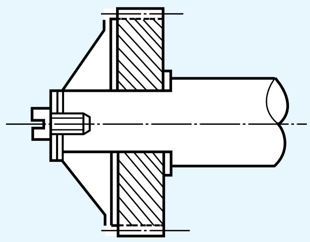
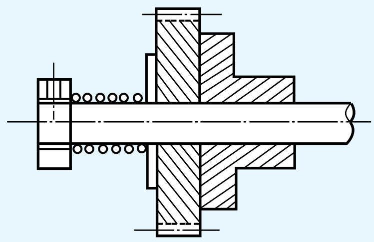
1		**Feather Key.** Location of key can be shifted in axial direction. Fabricate amount of effective length for shifting on the shaft.
2		**Parallel Key.** Used for fixing the gear at designated location.
3		**Woodruff Key.** Used for fixing the gear at designated location.
4		**Spline.** There are types of square and Involute spline. There are types of fixed gear and shifting gear to axial direction. Usage of Spline is for bigger transfer torque compared with Key.
5		**Knurling.** Shaft with Knurling press fit into bore of gear. If designed so that it slips when a specific load is applied, it can be used as a safety device.
6		**Tapering.** When gear is tightened to shaft, parts of tapering extend to interlock. This is for easy installation and improves concentricity between bore and shaft.
7		**Taper shaft.** When using bigger transfer torque, it is necessary to design Key way and Nut for fixing between gear and taper shaft. It is easy to obtain concentricity and to dismantle gear.

8		**D bore.** Made with combination of D-bore and D-cut shaft. Process of D-bore uses round broach with chamfering.
9		**Spring washer.** Lock nut is used for tensioning the spring washer, which is necessary for adjustment. However gear will slip when load to the gear is excessive.
10		**Clamp.** Generally used for dashboard to obtain level clearance between bore and shaft.
11		**Caulking**
12		**Taper bush.** Use for dimension ϕ12.7 or more of bore.
13		**Thread screw.** The most commonly used fixture element due to easy installation. However during operation, beware of slacking of screw and off-centre.

Dr. Masataka Senba, “Miniature size gear” partially extracted from newspaper company Nikkan Kogyo Shimbun, 1969.

Types of element (2)

For small module

① Fitted with Washer and screw	② Fitted with Caulking	③ Fitted with Pinning
		
④ Fitted with Thread screw	⑤ Fitting with Clamping washer	⑥ Fitting with Pressure punching
		
⑦ Fitting with Pressure pinning	⑧ Fitting with flat spring	⑨ Fitting with coiled spring
		

Chapter 3 Gear material and Heat treatment

3.1 Selecting gear material

When load to flank of gear is excessive, wearing off (pitting) of flank may occur easily. It is necessary to select material with greater strength of surface durability therefore the case hardening steel is recommended due to higher hardness.

The impact to flank of gear during operations causes damage to the gear tooth, therefore it is necessary to select steel with higher bending strength. Selected material should be able to apply induction hardening after quenching and annealing treatment. Such selection emphasizes on core hardness instead of surface strength.

Take note of the following while selecting material with manufacturing expense and productivity in mind.

1) For necessary strength for gear, select the material character by emphasizing on either Surface durability or Bending strength. Generally, ideal material selected for gear tooth should be tough and hard to withstand damaged by the load.
2) Suitable material for machining.
   Pitting occur easily in free cutting steel even after surface treatment is applied to gear. This material is unsuitable for gear even though it has good machinability.
3) Material which is easy to apply heat treatment and little deformation. Even if deformed after applying heat treatment, amount should be stable.
4) Material should be economical and easily obtained.

Table 1 shows the common gear materials used for transferring power.

Table 1. Iron and steel materials used for gear

Name of Standard	JIS number	Materials
Gray iron casting	G5501	FC200, 250, 300, 350
Spheroidal graphite iron casting(1)	G5502	FCD400, 450, 500
Carbon steel forging for general use	G5101	SC410, 450, 480
High tensile strength carbon steel casting and Low alloy steel casting for structural purposes	G5111	SCC3A, 3B SCCrM1, 3, SCNCrM2
Carbon steel for machine structural use	G4051	S38C ～ 58C, S09CK, S15CK, S20CK
Nickel chromium steel	G4102	SNC631, 836, 415, 815
Nickel chromium molybdenum steels	G4103	SNCM625, 630, 439, 447 SNCM220, 415, 420, 616, 815
Chromium steel	G4104	SCr415, 420
Chromium molybdenum steel	G4105	SCM435, 440, 415, 420, 421, 822
Aluminium chromium molybdenum steels	G4202	SACM645
Stainless steel bars	G4303	SUS304, 440C

Note (1) This material includes Ductile Cast Iron and meehanite
Remarks. For Case hardening, it is common to use SCM415 or SNCM415. SNC815 and SNCM815 are suitable for Spiral bevel gear. Please refer to Table 5 (Pg. 58) for Load, Material and its heat treatment.

Characteristics of Polyacetal

Recently, industries prefer to use various engineering plastics for machinery elements. We would like to introduce you to our commercialized KG-Polyacetal gears (one of the engineering plastics). Note that gear strength and heat resistance should be taken into consideration when comparing with metal gears.
There are 2 types of Polyacetal, uniformed formaldehyde and copolymerizated ethylene oxide. The former is called Acetal - homopolymer, the latter is called Acetal - copolymer. Usage condition: mean load and mean speed or less is recommended.
Polyacetal has following features. Please refer to the below.

- Physical characteristics - Thermoplastic resin. Used for extensive mass production. Polyacetal has excellent physical characteristics compared to all other resins.
- Wear characteristics - Regarding Wear proof, Polyacetal is excellent next to Polyamide due to little absorbency.
- Polyacetal has tendency for minute dimensional changes due to minute absorption. It has excellent fluidity and has less remainded strain for mold items.
- Chemical character - There will be no damages to the Polyacetal properties even after soaking it in organic solvent with inorganic drug without mineral acids for 6 months. However, use of phenol is not advisable. Polyacetal is extremely resistant against erosive Alkali. It will not be damaged by industrial lubricating oils, motor-oil, break-oil and even contact to copper material.
- Heat resistant - Polyacetal has excellent heat resistant features.

Table 2. Properties of Polyacetal

	Testing methods	Units	Numerical value
Specific gravity	ASTMD-792	-	1.41
Water absorption (soaked for 24 hour)	ASTMD-570	%	0.22
(60% RH)			0.16
Tensile strength (yield point)	ASTMD-638	N/mm^2	61
Tensile elongation (breaking point)	ASTMD-638	%	40
Modulus of elasticity in tension	ASTMD-638	N/mm^2	2,830
Flexural strength	ASTMD-790	N/mm^2	89
Flexural modulus	ASTMD-790	N/mm^2	2,590
Compressive strength (Deformation of 10%)	ASTMD-695	N/mm^2	103
Shear strength	ASTMD-732	N/mm^2	55
Izod impact value (with notch)	ASTMD-256	J/m	74
Rockwell hardness	ASTMD-785	M scale	78
		R scale	119
Taper abrasion (1kg.CS17 wheel)	ASTMD-1044	mg/100 cycle	14
Coefficient of dynamic friction (for steel)	Westover style friction testing machine	-	0.13
Poisson's ratio		-	0.35
Melting point	DSC analysis temperature 10°C/min	℃	165
Deflection temperature under load (182.4 N/cm^2)	ASTMD-648	℃	110
(45.1N/cm^2)			158
Coefficient of linear expansion	-25 ～ +25℃	$\times 10^{-5}$/℃	9
Combustion property	UL94	-	HB
Dielectric constant (10^2 ～ 10^6Hz)	ASTMD-150	-	3.7
Dielectric dissipation factor (10^2Hz)	ASTMD-150	-	0.001
(10^6Hz)			0.007
Surface resistance	ASTMD-257	Ω	1.0×10^{16}
Volume characteristic resistance	ASTMD-257	Ω·m	1.0×10^{12}

The above properties are for reference only. They are not covered under warranty.
Extract from Nippon Polypenco Co,. Ltd - Polypenco Acetal catalogue.

Features of Polyacetal gear

Strength of plastic gear compared with metal gear excluding external factors is $\frac{1}{6}$ to $\frac{1}{9}$. However, it is necessary to take factors like temperature, humidity and others into considerations.

Table 3. Circumferential speed and Limitation of frictional speed

Lubrication		Without lubricating oil	Lubricating oil
Circumferential speed for Spur and Bevel gears	m/s	6	12
Frictional speed for Worm gear pair	m/s	1	2.5

Lowest usage temperature limitation -38°C

Backlash for plastic gear

Plastic material has extremely smaller thermal conductivity and larger thermal expansion factor compared with metals. Plastic gear pair has higher tendency to change dimension compared with metal gear. Therefore KG has intentionally fabricated wider backlash plastic gears as compared with metal gears.

Combination of gear materials

The combination of materials for plastic gear pair, assuming combination between Polyacetal, metal material factor is 1.0. When combining two Polyacetals, material factor is 0.75. Therefore gear strength for Polyacetal gear pair becomes 75%.

We believe that engagement between Polyacetal and metal gears are best combination.
However, note that maximum surface roughness 6S at flank for metal gear is advised to prevent wear for plastic gears.

3.2 Heat Treatments

Refer to Table 4 for features of heat treatments.

Table 4. Features of Heat treatments

Contents	Induction hardening	Flame hardening	Case hardening	Nitrocarburizing	Nitriding
Materials	Carbon steel with 0.4-0.6% Carbon SCM435, SCM440 SMn443, SNC836 SNCM439, etc.	Carbon steel with 0.4-0.6% Carbon SK5-7, Ductile Cast Iron SCr435, SCr440 SCM435, SCM440, etc.	Carbon steel with below 0.23% Carbon SNC415, SNC815 SCM415, SCM420 SNCM420, etc.	① Low and mean content of Carbon steel ② Carbon, Alloy, Stainless and Cast steels	SACM645 and others For Nitriding process, material should consist of Aluminium and Chromium.
Heat treatment	Put gear into the coil of quenching machine then turn on high power of eddy current to the coil. Overheat the surface of gear and immediately apply jet cooling water to gear for instant cooling.. Long items can be fabricated with quenching by process of heated coil line continued by instant cooling in longitundinal direction.	Economical method of heat treatment compared with others if the Induction hardening was expensive (for small volume and extra large item). Heating only the part you wish to harden by burner to overheat. When surface becomes austenite composition, jet-spray the water for instant cooling. As a result, only this part hardens. Method of tempering is process of using low temperature tempering of 150°-200°C.	Gear together with charcoal and Carbonic barium are seald in a melting pot and heated for 4-8 hours at temperature 900°-950°C. Carbon permeates to gear surface. Use for producing a variety of items in small quantities. Gas carburizing is performed with easy adjustment for amount of Carbon carburizing, Depth of Carburizing has minute scales on the surface of gear. Environmentally friendly and has consistent quality. Process time is shorter than Solid carburizing and suitable for mass production.	① Soak gear with low and medium carbon content into the Salt bath (main constituent is NaCN), to produce film of 0.2mm or below on the gear surface. Processing temperature will be 750° to 900°C and it is suitable for small amount of production. This is an economical method but the salt bath is toxic and hazardous to health. ② Isonite Using method of Salt bath Nitriding NaCNO or Potassium bath to Nitride by nascent Nitrogen. Treatment temperature is 500° to 600°C and last for 24 hours. Effective hardness depth will be 0.015 to 0.020.	Material is modified to the Sorbite composition by quenching. After which the gear is put into the Nitriding furnace. When ammonia gas is injected into furnace with temperature 500° to 900°C, decomposed Nitrogen is absorbed to form a hard layer on the surface of the gear. Treatment hours will span from tens of hours to a few hundred hours depending on the depth of hardness required.
Hardening depth	It is difficult to have the bore, core and section to harden. Generally, steel material suitable for hardening is used to perform quick quenching on the surface. Core area keeps original composition. Harden surface with little oxidation using instant overheat and instant cooling. Perform thermal refining with quenching temperature of 30° to 50°C and provide water cooling to allow the Austenite to diffuse into the gear easily. There is high heat efficiency for direct overheating which causes the hardness of Tooth tip to be higher than Dedendum area of the gear.		Tolerance of case depth less than 0.2mm is difficult for Solid carburizing. Carburizing depth below 0.7mm is not suitable. Regardless of the shape of goods, same layer of hardness can be obtained. Mask the area that does not require hardening to prevent carburizing.	Isonite is an economical surface hardening method that saves time for hardening, self-lubricating and has low coefficient of friction.	There is less strain from heat influence during heat treatment by low temperature. Hardening layer provides surface with better wear resistance, heat resistance and anti-corrosion. Layer of hardening expands gear by 0.02mm to 0.03mm as Nitrogen is absorbed.
Productivity	Hardening to limited parts is possible. Heat treatment duration only takes a few seconds. Automated system is possible. Suitable for mass production.	Hardening to limited parts is possible. Heat treatment duration only takes a few seconds. Simple equipment has inconsistent hardness.	Heat treatment hardens whole body. Long heat treatment duration.	Economical cost and short treatment duration	Heat treatment hardens whole body. Very long heat treatment duration.
Hardness	HS55 ～ 75 HRC41 ～ 56	HS55 ～ 75 HRC41 ～ 56	HS70-85 HRC52 ～ 62	HS88-92 HRC64 ～ 66	HS100 以上 HRC68 以上
Strain	Smaller strain than quenching and tempering.	Larger strain than quenching and tempering.	Larger strain than induction hardening	Minute strain	Minute strain
Cost	Economical cost for mass production	Economical cost	More costly than induction hardening	Economical cost for mass production	Costly
Depth of hardness	0.8 ～ 7mm (Alloyed steel is over 4.0mm)	1 ～ 12mm (Alloyed steel is over 4.0mm)	Solid Carbunizing 0.7 ～ 5mm Gas Carbunizing 0.2 ～ 5mm	0.015 ～ 0.02mm (Specialized steel is 0.1 to 0.2)	0.1 ～ 0.6mm (Uneconomical to use above 0.4)
Feature	Suitable for mass production in simple form Electrically controlled automation system is possible Stable quenching Quenching to limited parts is possible Quenching equipment is expensive	No limitation for size and form Quenching to limited parts is possible Quenching equipment is economical cost Overheating temperature is difficult to control.	Easy to adjust carbon density Uniform depth of Carbunizing Easy to adjust depth of Carbunizing	Beware of polluting, as treated salt is deadly poisonous Vulnerable to impulse load	Excellent wear resistant, heat resistant and anti-corrosion Heat treatment after Nitriding is unnecessary Minute deformation No occurrence of hardening crack
Other applications	Chain wheel Pin	Crankshaft Camshaft	Shaft, Pin, Cam, Bush for Roller chain	Camshaft	Diesel injection nozzle Gauge

There is also Plasma nitriding, which is not mentioned here that causes minute strain compared to other methods by thermal influence while hardening tooth. It has been omitted from this report. Refer to Table 4 for features of Nitration.

3.3 Gear materials and Heat treatments

Refer to Table 5 for suitable materials and its method of heat treatment for load. Also refer to Table 6 for hardness range of heat treatment.

Table 5. Loads, Materials and Heat treatment methods

Load		Material number	Methods of heat treatment
Light load	Light impact load and minute wear off	S35C ～ S45C	Thermal refining (Quenching and Tempering)
	Slight wear resistance needed	S15CK	Carburizing, Quenching and Tempering (Depth of hardness 0.2 to 0.4mm)
Medium load	Medium strength and wear resistance needed	S35C ～ S45C	Induction hardening is lightly applied after Thermal refining. Hardness of Tooth tip is HRC47 to 56 (1)
		SCM415 SCr415	Carburizing, Quenching and Tempering (Depth of hardness 0.6 to 1.0). Surface hardness is from HRC 55 to 60.
	Fatigue strength needed	S40C ～ S45C	Induction hardening(2) is applied after Thermal refining. Depth of hardness should be slightly deeper. Apply Induction hardening to Root diameter. Hardness of Tooth tip surface is HRC47-56(1).
		SCM435 SCM440	Nitriding treatment, Gas nitrocarburizing, Tufftriding and etc. are applied after Thermal refining.
Heavy Load	Special impact resistance if needed	SNC815 SNCM420 SNCM815	Carburizing, Quenching and Tempering. Surface Hardness from HRC 58 to 64
	Wear resistance needed	SNCM420 SCM421 SCM822	Carburizing, Quenching and Tempering. Surface hardness is for HRC 62 and above
	Wear resistance and Fatigue strength needed	S45C S48C	Apply Induction hardening(2) to area of root diameter after Thermal refining. Hardness of Tooth tip is HRC 56-60(1)
Special load	Sand burning resistance needed	Nitriding steel	Apply Nitration treatment after Thermal refining
		Alloyed steel SCM435	Apply Nitration treatment after Thermal refining
	Anti-corrosion needed	Austenite, Ferrite, Martenstic group, Stainless steel	Consider other properties together with Anti-corrosion when selecting suitable heat treatment.
	Heat resistance needed	Fe-Cr-Ni Alloy	Apply suitable Heat treatment as required

Note
(1) Area of tooth flank near Bottomland is HRC 5-10 lower than HRC47-56.
(2) Motor generator system (MG) with low frequency is suitable for relatively large size gear.

Table 6. Hardness of Heat treatment

Name of steels	Material numbers	Hardness for Thermal refining Hs	Full quenching Hs	Induction hard-ening HRC	Surface hardness of Case hardening HRC	Core hardness of Case hardening HB
Nickel-chrome steel	SNC631	37-40	50-55	50-55	-	-
	SNC836	38-42	50-55	50-55	-	-
	SNC415	-	-	-	55-60	217-321
	SNC815	-	-	-	58-64	285-388
Nickel chrome molybdenum steel	SNCM439	43-51	65-70	-	-	-
	SNCM447	45-53	65-70	-	-	-
	SNCM220	-	-	-	58-64	248-341
	SNCM415	-	-	-	58-64	255-341
	SNCM420	-	-	-	58-64	293-375
	SNCM815	-	-	-	58-64	311-375
Chrome steel	SCr415	-	-	-	58-64	217-300
	SCr420	-	-	-	58-64	235-320
Chrome molybdenum steel	SCM435	37-40	45-50	45-50	-	-
	SCM440	38-42	50-55	(50-53)(2)	-	-
	SCM415	-	-	-	58-64	235-321
	SCM420	-	-	-	58-64	262-341
	SCM421	-	-	-	58-64	285-263
Carbon steel	S15CK	-	-	-	55-62(3)	131(4)
	S35C	25-35	35-45	35-40	-	-
	S45C	31-40	45-55	40-45	-	-
	S55C	33-42	55-65	45-50	-	-

Note
(1) Refer to Table 5 for Load, Material and Heat treatment. Core hardness is equivalent to Thermal refining hardness.
(2) Applying Induction hardening to teeth is not advisable.
(3) Hardness is (50-53) for water cooling and 50 - 55 is for oil cooling.
(4) Maximum hardness.

Guide

Table 7. Sizes of tooth and depth of Carburizing

Module mm	Range from m1.0 to m1.5	Range from m1.5 to m2.0	Range from m2.0 to m2.75	Range from m2.75 to m4.0	Range from m4.0 to m6.0	Range from m6.0 to m9.0	Range from m9.0 to m12.0
Depth of Carburizing mm	0.2-0.5	0.4-0.7	0.6-1.0	0.8-1.2	1.0-1.4	1.2-1.7	1.3-2.0

Note: Depth of Carburizing is rough outline for standard value for Gas carburizing. Solid or liquid Carburizing adopts a smaller amount than the above chart.

Chapter 4 Measurement for Tooth thickness

4.1 Method of measurement for Sector span

Put designated Number of teeth at parallel flat face of Tooth thickness of Micrometer and measure its distance. Nipped Number of teeth at flat face is commonly called **Sector span of teeth** z_m. Unlike other methods, reference surface is not needed.
For large Sector span of teeth, deviation in measurement occurs because of influence by Pressure angle deviation and different pressure angle on either sides of tooth. Furthermore, it is common practice to measure a few positions along the whole circumference of the gear to obtain the mean in order to take into consideration Pitch and Profile deviations.
It is the most common measurement method for hobbing process as it is able to measure Sector span of teeth during machining and convert measurement into cutter tool machining adjusting amount easily.

Sector span for Spur gear

(1) Sector span for Standard spur gear W

$$W = m\cos\alpha\{\pi(z_m - 0.5) + z\,\mathrm{inv}\alpha\}$$

(2) Sector span for Profile shifted spur gear W

$$W = m\cos\alpha\{\pi(z_m - 0.5) + z\,\mathrm{inv}\alpha\} + 2xm\sin\alpha$$

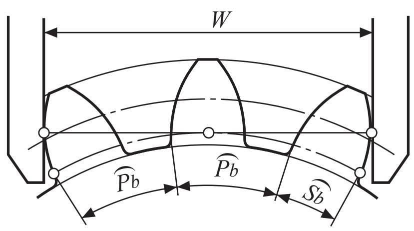
(3) Sector span of teeth

$$z_m = \frac{\alpha z}{180} + 0.5$$

Calculation for Sector span of teeth using nearest integer by above calculated formula.

Hereby
α : Reference pressure angle
m : module
z : Number of teeth
x : Rack shift coefficient

Table 1 shows the values of Sector span and Sector span of teeth with module 1.0 and Reference pressure angle 20°(Rack shift coefficient=0).



$\widehat{P_b}$: Base pitch
$\widehat{S_b}$: Base circular thickness

Fig. 1 Sector span

Table 1. Sector span for Standard spur gear

Adopted for module 1.0 with Pressure angle 20° (Rack shift coefficient $x = 0$).

Unit: mm

z	zm	W	z	zm	W	z	zm	W	z	zm	W	z	zm	W
			41	5	13.8588	81	10	29.1797	121	14	41.5484	161	18	53.9172
			42	5	13.8728	82	10	29.1937	122	14	41.5624	162	19	56.8833
			43	5	13.8868	83	10	29.2077	123	14	41.5765	163	19	26.8973
4	2	4.4842	44	5	13.9008	84	10	29.2217	124	14	41.5905	164	19	26.9113
5	2	4.4982	45	6	16.8670	85	10	29.2357	125	14	41.6045	165	19	56.9253
6	2	4.5122	46	6	16.8810	86	10	29.2497	126	15	44.5706	166	19	56.9394
7	2	4.5262	47	6	16.8950	87	10	29.2637	127	15	44.5846	167	19	56.9534
8	2	4.5402	48	6	16.9090	88	10	29.2777	128	15	44.5986	168	19	56.9674
9	2	4.5543	49	6	16.9230	89	10	29.2917	129	15	44.6126	169	19	56.9814
10	2	4.5683	50	6	16.9370	90	11	32.2579	130	15	44.6266	170	19	56.9954
11	2	4.5823	51	6	16.9510	91	11	32.2719	131	15	44.6406	171	20	59.9615
12	2	4.5963	52	6	16.9650	92	11	32.2859	132	15	44.6546	172	20	59.9755
13	2	4.6103	53	6	16.9790	93	11	32.2999	133	15	44.6686	173	20	59.9895
14	2	4.6243	54	7	19.9452	94	11	32.3139	134	15	44.6826	174	20	60.0035
15	2	4.6383	55	7	19.9592	95	11	32.3279	135	16	47.6488	175	20	60.0175
16	2	4.6523	56	7	19.9732	96	11	32.3419	136	16	47.6628	176	20	60.0315
17	2	4.6663	57	7	19.9872	97	11	32.3559	137	16	47.6768	177	20	60.0455
18	3	7.6324	58	7	20.0012	98	11	32.3699	138	16	47.6908	178	20	60.0595
19	3	7.6464	59	7	20.0152	99	12	35.3361	139	16	47.7048	179	20	60.0736
20	3	7.6604	60	7	20.0292	100	12	35.3501	140	16	47.7188	180	21	63.0397
21	3	7.6744	61	7	20.0432	101	12	35.3641	141	16	47.7328	181	21	63.0537
22	3	7.6885	62	7	20.0572	102	12	35.3781	142	16	47.7468	182	21	63.0677
23	3	7.7025	63	8	23.0233	103	12	35.3921	143	16	47.7608	183	21	63.0817
24	3	7.7165	64	8	23.0373	104	12	35.4061	144	17	50.7270	184	21	63.0957
25	3	7.7305	65	8	23.0513	105	12	35.4201	145	17	50.7410	185	21	63.1097
26	3	7.7445	66	8	23.0653	106	12	35.4341	146	17	50.7550	186	21	63.1237
27	4	10.7106	67	8	23.0794	107	12	35.4481	147	17	50.7690	187	21	63.1377
28	4	10.7246	68	8	23.0934	108	13	38.4142	148	17	50.7830	188	21	63.1517
29	4	10.7386	69	8	23.1074	109	13	38.4282	149	17	50.7970	189	22	66.1179
30	4	10.7526	70	8	23.1214	110	13	38.4423	150	17	50.8110	190	22	66.1319
31	4	10.7666	71	8	23.1354	111	13	38.4563	151	17	50.8250	191	22	66.1459
32	4	10.7806	72	9	26.1015	112	13	38.4703	152	17	50.8390	192	22	66.1599
33	4	10.7946	73	9	26.1155	113	13	38.4843	153	18	53.8051	193	22	66.1739
34	4	10.8086	74	9	26.1295	114	13	38.4983	154	18	53.8192	194	22	66.1879
35	4	10.8227	75	9	26.1435	115	13	38.5123	155	18	53.8332	195	22	66.2019
36	5	13.7888	76	9	26.1575	116	13	38.5263	156	18	53.8472	196	22	66.2159
37	5	13.8028	77	9	26.1715	117	14	41.4924	157	18	53.8612	197	22	66.2299
38	5	13.8168	78	9	26.1856	118	14	41.5064	158	18	53.8752	198	23	69.1961
39	5	13.8308	79	9	26.1996	119	14	41.5204	159	18	53.8892	199	23	69.2101
40	5	13.8448	80	9	26.2136	120	14	41.5344	160	18	53.9032	200	23	69.2241

Note: Calculation for Sector span (W) increases proportionately with module.

Sector span for Helical gear

(1) Sector span for Normal type of Standard helical gear

$$W = m_n \cos\alpha_n \{\pi(z_m - 0.5) + z\,\mathrm{inv}\,\alpha_t\}$$

(2) Sector span for Normal type of Rack shifted helical gear

$$W = m_n \cos\alpha_n \{\pi(z_m - 0.5) + z\,\mathrm{inv}\,\alpha_t\} + 2x_n m_n \sin\alpha_n$$

Sector span of gear z_m

$$z_m = \frac{\alpha_n z_v}{180} + 0.5$$

Calculation for Sector span of teeth uses nearest integer by above calculated formula.

Hereby

- α_n : Normal pressure angle
- α_t : Transverse pressure angle
- m_n : Normal module
- x_n : Normal rack shift coefficient
- z_v : Virtual number of teeth of Spur gear[1] ($z_v = z/\cos^3\beta$)

When measuring the Sector span for Helical gear, put the micrometer perpendicularly to the teeth. Minimum Facewidth is $W\sin\beta$ to prevent contact to flank from coming off.

Example,

When Normal module m_n=4.0, Number of teeth z=19, Normal pressure angle α_n = 20°, Reference cylinder helix angle β=26°42′ , Normal rack shift coefficient x_n=0.4.

Note: (1) Adopted the old Standard term.

① [1]Virtual Number of teeth of Spur gear z_v

$$z_v = z/\cos^3\beta = 19/\cos^3 26°42' = 26.65$$

② Sector span of teeth z_m

$$z_m = \frac{\alpha_n z_v}{180} + 0.5$$
$$= \frac{20 \cdot 26.65}{180} + 0.5 = 3.46 \fallingdotseq 4 \ \text{(Expressed as integer)}$$

③ Transverse pressure angle α_t

$$\alpha_t = \tan^{-1}(\tan\alpha_n/\cos\beta)$$
$$= \tan^{-1}(\tan 20°/\cos 26°42') = 22.16666°$$

④ $\mathrm{inv}\,\alpha_t$ (Involute function for α_t)

$$\mathrm{inv}\,\alpha_t = \tan\alpha_t - \alpha_t = \tan 22.16666° - 22.16666° \cdot \pi/180°$$
$$= 0.020532565$$

⑤ Sector span W

$$W = m_n \cos\alpha_n \{\pi(z_m - 0.5) + z\,\mathrm{inv}\,\alpha_t\} + 2x_n m_n \sin\alpha_n$$
$$= 4 \cdot \cos 20° \{\pi(4 - 0.5) + 19 \cdot 0.0020532565\}$$
$$+2 \cdot 0.4 \cdot 4 \cdot \sin 20° = 43.891 \ \text{(mm)}$$

⑥ Minimum Facewidth required for measurement of Sector span b

$$b = W\sin\beta = 43.891 \cdot \sin 26°42' = 19.72 \fallingdotseq 20 \ \text{(mm)}$$

Therefore, Facewidth above 20mm is needed. If Facewidth is below 20mm, use method of Over balls or Rollers to measure the Tooth thickness.

4.2 Method of measurement with Over balls or Rollers

For spur gear, putting Over balls or Rollers to Spacewidth. External gear is measured by outside dimension of Over ball or Rollers. Internal gear is measured by inner dimension of Over balls or Rollers.
Use method of Over balls or Rollers for Helical gear. Measurement for Internal gear, this method has advantages over others.

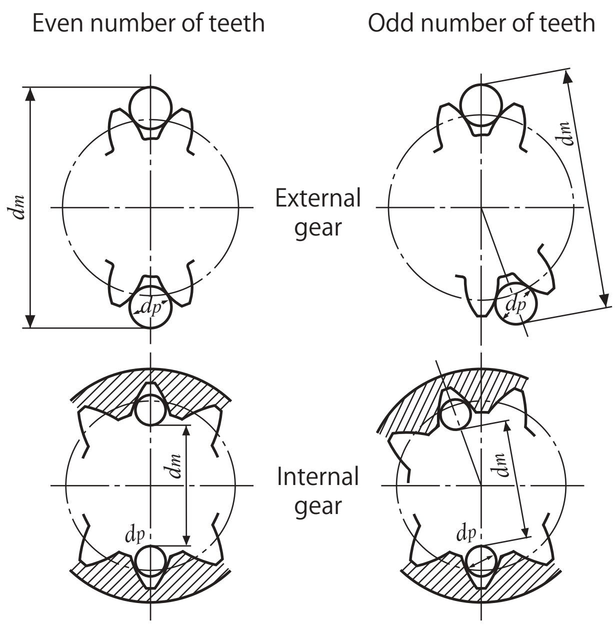


Fig. 2 Method of Over balls or Rollers

*Note for gear with small and odd number of teeth and large helix angle. We recommend using method of Over balls or Rollers due to unstable pins causing inaccurate measurement

Diameter for Over balls or Rollers

Refer to Fig. 3 to find diameter of Over balls or Rollers. Obtain the nearest available diameter from the graph as standard for Over balls or Rollers to measure the gear.

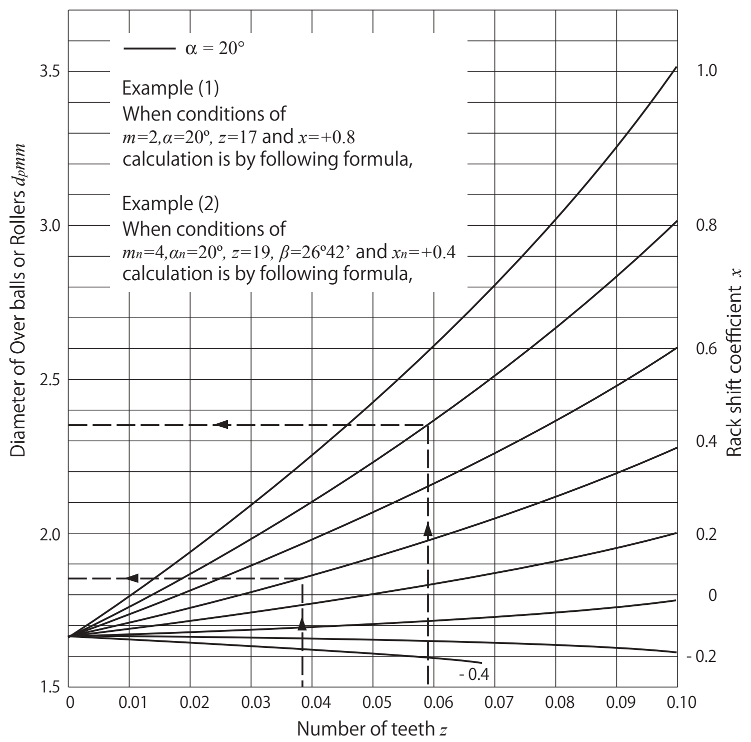


Fig. 3 Graph to find suitable diameter of Over balls or Rollers (for module m_n=1.0)
Increase proportionately by module used. Number of teeth of Helical gear is Virtual number of teeth of Spur gear.

Dimensions of Over balls or Rollers for Spur gear.

For even number of teeth, calculation is by following formula

$$d_m = \frac{zm\cos\alpha}{\cos\phi} + d_p$$

For odd number of teeth, calculation is by following formula

$$d_m = \frac{zm\cos\alpha}{\cos\phi}\cos\frac{90°}{z} + d_p$$

For invϕ, calculation is by following formula

$$\mathrm{inv}\phi = \frac{d_p}{zm\cos\alpha} - \left(\frac{\pi}{2z} - \mathrm{inv}\alpha\right) + \frac{2x\tan\alpha}{z}$$

Hereby

- d_m : Over balls or Rollers dimension(mm)
- z : Number of teeth
- x : Rack shift coefficient
- ϕ : Pressure angle (°) at pin centre
- d_p : Diameter of Practical Over balls or Rollers (mm)
- m : Module (mm)
- α : Reference pressure angle(°)

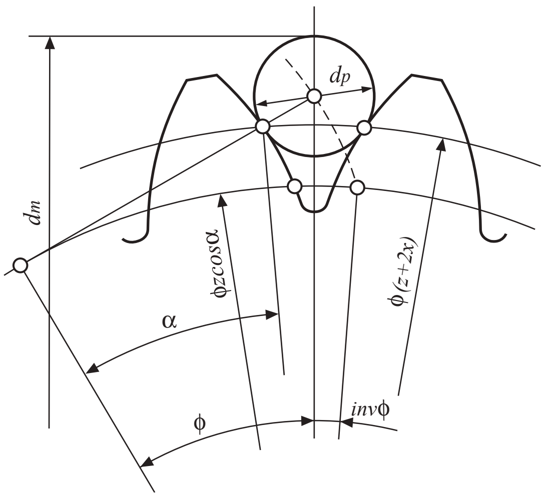


Fig. 4 Dimension of Over balls or Rollers for Spur gear

Example 1, even number of teeth

When module m=2.0, Number of teeth z=30, Reference pressure angle α=20° and Rack shift coefficient x= 0.15. Calculations of Over balls or Rollers dimensions is as follows,

① **Over balls or Rollers dimension d_p**

Refer to Fig. 3, d_p=1.73 multiply by module 2.0 = 3.46(mm)

Use nearest available pin size d_p=3.5(mm) instead of 3.46(mm)

② **Pressure angle ϕ at contact point between flank and Over balls or Rollers**

$$\mathrm{inv}\phi = \frac{3.5}{30 \bullet 2 \bullet \cos 20°} - \left(\frac{\pi}{2 \bullet 30} - \mathrm{inv}20°\right) + \frac{2 \bullet 0.15 \bullet \tan 20°}{30}$$

$$= 0.0282613 \ \ (\mathrm{inv}20° = 0.0149044)$$

$\phi = 24.5388°$ (See page 164 to 167 for Involute function charts)

③ **Over balls or Rollers dimension d_m**

$$d_m = \frac{30 \bullet 2 \bullet \cos 20°}{\cos 24.5388°} + 3.5 = 65.48 \ \ (\mathrm{mm})$$

Example 2, odd number of teeth

Follow example 1 for calculation method, Number of teeth is changed to 29 (Other data remains the same)

① **Over balls or Rollers dimension d_p** d_p=3.5(mm)

② **Pressure angle (°) at pin centre**

$$\mathrm{inv}\phi = \frac{3.5}{29 \bullet 2 \bullet \cos 20°} - \left(\frac{\pi}{2 \bullet 29} - \mathrm{inv}20°\right) + \frac{2 \bullet 0.15 \bullet \tan 20°}{29}$$

$$= 0.0287218$$

$\phi = 24.6645°$ (24°39′52″)

③ **Over balls or Rollers dimension d_m**

$$d_m = \frac{29 \bullet 2 \bullet \cos 20°}{\cos 24.6645°} \bullet \cos\frac{90°}{29} + 3.5 = 63.39 \ \ (\mathrm{mm})$$

Dimension of Over balls or Rollers for Internal gear

Calculation for even number of teeth is by following formula

$$d_m = \frac{zm\cos\alpha}{\cos\phi} - d_p$$

Calculation for odd number of teeth is by following formula

$$d_m = \frac{zm\cos\alpha}{\cos\phi}\cos\frac{90°}{z} - d_p$$

For invϕ, calculation is by following formula

$$\mathrm{inv}\phi = \left(\frac{\pi}{2z} + \mathrm{inv}\alpha\right) + \frac{2x\tan\alpha}{z} - \frac{d_p}{zm\cos\alpha}$$

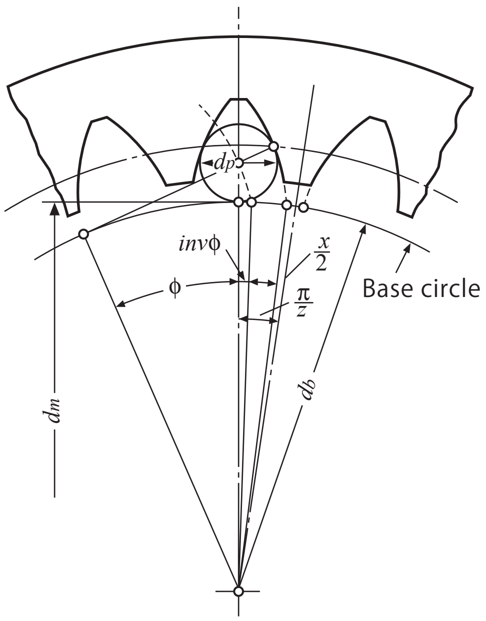


Fig. 5 Over balls or Rollers dimension for Internal gear

Example 1, even number of teeth

When module m=1.0, Number of teeth z=80, Reference pressure angleα = 20° and Rack shift coefficient x= 0.12. Calculations for dimensions of Over balls or Rollers is as follows,

① Over balls or Rollers diameter d_p

Refer to Fig. 3, d_p=1.68 multiply by module 1.0 = 1.68 (mm)

Use nearest available pin size d_p=1.70(mm) instead of 1.68(mm)

② Pressure angle (°) at pin centre

$$\mathrm{inv}\phi = \left(\frac{\pi}{2 \cdot 80} + \mathrm{inv}20°\right) - \frac{1.7}{80 \cdot 1 \cdot \cos 20°} + \frac{2 \cdot 0.12 \cdot \tan 20°}{80}$$

$$= 0.0130174$$

$$\phi = 19.145° \ \ (19°8'42'')$$

③ Over balls or Rollers dimension d_m

$$d_m = \frac{80 \cdot 1 \cdot \cos 20°}{\cos 19.145°} - 1.7 = 77.88 \ \ (\mathrm{mm})$$

Example 2, odd number of teeth

Number of teeth for calculation example 1 is changed to 81 (other data remains the same).

① Over balls or Rollers dimension d_p d_p=1.7(mm)

② Pressure angle (°) at pin centre

$$\mathrm{inv}\phi = \left(\frac{\pi}{2 \cdot 81} + \mathrm{inv}20°\right) - \frac{1.7}{80 \cdot 1 \cdot \cos 20°} + \frac{2 \cdot 0.12 \cdot \tan 20°}{81}$$

$$= 0.0130407$$

$$\phi = 19.156° \ \ (19°9'22'')$$

③ Over balls or Rollers dimension d_m

$$d_m = \frac{81 \cdot 1 \cdot \cos 20°}{\cos 19.156°} \cdot \cos\frac{90°}{81} - 1.7 = 78.86 \ \ (\mathrm{mm})$$

Over balls or Rollers for Straight tooth rack

$$d_m = h'' + \frac{d_p}{2}\left(1+\frac{1}{\sin\alpha}\right)-\frac{\pi m}{4\tan\alpha}$$

Hereby

h'' : Datum line (mm) is from Rack base to Reference line [1].

Helical rack is the same as straight rack at normal section. The above formula can be used. For calculation of Pressure angle α and module m, use α_n and m_n at normal section.

Example,

When module m=3.0, Reference pressure angle α=20° and Datum line h''=32. Calculations of Over balls or Rollers dimensions is as follows,

① Over balls or Rollers diameter d_p

Refer to Fig. 3, d_p=1.68 multiply by module 3.0 = 5.04 (mm)

Use nearest available pin size d_p=5.0(mm) instead of 5.04(mm)

② Over balls or Rollers dimension d_m

$$d_m = 32+\frac{5}{2}\bullet\left(1+\frac{1}{\sin 20°}\right)-\frac{\pi\bullet 3}{4\tan 20°}$$

$$= 35.34 \ \text{(mm)}$$

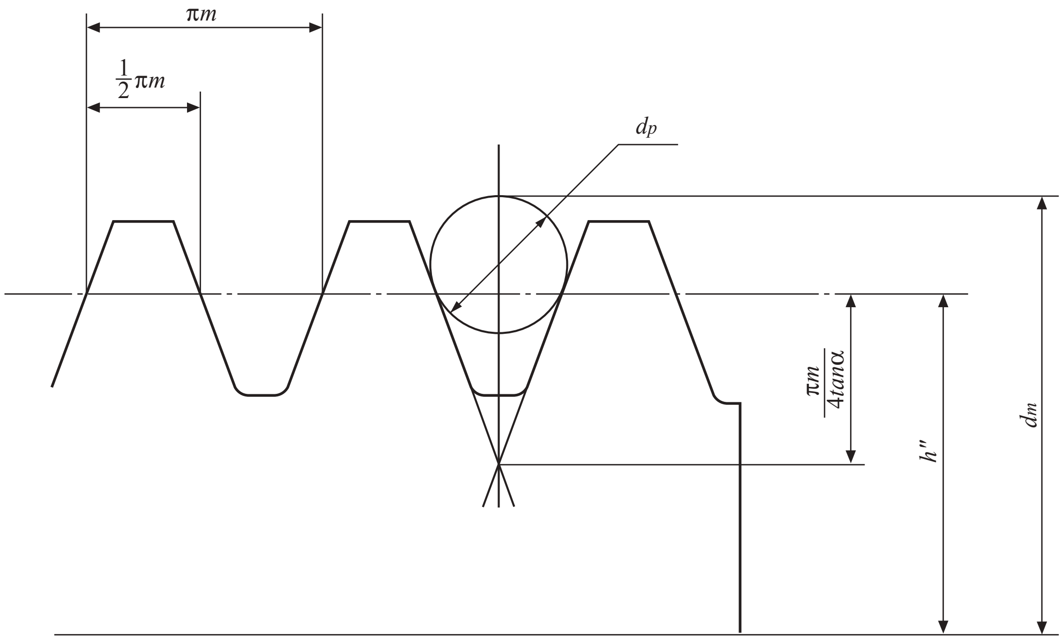


Fig. 6 Over balls or Rollers dimension for Straight rack

Note: (1) Adopted the old Standard term.

Over balls or Rollers dimension for Helical gear

Calculation for even number of teeth is by following formula

$$d_m = \frac{z m_t \cos\alpha_t}{\cos\phi} + d_p$$

Calculation for odd number of teeth is by following formula

$$d_m = \frac{z m_t \cos\alpha_t}{\cos\phi} \cos\frac{90°}{z} + d_p$$

For inv ϕ, calculation is by following formula

$$\mathrm{inv}\phi = \frac{d_p}{z m_n \cos\alpha_n} - \left(\frac{\pi}{2z} - \mathrm{inv}\alpha_t\right) + \frac{2 x_n \tan\alpha_n}{z}$$

Hereby

m_n : Normal module (mm)
α_n : Normal pressure angle(°)
x_n : Normal rack shift coefficient
m_t : Transverse module
α_t : Transverse pressure angle(°)

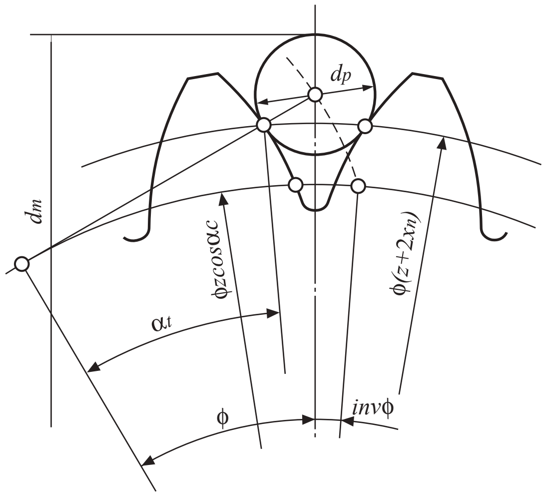


Fig. 7 Over balls or Rollers dimension for Helical gear (shown in axis section)

Example 1, even number of teeth

When module m=2.0, Number of teeth z=36, helix angle β=15°, Normal pressure angle α_n = 20° and Normal rack shift coefficient x_n= 0.05. Calculations of Over balls or Rollers dimensions is as follows,

① (1)Virtual Number of teeth of Spur gear z_v

$$z_v = \frac{z}{\cos^3\beta} = \frac{36}{\cos^3 15°} = 39.94 \fallingdotseq 40$$

Note: (1) Adopted the old Standard term.

② Over balls or Rollers diameter d_p

Refer to Fig. 3, d_p=1.7 multiply by module 2.0 = 3.4 (mm)
Use nearest available pin size d_p=3.5(mm) instead of 3.4(mm)

③ Transverse module m_t

$$m_t = \frac{m_n}{\cos\beta} = \frac{2}{\cos 15°} = 2.07055 \text{ (mm)}$$

④ Transverse pressure angle α_t

$$\alpha_t = \tan^{-1}\left(\frac{\tan\alpha_n}{\cos\beta}\right) = \tan^{-1}\left(\frac{\tan 20°}{\cos 15°}\right)$$
$$= 20.646896° \ (20°38'48'')$$

⑤ Pressure angle (°) at pin centre

$$\mathrm{inv}\phi = \frac{3.5}{36 \bullet 2 \bullet \cos 20°} - \left(\frac{\pi}{2 \bullet 36} - \mathrm{inv}20.646896°\right) + \frac{2 \bullet 0.05 \bullet \tan 20°}{36}$$
$$= 0.025562 \ (\mathrm{inv}20.646896° = 0.0164533)$$
$$\phi = 23.77°$$

(23°46′12″ See page 164 to 167 for Involute function charts)

⑥ Over balls or Rollers dimension d_m

$$d_m = \frac{36 \bullet 2.07055 \bullet 20.646896°}{\cos 23.77°} + 3.5 = 79.72 \text{ (mm)}$$

Example 2, odd number of teeth

Number of teeth for calculation example 1 is changed to 35 (other data remains the same).

① (1)Virtual Number of teeth of Spur gear z_v

$$z_v = \frac{z}{\cos^3\beta} = \frac{35}{\cos^3 15°} = 38.84 \fallingdotseq 39$$

② Over balls or Rollers diameter d_p

Refer to Fig. 3, d_p=1.7 multiply by module 2.0 = 3.4 (mm)
Use nearest available pin size d_p=3.5(mm) instead of 3.4(mm)

③ Transverse module m_t

m_t=2.07055(mm)
Calculations is the same as above in even number of teeth part ③

④ Transverse pressure angle α_t

α_t=20.646896° (20°38'48")
Calculations is the same as Example 1, even number of teeth part ④

⑤ Pressure angle at pin centre (°)

$$\mathrm{inv}\phi = \frac{3.5}{35 \bullet 2 \bullet \cos 20°} - \left(\frac{\pi}{2 \bullet 35} - \mathrm{inv}20.646896°\right) + \frac{2 \bullet 0.05 \bullet \tan 20°}{35}$$
$$= 0.025822 \ (\mathrm{inv}20.646896° = 0.0164533)$$
$$\phi = 23.8465°$$

(23°50′47″ See page 164 to 167 for Involute function charts)

⑥ Over balls or Rollers dimension d_m

$$d_m = \frac{35 \bullet 2.07055 \bullet \cos 20.646896}{\cos 23.8465°} \bullet \cos\frac{90°}{35}$$
$$+3.5 = 77.57 \text{ (mm)}$$

Over balls or Rollers dimension for Worm gear

To obtain Over balls or Rollers dimension for Worm gear, introduce following methods,

1) Substituted three wire method from thread screw used for measurement.
2) Use same calculation method of Rack for Worm gear.
3) Use same calculation method of Helical gear for Worm gear. However, only formula 3) is introduced.

$$d_m = d(1+A) + d_p + Ae^2 d\left\{\frac{1}{2(1+A)} + \frac{3}{8}e^2\right\} - A^2 e^4 d$$

$$A = \frac{1}{d\sin\gamma_b}\left(d_p - \frac{p_x}{2}\cos\gamma_b\right) \qquad e = \frac{z\,p_x}{\pi d}\cot\gamma_b$$

$$p_x = \frac{\pi m_n}{\cos\gamma} \qquad \gamma_b = \tan^{-1}\left(\frac{\tan\gamma}{\cos\alpha_t}\right)$$

$$\alpha_t = \tan^{-1}\left(\frac{\tan\alpha_n}{\sin\gamma}\right)$$

Hereby

- d : Pitch diameter of Worm gear (mm)
- z : Number of thread of Worm gear
- p_x : Axial pitch of Worm gear (mm)
- γ : Reference cylinder lead angle (°)
- γ_b : Base cylinder lead angle (°)

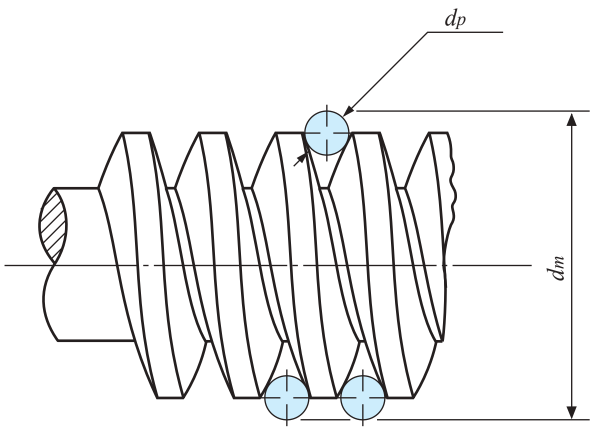


Fig. 8 Over balls or Rollers dimension of Worm gear

Example

When module m_n=2.0, Number of thread z=1, Pitch diameter of Worm gear d = 31, Normal pressure angle α_n=20° and Reference cylinder lead angle γ= 3°42′(3.7°). Calculations of Over balls or Rollers dimensions of Worm gear is as follows.

① **Over balls or Rollers diameter d_p**

Refer to Number of teeth(10 to ∞) in Fig. 3, d_p=1.68 and multiply by module 2.0 = 3.36 (mm)

Use nearest available pin size d_p=3.4 (mm) instead of 3.36(mm)

② **Transverse pressure angle α_t**

$$\alpha_t = \tan^{-1}\left(\frac{\tan 20°}{\sin 3.7°}\right) = 79.9459°$$

③ **Base cylinder lead angle γ_b**

$$\alpha_t = \tan^{-1}\left(\frac{\tan\alpha_n}{\cos\beta}\right) = \tan^{-1}\left(\frac{\tan 20°}{\cos 15°}\right)$$

④ **Axial pitch**

$$p_x = \frac{\pi \bullet 2}{\cos 3.7°} = 6.2963$$

⑤ A

$$A = \frac{1}{31\sin 20.3256°}\left(3.4 - \frac{6.2963}{2}\cos 20.3256°\right)$$

$$= 0.04159$$

⑥ e

$$e = \frac{1 \bullet 6.2963}{\pi \bullet 31}\cot 20.3256°$$

$$= 0.17453$$

⑦ **Over balls or Rollers dimension**

d_m = 35.71 (mm) (Substitution method omitted)

4.3 Measurement method with Gear tooth vernier

Measurement method based on Tip circle, measure Chordal tooth thickness upon Pitch cylinder. Refer to Fig. 9 for measurement method based upon Gear tooth vernier calipers. Fix gear tooth vernier calipers at theoretical value of Tooth depth h and measure deviation between actual Chordal tooth thickness s and its theoretical value. This is a time-honored measurement method with low accuracy due to influence from inconsistent measurement and Jaw conditions.

For Spur gear, calculation is by following formula :

$$\bar{h} = \frac{mz}{2}\left\{1 - \cos\left(\frac{\pi}{2z} + \frac{2x\tan\alpha}{z}\right)\right\} + \frac{da - d}{2}$$

$$\bar{s} = mz\sin\left(\frac{\pi}{2z} + \frac{2x\tan\alpha}{z}\right)$$

Hereby

$\bar{h}$	: Chordal addendum	$\bar{s}$	: Chordal tooth thickness
m	: Module	z	: Number of teeth
α	: Reference pressure angle	x	: Rack shift coefficient
d_a	: Tip (outside) diameter	d	: Reference diameter

Refer to Table 2. Below chart shows $\bar{h}$: Chordal addendum and $\bar{s}$: Chordal tooth thickness for gear with module 1.0 and Rack shift coefficient x=0.

For Helical gear, use Normal surface to measure module, Pressure angle and Rack shift coefficient using value of Normal. Number of teeth z uses (1)Virtual number of teeth for Spur gear.

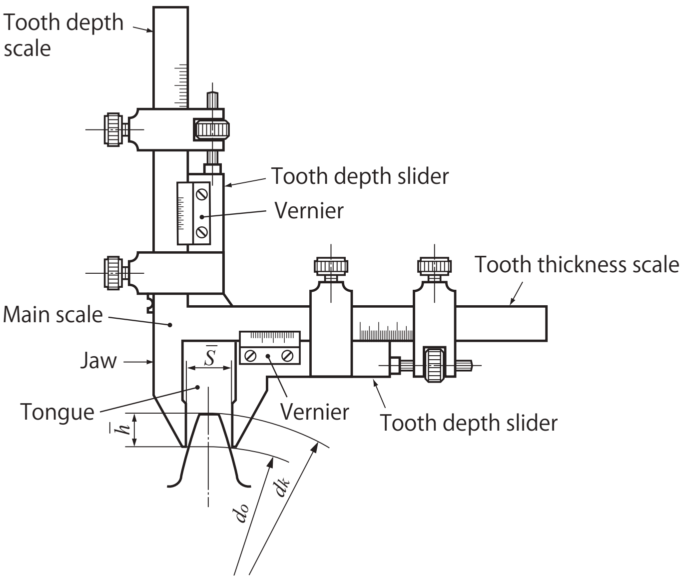


Fig. 9 Measurement with gear tooth vernier calipers

Table 2. Chordal tooth thickness for standard gear

z	$\bar{h}$ mm	$\bar{s}$ mm	z	$\bar{h}$ mm	$\bar{s}$ mm
12	1.0513	1.5663	35	1.0176	1.5703
13	1.0474	1.5670	40	1.0154	1.5704
14	1.0440	1.5675	45	1.0137	1.5705
15	1.0411	1.5679	50	1.0123	1.5705
16	1.0385	1.5683	60	1.0103	1.5706
17	1.0363	1.5686	70	1.0088	1.5706
18	1.0342	1.5688	80	1.0077	1.5707
19	1.0324	1.5690	90	1.0069	1.5707
20	1.0308	1.5692	100	1.0062	1.5707
22	1.0280	1.5695	120	1.0051	1.5708
24	1.0257	1.5697	150	1.0041	1.5708
26	1.0237	1.5698	200	1.0031	1.5708
28	1.0219	1.5700	∞	1.0000	1.5708
30	1.0206	1.5701			

Values of $\bar{h}$ and $\bar{s}$. (m=1, x=0)

Note: (1) Adopted the old Standard term.

< Reference > Gear analysis method

How to obtain module, Pressure angle and Rack shift coefficient for Involute spur gear,

There are various methods on how to obtain module and Pressure angle for Involute spur gear. Method introduced here is by Base pitch measurement.

There is a method of using Sector span to measure the Base pitch.

For Sector span, assuming n of Number of teeth is E_n. Reduce one tooth from n is E_{n-1}. Therefore Base pitch P_b is by following formula.

$$p_b = E_n - E_{n-1} \quad (1)$$
$$= \pi m \cos\alpha$$

Table 3. Base pitch

m \ α_0	14.5°	20°	22.5°	25°
1	3.042	2.952	2.902	2.847
1.25	3.802	3.690	3.628	3.559
1.5	4.562	4.428	4.354	4.271
1.75	5.323	5.166	5.079	4.983
2	6.083	5.904	5.805	5.695
2.25	6.843	6.642	6.531	6.406
2.5	7.604	7.380	7.256	7.118
2.75	8.364	8.118	7.982	7.830
3	9.125	8.856	8.707	8.542
3.25	9.885	9.594	9.433	9.254
3.5	10.645	10.332	10.159	9.965
3.75	11.406	11.070	10.884	10.677
4	12.166	11.809	11.610	11.389
4.5	13.687	13.285	13.061	12.813
5	15.208	14.761	14.512	14.236
5.5	16.728	16.237	15.963	15.6.60
6	18.249	17.713	17.415	17.0.84
6.5	19.770	19.189	18.866	18.507
7	21.291	20.665	20.317	19.931
8	24.332	23.617	23.220	22.778
9	27.374	26.569	26.122	25.625
10	30.415	29.521	29.025	28.473

Compare Base pitch calculation result by formula (1) with Base pitch in Table 3.

Example for Spur gear: Calculate module, Pressure angle and Rack shift coefficient for Number of teeth = 12 and Tip (outside) diameter =29.9mm.

Result of measurement for Sector span (Z_m) was as follows

Z_m= Two (2) teeth of Sector span $E_2 = 9.855$ mm

Z_m= Three (3) teeth of Sector span $E_3 = 15.758$ mm

Therefore calculate P_b by following formula (1)

$$p_b = E_3 - E_2$$
$$= 15.758 - 9.855$$
$$= 5.903\text{mm}$$

With reference to Base pitch chart in Table 3, module is 2.0 mm and Pressure angle is 20°.

Calculation formula for Rack shift coefficient

Calculation for Sector span W is by following formula

$$W = m\cos\alpha\{\pi(Zm - 0.5) + z\mathrm{inv}\alpha\} + 2\,xm\sin\alpha$$

Calculating Sector span W'' for Standard spur gear with pressure angle 20° is by following formula:

$$W' = m\cos\alpha\{\pi(Zm - 0.5) + z\mathrm{inv}\alpha\}$$
$$= m(0.01400554z + 2.95213zm - 1.47606)$$

Calculating Sector span W''' for Rack shifted spur gear with pressure angle 20° is by following formula:

$$W'' = W'[\text{standard}] + 2\,xm\sin\alpha$$
$$W'' = W'[\text{standard}] + 0.68404xm \quad (2)$$

[Standard] is abbreviation of Standard spur gear.

From above formula (2), calculation for Rack shift coefficient x is as follows:

$$x = \frac{W'' - W'}{0.68404m} \quad (3)$$

Therefore, results are W'''=9.855, W'' = 9.193. Rack shift coefficient x is 0-.484.

$$x = \frac{9.855 - 9.193}{2 \times 0.68404}$$
$$= 0.484$$

Chapter 5 Deviation for Gear and its measurement method

5.1 Correlation of deviations

Gear deviations are classified with individual and composite deviations. Shown in Fig. 1, individual deviation is a three-dimensional deviation in the directions as follows

1) Direction of Tooth depth refers to shape of Tooth profile and length of Tooth depth.
2) Direction of Tooth trace refers to inclination and unevenness of Tooth trace.
3) Direction of Tooth thickness refers to thickness of tooth and Tooth space.

These three types of individual deviations are measured by taking apart a three-dimensional deviation into a two-dimensional deviation. However, these individual deviations are correlated and the extent of correlation differs between the methods of production and measurement. Correlations of these individual deviations are shown in Fig. 2. Pay close observation to the strong correlation between Runout and other deviations in Table 2.
Another method to obtain measurements for Total deviation is to simultaneously measure three dimensions.

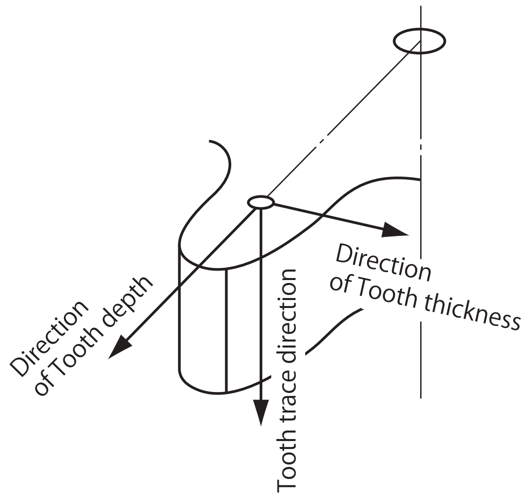


Fig. 1 Theory for three-dimensional deviations

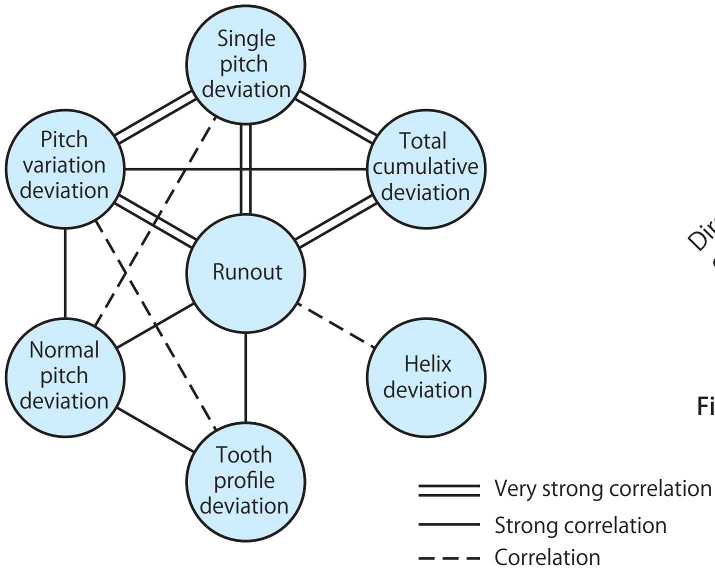


Fig. 2 Correlation with individual deviation (Ground spur gear)

5.2 Tooth profile deviations

The degree of **accuracy for Spur and Helical gears** in JIS B1702 (old). Standard is defined by "sum of positive (+) deviation and negative (-) deviations from actual tooth profile within the Tooth profile evaluation range measured in perpendicular direction to actual Tooth profile and correct Involute which crosses the intersection point of Pitch circle." . This explanation for Tooth profile deviation is for Axis profile only.
Definition of Profile evaluation range is range of Tooth profile curve when engaged with Mating gear. In short, not all range of Tooth depth engages with mating gear. Range in actual motion excludes Tooth tip and Dedendum.
However, Tooth profile deviation does not include parts with Profile modification. Refer to Fig. 3 for Tooth profile deviation.
Tooth profile deviation has Pressure angle and unevenness deviations. Normally these two deviations appear at the same time.
Tooth profile deviation is always indicated as maximum value in Tooth profile evaluation range. Allowable deviation is listed in System of accuracy defined in JIS B1702 (old). In JIS B1752 (old), **method of measurement for Spur and Helical gears** have following 3 types stipulated.

1) **Base disk method**: In accordance with gear specifications, use Base disk with diameter equivalent to that its Base circle to measure. (Refer to Fig.4)
2) **Base circle adjustment method**: Use Base circle with mechanism that enlarges or reduces the measuring pointer in accordance with the diameter ratio between native Base circle and gear Base circle. (Fig. 5)
3) **Operation method**: Use digital coordinates to measure the Tooth profile and compare with Theoretical involute profile to work out deviation.

There are other methods of Pitch disk, Master cam and Optics which are available but omitted here.

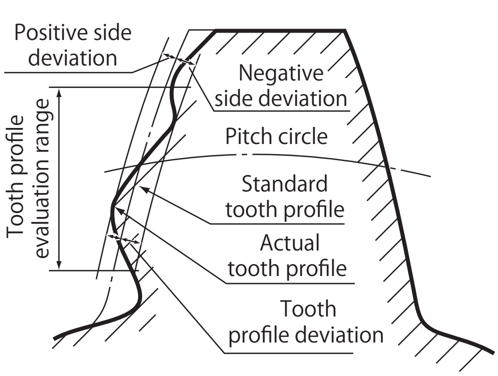


Fig. 3 Tooth profile deviation

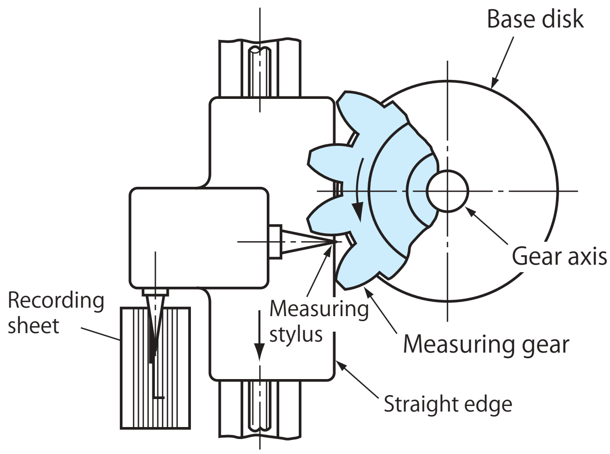


Fig. 4 Measuring method for Base disk

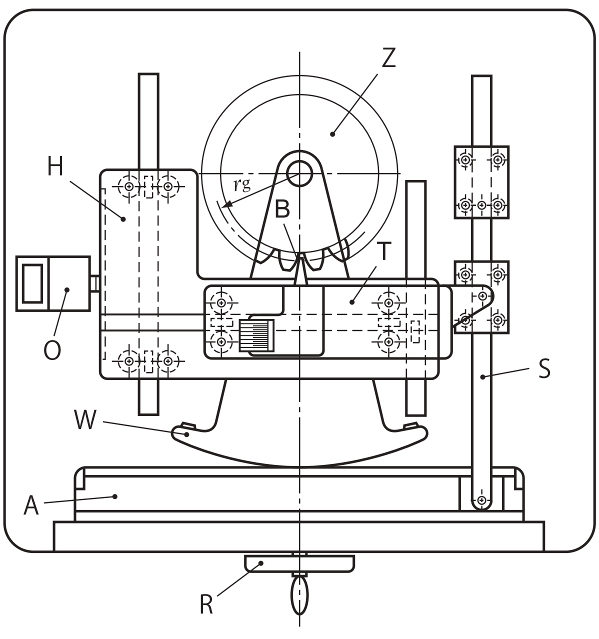


Fig. 5 Mechanism for Base circle adjustment equipment

5.3 Helix deviations

Helix deviation is the difference in dimension on Pitch cylinder measurement range between Actual tooth trace curved line and Theoretical curved line as defined in JIS B 1702.
For gear accuracy, only Helix deviation is classified by Facewidth and not by module or Pitch diameter. The Measuring stylus measures the Spur gear in axial direction. For Helical gear, gear is rotating while measuring pointer follows helix angle for measuring in axial direction.
Refer to Fig. 9, shows the measurement of Helix deviation for Helical gear.

In JIS B1702, 2 (two) regulation methods of measurement are as follows.

1) **Tooth trace creation method**: Refer to Fig. 9, measuring method by rotating gear on measurement stylus and either the measuring pointer or gear shifts in axial direction in the range of Theoretical tooth trace effective distance at Pitch cylinder.
2) **Operation method**: Coordinate of Tooth profile is measured digitally and compared with theoretical value of Involute tooth profile to calculate the deviation.

In addition, standard lead model can be used for comparison measurement method.

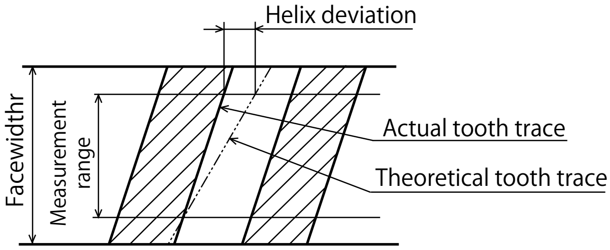


Fig. 6 Helix deviation

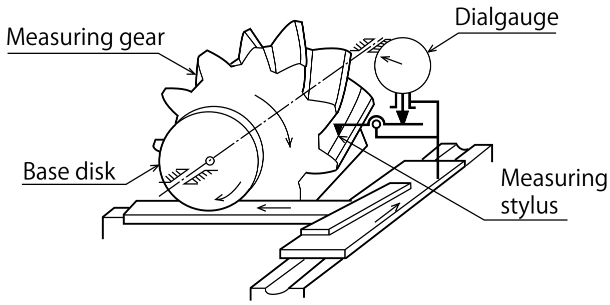


Fig. 9 Measurement for Helix deviation

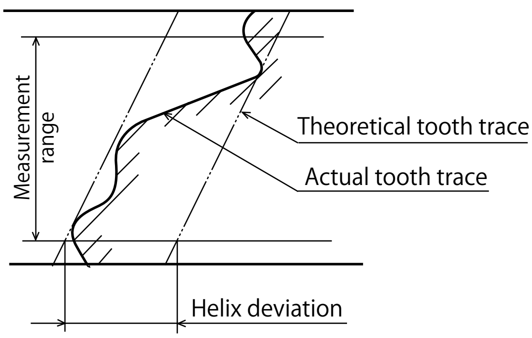


Fig. 7 Helix deviation (Wavy tooth)

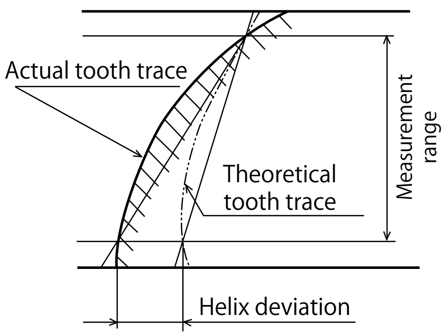


Fig. 8 Helix deviation (Crowned tooth)

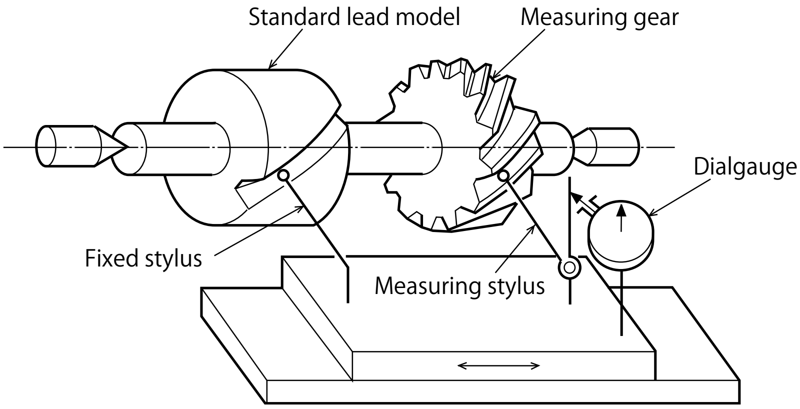


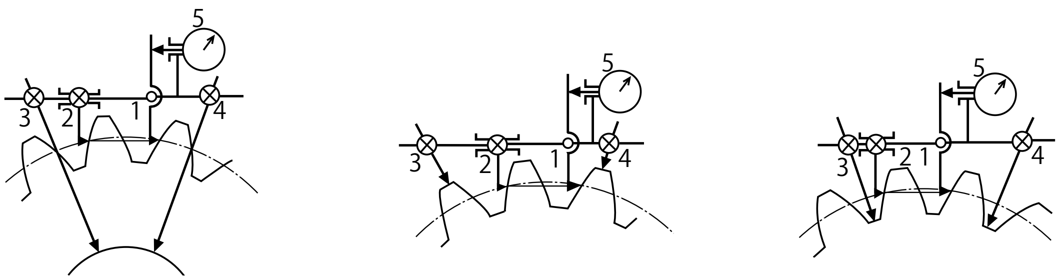
Fig. 10 Comparison measurement mechanism

5.4 Pitch deviations

Accuracy of pitch is important for high speed rotating gear. The deviations of Single pitch, Total cumulative pitch and Normal pitch are defined in JIS B 1702-1. Therefore each Allowable pitch deviation in each system of accuracy is stipulated.

Also, in the JIS B 1752, large number of measurement methods for Pitch deviation are stipulated. For examples, method of measurement for Circle pitch, there are In-line distance method (Refer to Fig. 11) and Angle device method (Refer to Fig. 12). For measurement method of Base pitch, there are Manual system method and Revolving centre method (Refer to Fig. 13).

(a) Revolving centre method (b) Tip cylinder method (c) Root cylinder method



1: Measuring stylus 2: Fixed stylus(3) 3, 4: Locating stylus 5: Dialgauge

Note (3) For fixed stylus, dialgauge is included to be used to establish zero location.

Fig. 11 Measurement for Circular pitch (In-line distance method)

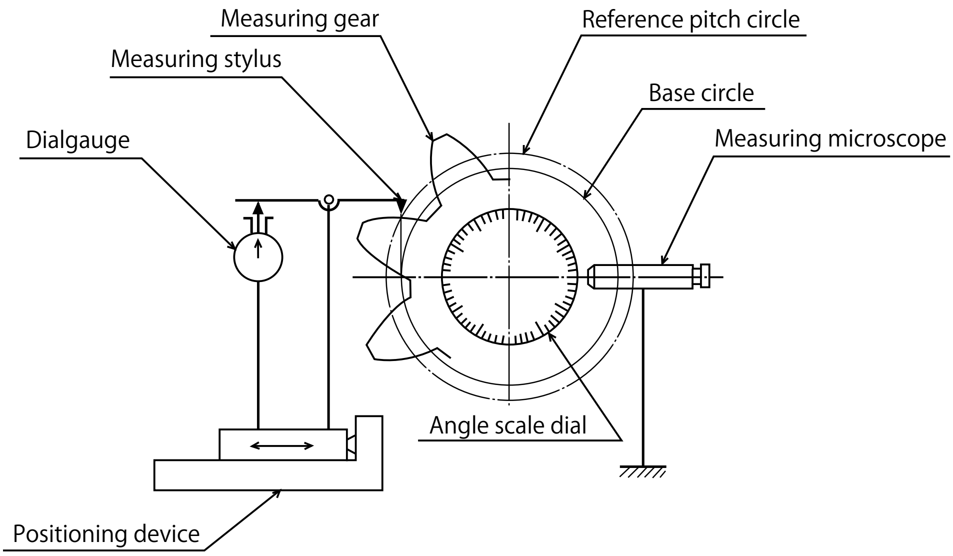
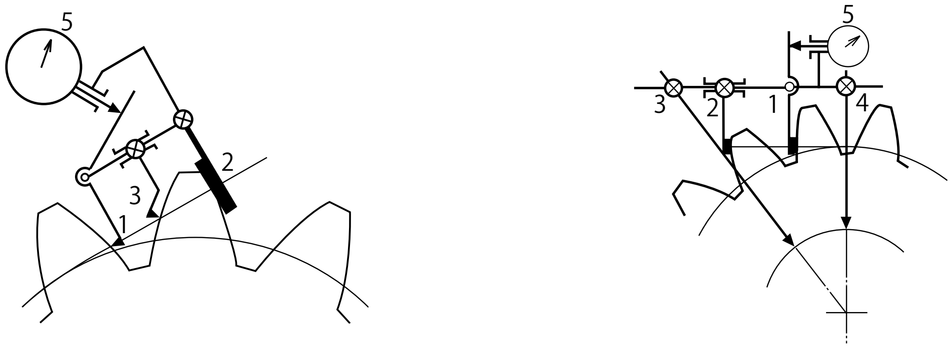


Fig. 12 Measurement methods for Circular pitch (Angle device method)

(a) Manual method (b) Revolving centre method



1: Measuring stylus 2: Fixed stylus 3, 4: Locating stylus 5: Dialgauge

Fig. 13 Measurement methods for Base pitch

5.5 Runout

Runout defined in **JIS B 1702 (old)** as "maximum difference in radius direction when contact pieces such as Over balls or Rollers are put to Tooth space near the Pitch circle. In short, the amount of off-centre measured between gear and axis. Deviations of Pitch, Pressure angle, Profile and others can influence Runout. If larger Runout occurs, it should be related to such deviations. To maintain minimum Runout, note that accurate bore tolerance is necessary. Pay special attention to chucking gear material to hobbing machine.

Below 1) and 2) are defined in **JIS B 1752 (old)**.
1) Use Over balls or Rollers for measurement
2) Measurement of pitch
Refer to Fig. 14, shows measurement of Over balls or Rollers.

Select diameter of the Over balls or Rollers to makes contact near the centre of effective tooth depth of the gear which is measured. Please refer to Fig. 3 in Chapter 4 (page 63) for graph to find suitable diameter of Over balls or Rollers (for module m_n=1.0).
For measurement, put Over balls or Rollers at the centre of Facewidth.

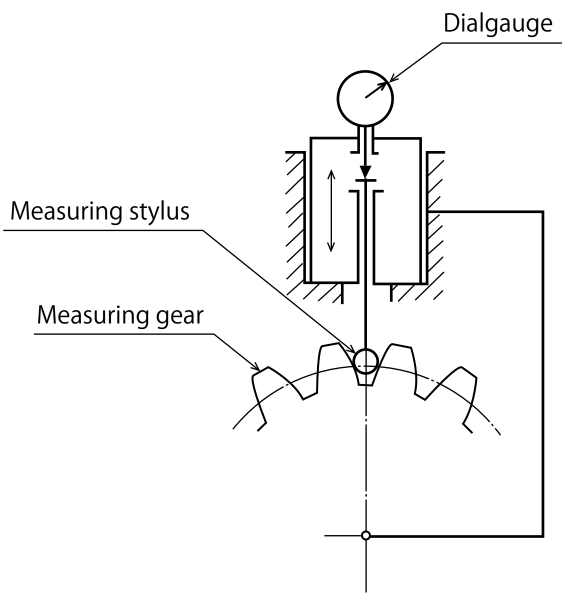


Fig. 14 Measurement of Runout

5.6 Radial composite deviation

Deviation and methods of measurements are introduced in 5.2 ～ 5.5 for individual deviations. These measurement methods are analyzed in two dimensions. On the other hand, perform engagement testing by engaging the gear to be measured with Mating gear or Cylindrical master gear and rotate to check gear condition. Even though accuracy of a gear is proper, problems do not occur unless the gear is actually engaged and rotated. Therefore gear performance is checked by engagement test.

Radial composite deviation is defined in **JIS B 1752 (old)**,

(a) For a gear on its own: Engaged with Cylindrical master gear without backlash and rotated to check for fluctuation of centre distance.

(b) For a gear pair: Engaged together (hereinafter called specific Gear pair) without backlash and rotated to check for fluctuation of centre distance.

Refer to Fig. 15 for example of Radial composite deviation and refer to Fig. 16 for method of measurement for Radial composite deviation.

Another deviation is Tooth-to-tooth radial composite deviation, which is omitted here.

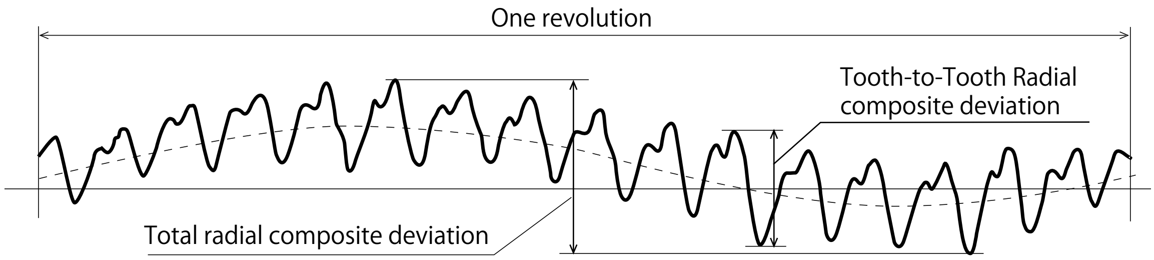


Fig. 15 Radial composite deviation (for gear on its own)

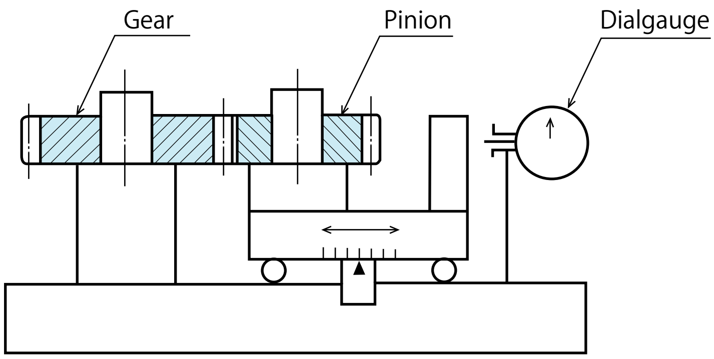


Fig.16 Method of measurement for Radial composite deviation

5.7 Precision of Spur and Helical gears

1. Introduction

In order to make **JIS Standard** consistent with **ISO Standard, JIS B 1702 (old)** : 1995 (Accuracy for the Spur and Helical gears) which had been used for a long time has been abolished and it was enacted as two regulations: **JIS B 1702-1**: 1998(Cylindrical gears- System of accuracy and Classification Article 1: Definition of Deviation and Allowable value of deviation relavent to corresponding Tooth flanks) and **JIS B 1702-2**: 1998 (Cylindrical gears - System of accuracy and Classification Article 2: Definition and Allowable values of deviation relevant to Radial composite deviation and Runout).
When comparing **JIS B 1702 (old)** with the **JIS B 1702-1 or 2**, classifications of module and Reference diameter (called Pitch diameter of **old JIS** B1702) are different. For example, **class 4 in JIS B 1702 (old)** may not be able to correspond to JIS B1702-1 or 2. The rough outline of **System of accuracy in JIS B1702-1 or 2** = System of accuracy in **JIS B 1702 (old)** class plus 4. However certain range of small or large Number of teeth are unable to correspond to above rough outline classification.
In due time, many standards established of **JIS** and **JGMA** based on the **JIS B 1702 (old)**. It will be revised to a new edition based on **JIS B 1702-1 or 2**. However, there are certain areas that cannot be resolved immediately.
Therefore, this new edition of KG catalogue indicates System of accuracy with comparison table between the JIS B1702-1 and JIS B1702 (old). Please refer to following System of accuracy. Firstly find gear accuracy from **JIS B1702-1** and compared with **JIS B 1702 (old)**. Secondly, use these correspondances to compare to other **JIS** and **JGMA** standards to obtain the total of each Reference or Allowable tolerance.
To search for accuracy of gears outside the range of KG-catalogue, please verify with JIS B 1702-1: 1998 and JIS B 1702-2:1998 (old and new) standard, as KG-catalogue does not cover all accuracy.

2. Types of Deviations for Allowable value compared between old and new JIS.

Extracted **JIS B 1702-1**: 1998 and **JIS B1702-2** :1998 (Refer to Table 1 to 11)

(1) Single pitch deviation
(2) Total cumulative pitch deviations
(3) Total profile deviation
(4) Runout
(5) Total radial composite deviation
(ISO 1328-2: Total radial composite tolerance)
(6) Tooth-to-tooth radial composite deviation

Refer to the following pages for comparison tables of the above 6 types of deviations. New and old JIS standards are classified by module.
It is recommended that the System of accuracy for new **JIS** prefixed with a figure N at the beginning to avoid confusion of new and old **JIS**.

3. Precaution when comparing Helical gear

New JIS uses Normal module to set the Allowable value for each deviations. However **old JIS** uses Transverse module instead. When comparing accuracy between new and old JIS standards for Helical gear of Normal module, calculation of Transverse module m_t is by the following formula from Normal module m_n and Reference cylinder helix angle β.

$$m_t = m_n / \cos\beta$$

4. Total helix deviation (old JIS: Lead error)

Refer to Table 12 to find Total helix deviation as extracted from **JIS B 1702-1**: 1998.

5. Material accuracy of Cylindrical gear.

Refer to Table 13 to 19 for material accuracy of Cylindrical gear.

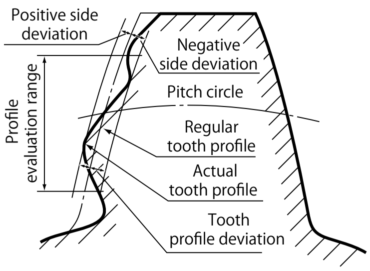


Fig. 17 Tooth profile deviations

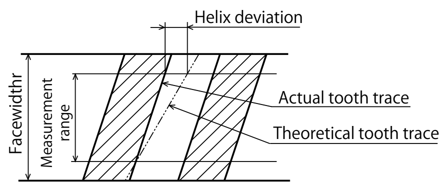


Fig. 18 Helix deviation

Table 1. The Allowable value of each deviation for module 0.5

Unit: μm

Deviations	System of accuracy for JIS B 1702-1 and 2: 1998								System of accuracy for JIS B 1702 and JGMA 116-01							
	No. of teeth	N4	N5	N6	N7	N8	N9	N10	No. of teeth	0	1	2	3	4	5	6
Single pitch deviations	10 - 40	3.3	4.7	6.5	9.5	13	19	26	7 - 12	2	3	5	7	9	13	19
	41 - 100	3.5	5	7	10	14	20	28	13 - 24	3	4	5	7	10	14	20
	101 - 250	3.8	5.5	7.5	11	15	21	30	25 - 50	3	4	6	8	11	16	22
									51 - 100	3	4	6	9	13	18	25
									101 - 200	4	5	7	10	14	20	29
Total cumulative pitch deviations	10 - 40	8	11	16	23	32	45	64	7 - 12	9	13	19	26	37	52	75
	41 - 100	10	14	20	29	41	57	81	13 - 24	10	14	20	29	41	57	81
	101 - 250	13	18	26	37	52	74	104	25 - 50	11	16	22	32	45	63	90
									51 - 100	13	18	25	36	50	71	100
									101 - 200	14	20	29	40	57	80	115
Total profile deviation	10 - 40	3.2	4.6	6.5	9	13	18	26	All range	2	3	5	7	10	14	20
	41 - 100	3.6	5	7.5	10	15	21	29								
	101 - 250	4.1	6	8.5	12	17	23	33								
Runout	10 - 40	6.5	9	13	18	25	36	51	7 - 12	7	9	13	19	26	37	52
	41 - 100	8	11	16	23	32	46	65	13 - 24	7	10	14	20	29	41	57
	101 - 250	10	15	21	29	42	59	83	25 - 50	8	11	16	22	32	45	63
									51 - 100	9	13	18	25	36	50	71
									101 - 200	10	14	20	29	40	57	80
Radial composite deviation Total contact	10 - 40	7.5	11	15	21	30	42	60	7 - 12	9	12	17	24	34	48	68
	41 - 100	9.5	13	19	26	37	52	74	13 - 24	9	13	18	26	37	52	73
	101 - 250	12	16	23	33	46	66	93	25 - 50	10	14	20	28	40	56	79
									51 - 100	11	15	22	31	44	62	87
									101 - 200	12	17	24	34	48	68	96
Tooth-to-tooth radial composite deviation	All range	1	2	2.5	3.5	5	7	10	All range	4	6	8	11	16	22	32

Table 2. The Allowable values of each deviation for module 0.75

Unit: μm

Deviations	System of accuracy for JIS B 1702-1 and 2: 1998								System of accuracy for JIS B 1702 and JGMA 116-01							
	No. of teeth	N4	N5	N6	N7	N8	N9	N10	No. of teeth	0	1	2	3	4	5	6
Single pitch deviation	7 - 26	3.3	4.7	6.5	9.5	13	19	26	8 - 16	3	4	5	8	11	15	21
	27 - 66	3.5	5	7	10	14	20	28	17 - 33	3	4	6	8	12	17	24
	67 - 166	3.8	5.5	7.5	11	15	21	30	34 - 66	3	5	7	9	13	19	26
									67 - 133	4	5	7	10	15	21	30
Total cumulative pitch deviations	7 - 26	8	11	16	23	32	45	64	8 - 16	11	15	21	30	43	60	86
	27 - 66	10	14	20	29	41	57	81	17 - 33	12	17	24	33	47	66	94
	67 - 166	13	18	26	37	52	74	104	34 - 66	13	19	26	37	53	74	105
									67 - 133	15	21	30	42	60	83	120
Total profile deviation	7 - 26	3.3	4.6	6.5	9	13	18	26	All range	3	4	6	8	11	16	22
	27 - 66	3.5	5	7.5	10	15	21	29								
	67 - 166	3.8	6	8.5	12	17	23	33								
Runout	7 - 26	6.5	9	13	18	25	36	51	8 - 16	8	11	15	21	30	43	60
	27 - 66	8	11	16	23	32	46	65	17 - 33	8	12	17	24	33	47	66
	67 - 166	10	15	21	29	42	59	83	34 - 66	9	13	19	26	37	53	74
									67 - 133	10	15	21	30	42	60	83
Radial composite deviation Total contact	7 - 26	8	12	16	23	33	46	66	8 - 16	10	14	20	28	39	55	78
	27 - 66	10	14	20	28	40	56	80	17 - 33	11	15	21	30	42	60	84
	67 - 166	12	17	25	35	49	70	98	34 - 66	12	16	23	33	46	65	92
									67 - 133	13	18	25	36	51	72	100
Tooth-to-tooth radial composite deviation	7 - 66	2	2.5	4	5.5	7.5	11	15	All range	4	6	9	13	18	25	36
	67 - 166	2	3	4	5.5	8	11	16								

Table 3. The Allowable value of each deviation for module 0.8

Unit: μm

Deviations	System of accuracy for JIS B 1702-1 and 2: 1998								System of accuracy for JIS B 1702 and JGMA 116-01							
	No. of teeth	N4	N5	N6	N7	N8	N9	N10	No. of teeth	0	1	2	3	4	5	6
Single pitch deviations	7 - 25	3.3	4.7	6.5	9.5	13	19	26	8 - 15	3	4	5	8	11	15	21
	26 - 62	3.5	5	7	10	14	20	28	16 - 31	3	4	6	8	12	17	24
	63 - 156	3.8	5.5	7.5	11	15	21	30	32 - 62	3	5	7	9	13	19	26
									63 - 125	4	5	7	10	15	21	30
Total cumulative pitch deviations	7 - 25	8	11	16	23	32	45	64	8 - 15	11	15	21	30	43	60	86
	26 - 62	10	14	20	29	41	57	81	16 - 31	12	17	24	33	47	66	94
	63 - 156	13	18	26	37	52	74	104	32 - 62	13	19	26	37	53	74	105
									63 - 125	15	21	30	42	60	83	120
Total profile deviation	- 20	3.2	4.6	6.5	9	13	18	26	All range	3	4	6	8	11	16	22
	21 - 50	3.6	5	7.5	10	15	21	29								
	51 - 125	4.1	6	8.5	12	17	23	33								
Runout	7 - 25	6.5	9	13	18	25	36	51	8 - 15	8	11	15	21	30	43	60
	26 - 62	8	11	16	23	32	46	65	16 - 31	8	12	17	24	33	47	66
	63 - 156	10	15	21	29	42	59	83	32 - 62	9	13	19	26	37	53	74
									63 - 125	10	15	21	30	42	60	83
Radial composite deviation Total contact	7 - 25	8	12	16	23	33	46	66	8 - 15	10	14	20	28	39	55	78
	26 - 62	10	14	20	28	40	56	80	16 - 31	11	15	21	30	42	60	84
	63 - 156	12	17	25	35	49	70	98	32 - 62	12	16	23	33	46	65	92
									63 - 125	13	18	25	36	51	72	100
Tooth-to-tooth radial composite deviation	7 - 62	2	2.5	4	5.5	7.5	11	15	All range	4	6	9	13	18	25	36
	63 - 156	2	3	4	5.5	8	11	16								

Table 4. The Allowable value of each deviation for module 1.0

Unit: μm

Deviations	System of accuracy for JIS B 1702-1 and 2: 1998								System of accuracy for JIS B 1702 and JGMA 116-01							
	No. of teeth	N4	N5	N6	N7	N8	N9	N10	No. of teeth	0	1	2	3	4	5	6
Single pitch deviations	- 20	3.3	4.7	6.5	9.5	13	19	26	7 - 12	3	4	5	8	11	15	21
	21 - 50	3.5	5	7	10	14	20	28	13 - 25	3	4	6	8	12	17	24
	51 - 125	3.8	5.5	7.5	11	15	21	30	26 - 50	3	5	7	9	13	19	26
									51 - 100	4	5	7	10	15	21	30
									101 - 200	4	6	9	12	17	24	34
Total cumulative pitch deviations	- 20	8	11	16	23	32	45	64	7 - 12	11	15	21	30	43	60	86
	21 - 50	10	14	20	29	41	57	81	13 - 25	12	17	24	33	47	66	44
	51 - 125	13	18	26	37	52	74	104	26 - 50	13	19	26	37	53	74	105
									51 - 100	15	21	30	42	60	83	120
									101 - 200	17	24	34	48	68	95	135
Total profile deviation	- 20	3.2	4.6	6.5	9	13	18	26	All range	3	4	6	8	11	16	22
	21 - 50	3.6	5	7.5	10	15	21	29								
	51 - 125	4.1	6	8.5	12	17	23	33								
Runout	- 20	6.5	9	13	18	25	36	51	7 - 12	8	11	15	21	30	43	60
	21 - 50	8	11	16	23	32	46	65	13 - 25	8	12	17	24	33	47	66
	51 - 125	10	15	21	29	42	59	83	26 - 50	9	13	19	26	37	53	74
									51 - 100	10	15	21	30	42	60	83
									101 - 200	12	17	24	34	48	68	95
Radial composite deviation Total contact	- 20	9	12	18	25	35	50	70	7 - 12	10	14	20	28	39	55	78
	21 - 50	11	15	21	30	42	60	85	13 - 25	11	15	21	30	42	60	84
	51 - 125	13	18	26	36	52	73	103	26 - 50	12	16	23	33	46	65	92
									51 - 100	13	18	25	36	51	72	100
									101 - 200	14	20	28	40	57	81	115
Tooth-to-tooth radial composite deviation	All range	2.5	3.5	5	7	10	14	20	All range	4	6	9	13	18	25	36

Table 5. The Allowable value of each deviation for module 1.25

Unit: μm

Deviations	System of accuracy for JIS B 1702-1 and 2: 1998								System of accuracy for JIS B 1702 and JGMA 116-01							
	No. of teeth	N4	N5	N6	N7	N8	N9	N10	No. of teeth	0	1	2	3	4	5	6
Single pitch deviations	- 16	3.3	4.7	6.5	9.5	13	19	26	- 9	3	4	6	8	11	16	23
	17 - 40	3.5	5	7	10	14	20	28	10 - 20	3	4	6	9	12	18	25
	41 - 100	3.8	5.5	7.5	11	15	21	30	21 - 40	3	5	7	10	14	19	28
	101 - 224	4.2	6	8.5	12	17	24	34	41 - 80	4	6	8	11	16	22	31
									81 - 160	4	6	9	12	18	25	35
Total cumulative pitch deviations	- 16	8	11	16	23	32	45	64	- 9	11	16	23	32	45	64	91
	17 - 40	10	14	20	29	41	57	81	10 - 20	12	18	25	35	50	70	100
	41 - 100	13	18	26	37	52	74	104	21 - 40	14	19	28	39	55	77	110
	101 - 224	17	24	35	49	69	98	138	41 - 80	16	22	31	44	62	87	125
									81 - 160	18	25	35	50	71	99	140
Total profile deviation	- 16	3.2	4.6	6.5	9	13	18	26	All range	3	4	6	9	13	18	25
	17 - 40	3.6	5	7.5	10	15	21	29								
	41 - 100	4.1	6	8.5	12	17	23	33								
	101 - 224	4.9	7	10	14	20	28	39								
Runout	- 16	6.5	9	13	18	25	36	51	- 9	8	11	16	23	32	45	64
	17 - 40	8	11	16	23	32	46	65	10 - 20	9	12	18	25	35	50	70
	41 - 100	10	15	21	29	42	59	83	21 - 40	10	14	19	28	39	55	77
	101 - 224	14	20	28	39	55	78	110	41 - 80	11	16	22	31	44	62	87
									81 - 160	12	18	25	35	50	71	99
Radial composite deviation Total contact	- 16	10	14	19	27	38	54	76	- 9	10	15	21	30	42	59	84
	17 - 40	11	16	23	32	45	64	91	10 - 20	11	16	23	32	45	64	90
	41 - 100	14	19	27	39	55	77	109	21 - 40	12	17	25	35	49	69	98
	101 - 224	17	24	34	48	68	97	137	41 - 80	13	19	27	38	54	76	105
									81 - 160	15	21	30	42	60	85	120
Tooth-to-tooth radial composite deviation	- 40	3.0	4.5	6.5	9.0	13	18	25	All range	5	7	10	14	20	28	40
	41 - 224	3.0	4.5	6.5	9.0	13	18	26								

Table 6. The Allowable value of each deviation for module 1.5

Unit: μm

Deviations	System of accuracy for JIS B 1702-1 and 2: 1998								System of accuracy for JIS B 1702 and JGMA 116-01							
	No. of teeth	N4	N5	N6	N7	N8	N9	N10	No. of teeth	0	1	2	3	4	5	6
Single pitch deviations	- 13	3.3	4.7	6.5	9.5	13	19	26	- 8	3	4	6	8	11	16	23
	14 - 33	3.5	5	7	10	14	20	28	9 - 16	3	4	6	9	12	18	25
	34 - 83	3.8	5.5	7.5	11	15	21	30	17 - 33	3	5	7	10	14	19	28
	84 - 186	4.2	6	8.5	12	17	24	34	34 - 66	4	6	8	11	16	22	31
									67 - 133	4	6	9	12	18	25	35
Total cumulative pitch deviations	- 13	8	11	16	23	32	45	64	- 8	11	16	23	32	45	64	91
	14 - 33	10	14	20	29	41	57	81	9 - 16	12	18	25	35	50	70	100
	34 - 83	13	18	26	37	52	74	104	17 - 33	14	19	28	39	55	77	110
	84 - 186	17	24	35	49	69	98	138	34 - 66	16	22	31	44	62	87	125
									67 - 133	18	25	35	50	71	99	140
Total profile deviation	- 13	3.2	4.6	6.5	9	13	18	26	All range	3	4	6	9	13	18	25
	14 - 33	3.6	5	7.5	10	15	21	29								
	34 - 83	4.1	6	8.5	12	17	23	33								
	84 - 186	4.9	7	10	14	20	28	39								
Runout	- 13	6.5	9	13	18	25	36	51	- 8	8	11	16	23	32	45	64
	14 - 33	8	11	16	23	32	46	65	9 - 16	9	12	18	25	35	50	70
	34 - 83	10	15	21	29	42	59	83	17 - 33	10	14	19	28	39	55	77
	84 - 186	14	20	28	39	55	78	110	34 - 66	11	16	22	31	44	62	87
									67 - 133	12	18	25	35	50	71	99
Radial composite deviation Total contact	- 13	10	14	19	27	38	54	76	- 8	10	15	21	30	42	59	84
	14 - 33	11	16	23	32	45	64	91	9 - 16	11	16	23	32	45	64	90
	34 - 83	14	19	27	39	55	77	109	17 - 33	12	17	25	35	49	69	98
	84 - 186	17	24	34	48	68	97	137	34 - 66	13	19	27	38	54	76	105
									67 - 133	15	21	30	42	60	85	120
Tooth-to-tooth radial composite deviation	- 33	3	4.5	6.5	9	13	18	25	All range	5	7	10	14	20	28	40
	34 - 186	3	4.5	6.5	9	13	18	26								

Table 7. The Allowable value of each deviation for module 2.0

Unit: μm

Deviations	System of accuracy for JIS B 1702-1 and 2: 1998								System of accuracy for JIS B 1702 and JGMA 116-01							
	No. of teeth	N4	N5	N6	N7	N8	N9	N10	No. of teeth	0	1	2	3	4	5	6
Single pitch deviations	- 10	3.3	4.7	6.5	9.5	13	19	26	7 - 12	3	5	7	9	13	19	27
	11 - 25	3.5	5	7	10	14	20	28	13 - 25	4	5	7	10	15	21	30
	26 - 62	3.8	5.5	7.5	11	15	21	30	26 - 50	4	6	8	12	16	23	33
	63 - 140	4.2	6	8.5	12	17	24	34	51 - 100	5	7	9	13	19	26	37
Total cumulative pitch deviations	- 10	8	11	16	23	32	45	64	7 - 12	13	19	27	38	53	75	105
	11 - 25	10	14	20	29	41	57	81	13 - 25	15	21	30	42	59	83	120
	26 - 62	13	18	26	37	52	74	104	26 - 50	16	23	33	46	66	92	130
	63 - 140	17	24	35	49	69	98	138	51 - 100	19	26	37	52	74	105	150
Total profile deviation	- 10	3.2	4.6	6.5	9	13	18	26	All range	4	5	7	10	15	21	29
	11 - 25	3.6	5	7.5	10	15	21	29								
	26 - 62	4.1	6	8.5	12	17	23	33								
	63 - 140	4.9	7	10	14	20	28	39								
Runout	- 10	6.5	9	13	18	25	36	51	7 - 2	9	13	19	27	38	53	75
	11 - 25	8	11	16	23	32	46	65	13 - 25	10	15	21	30	42	59	83
	26 - 62	10	15	21	29	42	59	83	26 - 50	12	16	23	33	46	66	92
	63 - 140	14	20	28	39	55	78	110	51 - 100	13	19	26	37	52	74	105
Radial composite deviation Total contact	- 10	11	16	22	32	45	63	89	7 - 12	12	17	25	35	49	70	98
	11 - 25	13	18	26	37	52	73	103	13 - 25	13	19	27	38	53	75	105
	26 - 62	15	22	31	43	61	86	122	26 - 50	15	21	29	41	58	82	115
	63 - 140	19	26	37	53	75	106	149	51 - 100	16	23	32	45	64	91	130
Tooth-to-tooth radial composite deviation	- 62	4.5	6.5	9.5	13	19	26	37	All range	6	8	12	16	23	33	47
	63 - 140	4.5	6.5	9.5	13	19	27	38								

Table 8. The Allowable value of each deviation for module 2.5

Unit: μm

Deviations	System of accuracy for JIS B 1702-1 and 2: 1998								System of accuracy for JIS B 1702 and JGMA 116-01							
	No. of teeth	N4	N5	N6	N7	N8	N9	N10	No. of teeth	0	1	2	3	4	5	6
Single pitch deviations	- 8	3.7	5	7.5	10	15	21	29	- 10	3	5	7	9	13	19	27
	9 - 20	3.9	5.5	7.5	11	15	22	31	11 - 20	4	5	7	10	15	21	30
	21 - 50	4.1	6	8.5	12	17	23	33	21 - 40	4	6	8	12	16	23	33
	51 ～ 112	4.6	6.5	9	13	18	26	36	41 ～ 80	5	7	9	13	19	26	37
Total cumulative pitch deviations	～ 8	8.5	12	17	23	33	47	66	～ 10	13	19	27	38	53	75	105
	9 ～ 20	10	15	21	30	42	59	84	11 ～ 20	15	21	30	42	59	83	120
	21 ～ 50	13	19	27	38	53	76	107	21 ～ 40	16	23	33	46	66	92	130
	51 ～ 112	18	25	35	50	70	100	141	41 ～ 80	19	26	37	52	74	105	150
Total profile deviation	～ 8	4.7	6.5	9.5	13	19	26	37	All range	4	5	7	10	15	21	29
	9 ～ 20	5	7	10	14	20	29	40								
	21 ～ 50	5.5	8	11	16	22	31	44								
	51 ～ 112	6.5	9	13	18	25	36	50								
Runout	～ 8	6.5	9.5	13	19	27	38	53	～ 10	9	13	19	27	38	53	75
	9 ～ 20	8.5	12	17	24	34	47	67	11 ～ 20	10	15	21	30	42	59	83
	21 ～ 50	11	15	21	30	43	61	86	21 ～ 40	12	16	23	33	46	66	92
	51 ～ 112	14	20	28	40	56	80	113	41 ～ 80	13	19	26	37	52	74	105
Radial composite deviation Total contact	～ 8								～ 10	12	17	25	35	49	70	98
	9 ～ 20	13	18	26	37	52	73	103	11 ～ 20	13	19	27	38	53	75	105
	21 ～ 50	15	22	31	43	61	86	122	21 ～ 40	15	21	29	41	58	82	115
	51 ～ 112	19	26	37	53	75	106	149	41 ～ 80	16	23	32	45	64	91	130
Tooth-to-tooth radial composite deviation	～ 50	4.5	6.5	9.5	13	19	26	37	All range	6	8	12	16	23	33	47
	51 ～ 112	4.5	6.5	9.5	13	19	27	38								

Table 9. The Allowable value of each deviation for module 3.0

Unit: μm

Deviations	System of accuracy for JIS B 1702-1 and 2: 1998								System of accuracy for JIS B 1702 and JGMA 116-01							
	No. of teeth	N4	N5	N6	N7	N8	N9	N10	No. of teeth	0	1	2	3	4	5	6
Single pitch deviations	7 - 16	3.9	5.5	7.5	11	15	22	31	8	This is beyond the standard, applied Allowable value of No. of teeth of 9.						
	17 - 41	4.1	6	8.5	12	17	23	33	9 - 16	4	6	8	11	16	23	33
	42 - 93	4.6	6.5	9	13	18	26	36	17 - 33	4	6	9	13	18	25	36
									34 - 66	5	7	10	14	20	28	40
Total cumulative pitch deviations	7 - 16	10	15	21	30	42	59	84	8	This is beyond the standard, applied Allowable value of No. of teeth of 9.						
	17 - 41	13	19	27	38	53	76	107	9 - 16	16	23	33	46	65	91	130
	42 - 93	18	25	35	50	70	100	141	17 - 33	18	25	36	51	72	100	145
									34 - 66	20	28	40	57	81	115	160
Total profile deviation	7 - 16	5	7	10	14	20	29	40	All range	4	6	9	13	18	25	36
	17 - 41	5.5	8	11	16	22	31	44								
	42 - 93	6.5	9	13	18	25	36	50								
Runout	7 - 16	8.5	12	17	24	34	47	67	8	This is beyond the standard, applied Allowable value of No. of teeth of 9.						
	17 - 41	11	15	21	30	43	61	86	9 - 16	11	16	23	33	46	65	91
	42 - 93	14	20	28	40	56	80	113	17 - 33	13	18	25	36	51	72	100
									34 - 66	14	20	28	40	57	81	115
Radial composite deviation Total contact	7 - 16	16	22	31	44	63	89	126	8	This is beyond the standard, applied Allowable value of No. of teeth of 9.						
	17 - 41	18	25	36	51	72	102	144	9 - 16	15	21	30	43	60	85	120
	42 - 93	21	30	43	61	86	121	172	17 - 33	16	23	32	46	65	92	130
									34 - 66	18	25	35	50	71	100	140
Tooth-to-tooth radial composite deviation	- 93	7.5	10	15	21	29	41	58	All range	7	10	13	20	29	40	57

Table 10. The Allowable value of each deviation for module 4.0

Unit: μm

Deviations	System of accuracy for JIS B 1702-1 and 2: 1998								System of accuracy for JIS B 1702 and JGMA 116-01							
	No. of teeth	N4	N5	N6	N7	N8	N9	N10	No. of teeth	0	1	2	3	4	5	6
Single pitch deviations	- 12	4.3	6	8.5	12	17	24	34	- 124		6	8	11	16	23	33
	13 - 31	4.6	6.5	9	13	18	26	36	13 - 25	4	6	9	13	18	25	36
	32 - 70	5	7	10	14	20	28	40	26 - 50	5	7	10	14	20	28	40
Total cumulative pitch deviations	- 12	11	15	22	31	44	62	87	- 12	16	23	33	46	65	91	130
	13 - 31	14	19	28	39	55	78	110	13 - 25	18	25	36	51	72	100	145
	32 - 70	18	25	36	51	72	102	144	26 - 50	20	28	40	57	81	115	160
Total profile deviation	- 12	6	9	12	18	25	35	50	All range	4	6	9	13	18	25	36
	13 - 31	6.5	9.5	13	19	27	38	54								
	32 - 70	7.5	11	15	21	30	42	60								
Runout	- 12	8.5	12	17	25	35	49	70	- 12	11	16	23	33	46	65	91
	13 - 31	11	16	22	31	44	62	88	13 - 25	13	18	25	36	51	72	100
	32 - 70	14	20	29	41	58	82	115	26 - 50	14	20	28	40	57	81	115
Radial composite deviation Total contact	- 12	16	22	31	44	63	89	126	- 12	15	21	30	43	60	85	120
	13 - 31	18	25	36	51	72	102	144	13 - 25	16	23	32	46	65	92	130
	32 - 70	21	30	43	61	86	121	172	26 - 50	18	25	35	50	71	100	140
Tooth-to-tooth radial composite deviation	- 70	7.5	10	15	21	29	41	58	All range	7	10	13	20	29	40	57

Table 11. The Allowable value of each deviation for module 5.0

Unit: μm

Deviations	System of accuracy for JIS B 1702-1 and 2: 1998								System of accuracy for JIS B 1702 and JGMA 116-01							
	No. of teeth	N4	N5	N6	N7	N8	N9	N10	No. of teeth	0	1	2	3	4	5	6
Single pitch deviations	- 10	4.3	6	8.5	12	17	24	34	- 10	5	7	9	13	19	26	37
	11 - 25	4.6	6.5	9	13	18	26	36	11 - 20	5	7	10	14	20	28	40
	26 - 56	5	7	10	14	20	28	40	21 - 40	6	8	11	16	22	32	45
Total cumulative pitch deviations	- 10	11	15	22	31	44	62	87	- 10	19	26	37	52	74	105	150
	11 - 25	14	19	28	39	55	78	110	11 - 20	20	28	40	57	81	115	160
	26 - 56	18	25	36	51	72	102	144	21 - 40	22	32	45	63	90	125	180
Total profile deviation	- 10	6	9	12	18	25	35	50	All range	6	8	11	16	23	32	45
	11 - 25	6.5	9.5	13	19	27	38	54								
	26 - 56	7.5	11	15	21	30	42	60								
Runout	- 10	8.5	12	17	25	35	49	70	- 10	13	19	26	37	52	74	105
	11 - 25	11	16	22	31	44	62	88	11 - 20	14	20	28	40	57	81	115
	26 - 56	14	20	29	41	58	82	115	21 - 40	15	22	32	45	63	90	125
Radial composite deviation Total contact	- 10	20	28	39	56	79	111	157	- 10	18	25	35	50	70	100	140
	11 - 25	22	31	44	62	88	124	176	11 - 20	19	27	38	53	75	105	150
	26 - 56	25	36	51	72	102	144	203	21 - 40	20	29	41	58	81	115	160
Tooth-to-tooth radial composite deviation	- 56	11	15	22	31	44	62	87	All range	9	13	18	26	36	51	73

Table 12. Total helix deviation

Reference diameter d mm	Facewidth b mm	Reference diameter						
		N4	N5	N6	N7	N8	N9	N10
		μm						
5 ≦ d ≦ 20	4 ≦ b ≦ 10	4.3	6	8.5	12	17	24	35
	10 < b ≦ 20	4.9	7	9.5	14	19	28	39
	20 < b ≦ 40	5.5	8	11	16	22	31	45
20 < d ≦ 50	4 ≦ b ≦ 10	4.5	6.5	9	13	18	25	36
	10 < b ≦ 20	5	7	10	14	20	29	40
	20 < b ≦ 40	5.5	8	11	16	23	32	46
50 < d ≦ 125	4 ≦ b ≦ 10	4.7	6.5	9.5	13	19	27	38
	10 < b ≦ 20	5.5	7.5	11	15	21	30	42
	20 < b ≦ 40	6	8.5	12	17	24	34	48
	40 < b ≦ 80	7	10	14	20	28	39	56
125 < d ≦ 280	4 ≦ b ≦ 10	5	7	10	14	20	29	40
	10 < b ≦ 20	5.5	8	11	16	22	32	45
	20 < b ≦ 40	6.5	9	13	18	25	36	50
	40 < b ≦ 80	7.5	10	15	21	29	41	58
280 < d ≦ 560	10 < b ≦ 20	6	8.5	12	17	24	34	48
	20 < b ≦ 40	6.5	9.5	13	19	27	38	54
	40 < b ≦ 80	7.5	11	15	22	31	44	62
	80 < b ≦ 160	9	13	18	26	36	52	73

Table 13. Allowable value of Runout for material of Outside diameter (JIS B 1702 old)

Unit: μm

da = Outside diameter (mm)	1.5 < da ≦ 3.0	3 < da ≦ 6	6 < da ≦ 12	12 < da ≦ 25	25 < da ≦ 50	50 < da ≦ 100	100 < da ≦ 200	200 < da ≦ 400	400 < da ≦ 800	800 < da ≦ 1,600	1,600 < da ≦ 3,200
Class 0	3	4	4	4	5	5	6	6	7	9	10
Class 1	5	5	5	6	6	7	8	9	10	12	14
Class 2	7	7	8	8	9	10	11	13	15	17	20
Class 3	10	10	11	12	13	14	16	18	20	24	28
Class 4	14	14	15	17	18	20	22	25	29	34	40
Class 5	19	20	22	23	26	28	31	36	41	47	56
Class 6	28	29	31	33	36	40	45	51	58	60	80
Class 7	55	58	62	67	73	80	90	100	115	135	160
Class 8	110	115	125	135	145	160	180	200	230	270	320

Table 14. Allowable value of Runout for material of side flank (JIS B 1702 old) for class 0 gear.

Unit: μm

d = Reference diameter (mm)		1.5 < da ≦ 3.0	3 < da ≦ 6	6 < da ≦ 12	12 < da ≦ 25	25 < da ≦ 50	50 < da ≦ 100	100 < da ≦ 200	200 < da ≦ 400	400 < da ≦ 800	800 < da ≦ 1,600	1,600 < da ≦ 3,200
b= Facewidth (mm)	b < 3.0	2	2	3	4	5	8	-	-	-	-	-
	3 < b ≦ 6	2	2	3	3	5	8	13	-	-	-	-
	6 < b ≦ 12	2	2	3	3	5	7	12	23	-	-	-
	12 < b ≦ 25	2	2	3	3	4	7	11	20	38	-	-
	25 < b ≦ 50	-	2	3	3	4	6	9	16	30	59	-
	50 < b ≦ 100	-	-	2	3	3	5	7	12	22	42	82
	100 < b ≦ 200	-	-	-	3	3	4	5	8	15	27	52
	200 < b ≦ 400	-	-	-	-	3	3	4	6	9	17	31
	400 < b ≦ 800	-	-	-	-	-	3	3	4	6	10	18

Table 15. Allowable value of Runout for material of side flank (JIS B 1702 old) for class 1 gear.

Unit: μm

d = Reference diameter (mm)		1.5 < da ≦ 3.0	3 < da ≦ 6	6 < da ≦ 12	12 < da ≦ 25	25 < da ≦ 50	50 < da ≦ 100	100 < da ≦ 200	200 < da ≦ 400	400 < da ≦ 800	800 < da ≦ 1,600	1,600 < da ≦ 3,200
b= Facewidth (mm)	b < 3.0	3	3	4	5	7	11	-	-	-	-	-
	3 < b ≦ 6	3	3	4	5	7	11	19	-	-	-	-
	6 < b ≦ 12	3	3	4	5	7	10	18	32	-	-	-
	12 < b ≦ 25	3	3	4	5	6	9	16	28	53	-	-
	25 < b ≦ 50	-	3	4	4	5	8	13	23	43	83	-
	50 < b ≦ 100	-	-	3	4	5	7	10	17	31	59	115
	100 < b ≦ 200	-	-	-	4	4	5	7	12	21	38	74
	200 < b ≦ 400	-	-	-	-	4	4	6	8	13	23	44
	400 < b ≦ 800	-	-	-	-	-	4	4	6	9	14	25

Table 16. Allowable value of Runout for material of side flank (JIS B 1702 old) for class 2 gear.

Unit: μm

d = Reference diameter (mm)		1.5 < da ≦ 3.0	3 < da ≦ 6	6 < da ≦ 12	12 < da ≦ 25	25 < da ≦ 50	50 < da ≦ 100	100 < da ≦ 200	200 < da ≦ 400	400 < da ≦ 800	800 < da ≦ 1,600	1,600 < da ≦ 3,200
b= Facewidth (mm)	b < 3.0	5	5	6	7	10	16	-	-	-	-	-
	3 < b ≦ 6	5	5	6	7	10	15	26	-	-	-	-
	6 < b ≦ 12	5	5	5	7	9	14	25	45	-	-	-
	12 < b ≦ 25	4	5	5	6	9	13	22	40	75	-	-
	25 < b ≦ 50	-	5	5	6	8	11	18	32	60	115	-
	50 < b ≦ 100	-	-	5	5	7	9	14	24	44	83	160
	100 < b ≦ 200	-	-	-	5	6	7	10	17	29	54	105
	200 < b ≦ 400	-	-	-	-	5	6	8	11	18	33	61
	400 < b ≦ 800	-	-	-	-	-	5	6	8	12	20	35

Table 17. Allowable value of Runout for material of side flank (JIS B 1702 old) for class 3 gear.

Unit: μm

d = Reference diameter (mm)		1.5 < da ≦ 3.0	3 < da ≦ 6	6 < da ≦ 12	12 < da ≦ 25	25 < da ≦ 50	50 < da ≦ 100	100 < da ≦ 200	200 < da ≦ 400	400 < da ≦ 800	800 < da ≦ 1,600	1,600 < da ≦ 3,200
b= Facewidth (mm)	b < 3.0	6	7	8	10	14	22	-	-	-	-	-
	3 < b ≦ 6	6	7	8	10	14	22	37	-	-	-	-
	6 < b ≦ 12	6	7	8	10	13	21	35	64	-	-	-
	12 < b ≦ 25	6	7	8	9	12	19	31	56	105	-	-
	25 < b ≦ 50	-	7	7	8	11	16	26	46	86	165	-
	50 < b ≦ 100	-	-	7	8	10	13	20	34	62	120	230
	100 < b ≦ 200	-	-	-	7	8	10	15	24	41	77	150
	200 < b ≦ 400	-	-	-	-	7	9	11	16	26	47	88
	400 < b ≦ 800	-	-	-	-	-	7	9	12	17	28	50

Table 18. Allowable value of Runout for material of side flank (JIS B 1702 old) for class 4 gear.

Unit: μm

d = Reference diameter (mm)		1.5 < da ≦ 3.0	3 < da ≦ 6	6 < da ≦ 12	12 < da ≦ 25	25 < da ≦ 50	50 < da ≦ 100	100 < da ≦ 200	200 < da ≦ 400	400 < da ≦ 800	800 < da ≦ 1,600	1,600 < da ≦ 3,200
b= Facewidth (mm)	b < 3.0	9	10	11	14	20	31	-	-	-	-	-
	3 < b ≦ 6	9	10	11	14	19	30	52	-	-	-	-
	6 < b ≦ 12	9	10	11	13	19	29	49	90	-	-	-
	12 < b ≦ 25	9	9	11	13	17	26	44	79	150	-	-
	25 < b ≦ 50	-	9	10	12	15	22	36	64	120	230	-
	50 < b ≦ 100	-	-	10	11	13	18	28	48	87	165	320
	100 < b ≦ 200	-	-	-	10	12	15	21	33	58	110	210
	200 < b ≦ 400	-	-	-	-	10	12	16	23	37	66	125
	400 < b ≦ 800	-	-	-	-	-	10	12	16	24	39	70

Table 19. Allowable value of Runout for material of side flank (JIS B 1702 old) for class 5 gear.

Unit: μm

d = Reference diameter (mm)		1.5 < da ≦ 3.0	3 < da ≦ 6	6 < da ≦ 12	12 < da ≦ 25	25 < da ≦ 50	50 < da ≦ 100	100 < da ≦ 200	200 < da ≦ 400	400 < da ≦ 800	800 < da ≦ 1,600	1,600 < da ≦ 3,200
b= Facewidth (mm)	b < 3.0	13	14	16	20	28	45	-	-	-	-	-
	3 < b ≦ 6	13	14	16	20	28	43	75	-	-	-	-
	6 < b ≦ 12	13	14	15	19	27	41	70	130	-	-	-
	12 < b ≦ 25	13	14	15	18	25	37	62	115	210	-	-
	25 < b ≦ 50	-	13	14	17	22	32	52	92	170	330	-
	50 < b ≦ 100	-	-	14	15	19	26	40	68	125	240	469
	100 < b ≦ 200	-	-	-	14	16	21	30	47	83	155	300
	200 < b ≦ 400	-	-	-	-	15	17	22	32	53	94	175
	400 < b ≦ 800	-	-	-	-	-	15	18	23	34	56	190

Accuracy for Bevel gear
JIS B 1704 (Extracts)

1. Applicable Range covers accuracy of Bevel gear with Outer transverse module 0.4 to 25.0 and Outer pitch diameter 3.0 mm to 1,600.00 mm

Remark: Above applicable range can be used for Hypoid gear.

2. The meanings of gear terms. Standard terms are used as follow.

(1) Single pitch deviation.
Amount of actual pitch on Pitch circle at Mean cone distance of adjacent teeth subtracted by its correct pitch.

(2) Pitch variation deviation.
The Absolute amount of difference between adjacent two pitches on Pitch circle at Mean cone distance.

(3) Total cumulative pitch deviations.
The value from amount of correct pitch subtracted by sum of actual pitch with any adjacent two pitches at Mean cone distance.

(4) Runout.
Maximum difference at location of radius direction when contact piece such as Over balls or Rollers are put to Tooth space near Pitch circle.

3. System of accuracy for gears is classified into 9 classes. Can select to combine from different classes with different deviation or choose only necessary items in accordance to the usage purpose. There are the classes 0, 1, 2, 3, 4, 5, 6, 7, 8.

4. Allowable value For classification of System of accuracy, refer to following pages for Allowable values of Single pitch deviation, Pitch variation deviation, Total cumulative pitch deviation and Runout.

Table 20. Allowable tolerances for Transverse module 0.4 to 0.6.

Unit: μm

System of accuracy	Deviations	d = Pitch diameter (mm)					
		3.0 < d ≦ 6.0	6.0 < d ≦ 12.0	12.0 < d ≦ 25.0	25.0 < d ≦ 50.0	50.0 < d ≦ 100.0	100.0 < d ≦ 200.0
0	Single pitch deviation (±)	3	4	4	4	4	5
	Pitch variation deviation	4	5	5	5	6	6
	Total cumulative pitch deviations (±)	14	14	15	16	18	19
	Runout	5	7	10	14	20	28
1	Single pitch deviation (±)	6	6	7	7	8	8
	Pitch variation deviation	8	8	9	9	10	11
	Total cumulative pitch deviations (±)	25	26	27	29	31	34
	Runout	7	10	15	21	30	43
2	Single pitch deviation (±)	11	12	12	13	14	15
	Pitch variation deviation	15	15	16	17	18	20
	Total cumulative pitch deviations (±)	46	47	50	52	56	60
	Runout	11	15	22	31	45	63
3	Runout	16	24	33	48	67	95
4	Runout	25	35	50	71	100	145
5	Runout	37	52	75	105	150	210
6	Runout	56	79	110	160	230	320

Table 21. Allowable tolerances for Transverse module above 0.6 to 1.0

Unit: μm

System of accuracy	Deviations	d = Pitch diameter (mm)					
		3.0 < d ≦ 6.0	6.0 < d ≦ 12.0	12.0 < d ≦ 25.0	25.0 < d ≦ 50.0	50.0 < d ≦ 100.0	100.0 < d ≦ 200.0
0	Single pitch deviation (±)	4	4	4	4	5	5
	Pitch variation deviation	5	5	5	5	6	6
	Total cumulative pitch deviations (±)	14	15	16	17	18	20
	Runout	5	7	10	14	20	28
1	Single pitch deviation (±)	6	7	7	7	8	9
	Pitch variation deviation	8	9	9	10	10	11
	Total cumulative pitch deviations (±)	25	26	28	30	32	34
	Runout	7	10	15	21	30	43
2	Single pitch deviation (±)	12	12	13	13	14	15
	Pitch variation deviation	15	16	16	17	18	20
	Total cumulative pitch deviations (±)	46	48	50	53	57	61
	Runout	11	15	22	31	45	63
3	Runout	16	24	33	48	67	95
4	Runout	25	35	50	71	100	145
5	Runout	37	52	75	105	150	210
6	Runout	56	79	110	160	230	320

Table 22. Allowable tolerances for Transverse module above 1.0 to 1.6.

Unit: μm

System of accuracy	Deviations	d = Pitch diameter (mm)					
		3.0 < d ≦ 6.0	6.0 < d ≦ 12.0	12.0 < d ≦ 25.0	25.0 < d ≦ 50.0	50.0 < d ≦ 100.0	100.0 < d ≦ 200.0
0	Single pitch deviation (±)	4	4	4	5	5	6
	Pitch variation deviation	5	5	6	6	7	7
	Total cumulative pitch deviations (±)	15	16	17	19	20	22
	Runout	7	10	14	20	28	40
1	Single pitch deviation (±)	7	7	8	8	9	10
	Pitch variation deviation	9	9	10	11	11	13
	Total cumulative pitch deviations (±)	27	29	30	32	35	39
	Runout	10	15	21	30	43	60
2	Single pitch deviation (±)	12	13	14	14	16	17
	Pitch variation deviation	16	17	18	19	20	22
	Total cumulative pitch deviations (±)	49	52	54	58	62	68
	Runout	15	22	31	45	63	89
3	Single pitch deviation (±)	23	23	25	26	28	30
	Pitch variation deviation	29	30	32	34	36	39
	Total cumulative pitch deviations (±)	90	94	98	105	110	120
	Runout	24	33	48	67	95	135
4	Single pitch deviation (±)	41	42	44	46	49	52
	Pitch variation deviation	53	55	57	60	63	68
	Total cumulative pitch deviations (±)	165	170	175	185	195	210
	Runout	35	50	71	100	145	200
5	Runout	52	75	105	150	210	300
6	Runout	79	110	160	230	320	450

Table 23. Allowable tolerances for Transverse module above 1.6 to 2.5.

Unit: μm

System of accuracy	Deviations	d = Pitch diameter (mm)					
		3.0 < d ≦ 6.0	6.0 < d ≦ 12.0	12.0 < d ≦ 25.0	25.0 < d ≦ 50.0	50.0 < d ≦ 100.0	100.0 < d ≦ 200.0
0	Single pitch deviation (±)	4	4	5	5	6	6
	Pitch variation deviation	5	6	6	7	8	9
	Total cumulative pitch deviations (±)	17	18	19	21	23	26
	Runout	10	14	20	28	40	56
1	Single pitch deviation (±)	7	8	8	9	10	11
	Pitch variation deviation	10	10	11	12	13	14
	Total cumulative pitch deviations (±)	30	32	34	36	40	44
	Runout	15	21	30	43	60	86
2	Single pitch deviation (±)	13	14	15	16	17	19
	Pitch variation deviation	17	18	19	21	23	25
	Total cumulative pitch deviations (±)	54	56	60	64	69	76
	Runout	22	31	45	63	89	125
3	Single pitch deviation (±)	24	25	27	28	31	33
	Pitch variation deviation	31	33	35	37	40	43
	Total cumulative pitch deviations (±)	97	100	105	115	120	135
	Runout	33	48	67	95	135	190
4	Single pitch deviation (±)	43	45	47	50	55	57
	Pitch variation deviation	56	58	61	65	69	75
	Total cumulative pitch deviations (±)	170	180	190	200	210	239
	Runout	50	71	100	145	200	290
5	Pitch variation deviation	110	115	120	125	132	150
	Runout	75	105	150	210	300	430
6	Pitch variation deviation	210	220	240	250	270	290
	Runout	110	160	230	320	450	640

Table 24. Allowable tolerances for Transverse module above 2.5 to 4.0.

Unit: μm

System of accuracy	Deviations	d = Pitch diameter (mm)						
		12.0 < d ≦ 25.0	25.0 < d ≦ 50.0	50.0 < d ≦ 100.0	100.0 < d ≦ 200.0	200.0 < d ≦ 400.0	400.0 < d ≦ 800.0	800.0 < d ≦ 1,600.0
0	Single pitch deviation (±)	5	5	5	6	6	7	8
	Pitch variation deviation	6	6	7	7	8	9	10
	Total cumulative pitch deviations (±)	18	19	21	22	24	27	31
	Runout	10	14	20	28	40	56	79
1	Single pitch deviation (±)	8	8	9	10	10	12	13
	Pitch variation deviation	10	11	12	12	14	15	17
	Total cumulative pitch deviations (±)	32	33	36	38	42	46	51
	Runout	15	21	30	43	60	86	120
2	Single pitch deviation (±)	14	15	16	17	18	20	22
	Pitch variation deviation	18	19	20	22	24	26	29
	Total cumulative pitch deviations (±)	57	59	63	67	72	79	88
	Runout	22	31	45	63	89	125	180
3	Single pitch deviation (±)	25	27	28	30	32	35	38
	Pitch variation deviation	33	34	36	39	41	45	49
	Total cumulative pitch deviations (±)	100	105	110	120	130	140	150
	Runout	33	48	67	95	135	190	270
4	Single pitch deviation (±)	45	47	50	52	55	59	65
	Pitch variation deviation	59	61	65	67	72	77	84
	Total cumulative pitch deviations (±)	180	185	200	210	220	240	260
	Runout	50	71	100	145	200	290	400
5	Pitch variation deviation	115	120	125	130	135	155	170
	Runout	75	105	150	210	300	430	600
6	Pitch variation deviation	220	240	250	260	280	290	310
	Runout	110	160	230	320	450	640	900
7	Runout	250	360	500	720	1000	1450	2000

Table 25. Allowable tolerances for Transverse module above 4.0 to 6.0

Unit: μm

System of accuracy	Deviations	d = Pitch diameter (mm)					
		25.0 < d ≦ 50.0	50.0 < d ≦ 100.0	100.0 < d ≦ 200.0	200.0 < d ≦ 400.0	400.0 < d ≦ 800.0	800.0 < d ≦ 1,600.0
0	Single pitch deviation (±)	5	6	6	7	7	8
	Pitch variation deviation	7	7	8	9	9	11
	Total cumulative pitch deviations (±)	21	22	24	26	29	32
	Runout	14	20	28	40	56	79
1	Single pitch deviation (±)	9	10	10	11	12	14
	Pitch variation deviation	12	12	13	14	16	18
	Total cumulative pitch deviations (±)	36	38	41	45	49	54
	Runout	21	30	43	60	86	120
2	Single pitch deviation (±)	16	17	18	19	21	23
	Pitch variation deviation	21	22	23	25	27	30
	Total cumulative pitch deviations (±)	64	67	72	77	84	92
	Runout	31	45	63	89	125	180
3	Single pitch deviation (±)	28	30	31	34	36	40
	Pitch variation deviation	37	39	41	44	47	52
	Total cumulative pitch deviations (±)	115	120	125	135	145	160
	Runout	48	67	95	135	190	270
4	Single pitch deviation (±)	50	52	54	58	62	68
	Pitch variation deviation	65	67	71	75	81	88
	Total cumulative pitch deviations (±)	200	210	220	230	250	270
	Runout	71	100	145	200	290	400
5	Pitch variation deviation	125	130	135	150	165	175
	Runout	105	150	210	300	430	600
6	Pitch variation deviation	250	260	270	290	300	330
	Runout	160	230	320	450	640	900
7	Runout	360	500	720	1000	1450	2000

Table 26. Allowable tolerances for Transverse module above 6.0 to 10.0

Unit: μm

System of accuracy	Deviations	d = Pitch diameter (mm)					
		25.0 < d ≦ 50.0	50.0 < d ≦ 100.0	100.0 < d ≦ 200.0	200.0 < d ≦ 400.0	400.0 < d ≦ 800.0	800.0 < d ≦ 1,600.0
0	Single pitch deviation (±)	6	6	7	7	8	9
	Pitch variation deviation	8	8	9	9	10	11
	Total cumulative pitch deviations (±)	24	25	27	29	32	35
	Runout	14	20	28	40	56	79
1	Single pitch deviation (±)	10	11	11	12	13	15
	Pitch variation deviation	13	14	15	16	17	19
	Total cumulative pitch deviations (±)	41	43	46	49	54	59
	Runout	21	30	43	60	86	120
2	Single pitch deviation (±)	18	19	20	21	23	25
	Pitch variation deviation	23	24	26	27	30	32
	Total cumulative pitch deviations (±)	71	75	79	84	91	100
	Runout	31	45	63	89	125	180
3	Single pitch deviation (±)	31	33	34	37	39	43
	Pitch variation deviation	41	42	45	48	51	56
	Total cumulative pitch deviations (±)	125	130	140	145	155	170
	Runout	48	67	95	135	190	270
4	Single pitch deviation (±)	54	56	59	62	67	72
	Pitch variation deviation	71	73	77	81	87	100
	Total cumulative pitch deviations (±)	220	230	240	250	270	290
	Runout	71	100	145	220	290	400
5	Pitch variation deviation	135	140	155	165	175	185
	Runout	105	150	210	300	430	600
6	Pitch variation deviation	270	280	290	310	320	340
	Runout	160	230	320	450	640	900
7	Runout	360	500	720	1000	1450	2000

Table 27. Allowable tolerance for Tip angle of material

Unit: Minutes

System of accuracy	b = Facewidth (mm)			
	b < 1.6	1.6 < b ≦ 6	6.0 < b ≦ 25.0	b > 25.0
1, 2	0 +60	0 +20	0 +10	0 +8
3, 4	0 +100	0 +30	0 +20	0 +15
5, 6	0 +120	0 +40	0 +25	0 +20
7, 8	0 +150	0 +60	0 +30	0 +25

Details for Allowable distance from Outside diameter of material or Crown circle to Reference back cone ⋯ omitted.

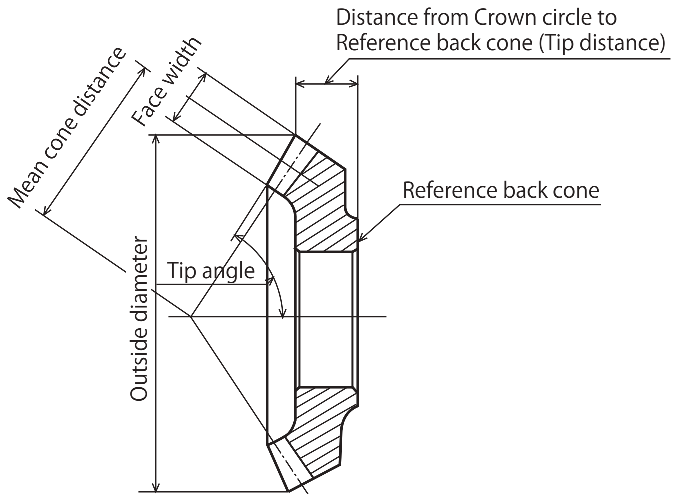


Fig. 19 Terms for Bevel gear

1. Allowable value of Runout for material's Cone surface. When using Reference surface for gear cutting or measurement with material's Tip cone surface. Allowable Runout for material's Tip cone surface is indicated in Table below. While material's Back cone and Front cone surface is used for Reference surface, values below may also be used.

Note (1): Runout for material's cone surface is difference between maximum and minimum readings of an indicator when turning the material with the indicater placed firmly near the heel of cone perpendicular to cone surface.

Table 28. Allowable value of Runout for material's Cone surface

Unit: μm

System of accuracy	d = Pitch diameter (mm)								
	3.0 < d ≦ 6.0	6.0 < d ≦ 12.0	12.0 < d ≦ 25.0	25.0 < d ≦ 50.0	50.0 < d ≦ 100.0	100.0 < d ≦ 200.0	200.0 < d ≦ 400.0	400.0 < d ≦ 800.0	800.0 < d ≦ 1,600.0
1, 2	14	15	17	18	20	22	25	30	34
3, 4	33	35	38	41	45	51	57	66	76
5, 6	73	77	83	91	100	110	125	145	170
7, 8	-	-	185	200	220	250	280	330	380

2. Allowable value of Runout for side flank of material. For the material of Bevel gear with shaft or bore, refer to Table 29, shows Allowable value of Runout for side flank of material when using the Reference surface as flat face perpendicular to axis for gear cutting.

Note (1): Runout for material's side flank is difference between maximum and minimum readings of an indicator when turning the material with the indicater placed firmly near the heel of Reference side face.

Table 29. Allowable value of Runout for material's side flank

Unit: μm

System of accuracy	d = Pitch diameter (mm)								
	3.0 < d ≦ 6.0	6.0 < d ≦ 12.0	12.0 < d ≦ 25.0	25.0 < d ≦ 50.0	50.0 < d ≦ 100.0	100.0 < d ≦ 200.0	200.0 < d ≦ 400.0	400.0 < d ≦ 800.0	800.0 < d ≦ 1,600.0
1, 2	6	6	7	7	8	9	10	12	14
3, 4	16	17	19	20	22	25	28	33	38
5, 6	46	49	53	57	63	71	80	92	105
7, 8	-	-	150	165	180	200	230	260	310

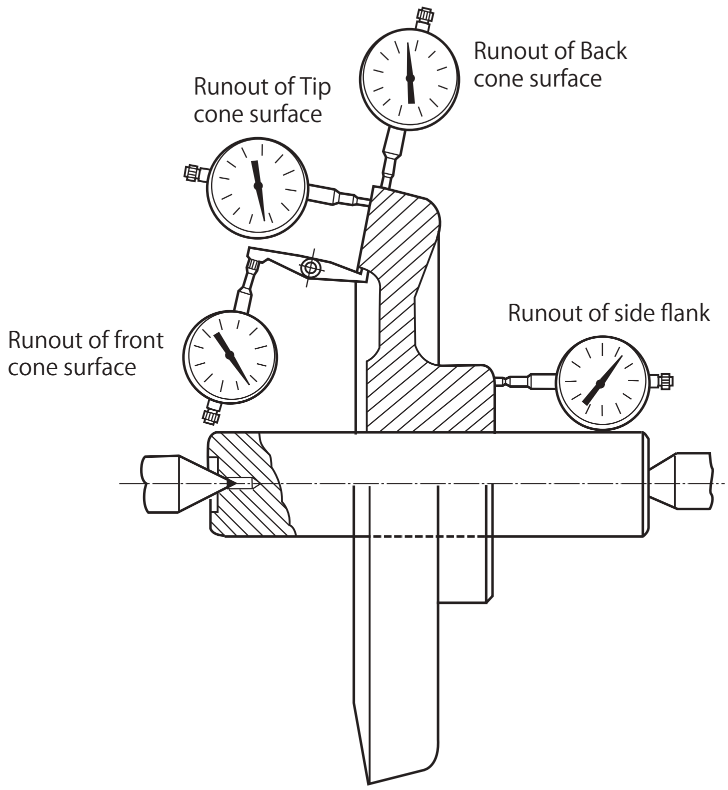


Fig. 20 Runout of Bevel gear with bore

Chapter 6 Gear assembly

6.1 Advice on gear assembly

When assembling the gear pair, please note the following recommended points.

(1) Beware of gear with scratches and rust, handle gear with care.
Small scratch marks may cause noise.

(2) Remove sharp edges near tooth flank.
It is advisable to perform chamfering by semi top process to remove sharp edges on the Tooth tip. If chamfering is not performed, be sure to find and remove scratch marks or burrs on the gear.

(3) Measure the backlash.
Backlash regardless big or small causes noise. It is necessary to maintain proper backlash. If not, it is necessary to adjust centre distance. For details on KG-Backlash, please refer to page 24 of Technical Data and page 23 of front pages.

(4) Confirm tooth bearing.
Noise and oscillation is caused by poor tooth contact. Poor tooth bearing also harms the durability of the gear. Please refer to page 96 of section 6.4 for more on tooth contact.

(5) Use suitable type of lubricating oil in proper amounts.
Refer to pages 99 to 103 of sections 6.5 and 6.6 for suitable type of lubricating oil in proper amounts.

(6) Perform warm up and test run.
We recommend that warm up and test run be performed before actual operations in order to improve hardness and strength of tooth flank.

(When applying Heat treatment to pinion only) Especially for Worm gear pair, warm up and test run is recommended to improve area of tooth bearing and surface strength.
Tooth profile for Worm gear pair has complicated curved surface compared with other gears making it difficult to fabricate Worm gear pair with improved accuracy. There are limitations to surface roughness when processed with lath only.
It is necessary to perform warm up and test run for Worm gear pair. Do not apply full load or close to full load to Worm gear pair or scuffing will occur easily.
For Warm up and Test run, gives improved evenness of tooth flank and increased tooth contact area (per square measure), which reduces the load (per square measure). It will also improve wear resistance against work hardening of tooth flank.
Therefore it provides a longer lifespan for the gear and reduces the noise level and oscillation.
Method of Warm up and Test run. Firstly check the tooth contact while applying empty load and then gradually increase load to the gear.
We recommend changing all the lubricating oil after warm up and test run. Subsequently we recommend that the lubricating oil be changed every 6 months or 25,000 hours which ever comes first.

In addition, take note of dynamic balance and assembling method as recommended.

6.2 Centre distance for Spur and Helical gears

Gear assembly with accurate working centre distance is recommended for Spur and Helical gears. Fig. 1 shows an extract from the Allowable deviations of Centre distance for Spur and Helical gears as defined in **JGMA 1101-1 (2000 Japan Gear Manufacturing Association)**.

Allowable tolerance for Centre distance

(1) Accuracy standard for Spur and Helical gears

Table 1 shows Allowable deviation of Centre distance for classes N3 to N12 gears of **JIS B 1702-1** and **JIS B 1702-2** (covers only ground and hobbing gears)

(2) Centre distance: Shortest distance from centre of axes of Parallel spur gear pair or gear pair with Non-parallel and Non-intersecting axes.

Table 1. Allowable tolerances of Centre distance for the gear

Unit: μm

System of accuracy a = Centre distance (mm)	N3, N4	N5, N6	N7, N8	N9, N10	N11, N12
5.0 < a ≦ 20.0	± 6	± 10	± 16	± 26	± 65
20.0 < a ≦ 50.0	± 8	± 12	± 20	± 31	± 80
50.0 < a ≦ 125.0	± 12	± 20	± 32	± 50	± 125
125.0 < a ≦ 280.0	± 16	± 26	± 40	± 65	± 160
280.0 < a ≦ 560.0	± 22	± 35	± 55	± 88	± 220
560.0 < a ≦ 1,000.0	± 28	± 45	± 70	± 115	± 280
1,000.0 < a ≦ 1,600.0	± 39	± 62	± 98	± 155	± 390
1,600.0 < a ≦ 2,500.0	± 55	± 88	± 140	± 220	± 550
2,500.0 < a ≦ 4,000.0	± 84	± 130	± 205	± 330	± 825

*The above chart uses ± symbol. It is recommended to use positive side tolerances for External gear pair and negative side tolerance for Internal gear pair.

6.3 Parallelism of axes for Spur and Helical gears

Extract from JGMA1102 (2000) is as follows.
0. Preface: This standard stipulates Allowable value of parallel accuracy for Spur and Helical gears.
Basically, these standards are consistant with recommended values from ISO/TR10064-3 (1996).

1. Application range

This standard stipulates the parallel accuracy for steel-made Involute spur and helical gears. Therefore gears covered by this standard are simply called Gear.

(1) Normal module: 0.5 to 70.0 (mm)
(2) Reference diameter: 5.0 to 10,000.0 (mm)
(3) Facewidth: From 4.0 to 1,000.0 (mm)

Remark 1. Double helical gear axis is also covered.
Remark 2. The above mentioned Standard is quoted from:
ISO/TR 10064-3 (1996) Cylindrical gears- Code of inspection practice- Part 3
JIS B 0102 (1999) International gear notation - Symbols for geometrical data
JIS B 1702-1 (1998) Cylindrical gears- ISO System of accuracy Classification-Article 1:
Definition and allowable values of deviations relavent to corresponding flanks of the gear teeth.
ISO/TR 10064-3 (1996) Cylindrical gears- Code of Inspection Practice- Part 3

2. Definition of terms

Definition for this standard is from JIS B 0102 (1999) (Terms of Tooth Flank-Geometric Definition) and following details.

(1) **Parallel accuracy of axis**: Composes of accuracy of parallel deviation and Non-parallel and Non-intersection deviations.

(2) **Parallel deviation of axis**: Distance between C and 0[(1)] (Refer to Fig. 1) where both ends of measurement distance L on the a-axis on one side of the gear are points A and B; and Flat face H is surface to include one of the points A and one of shaft centre b (b-axis); and flat face V is surface through point A and parallel to b and perpendicular to flat face H, and orthogonal projection of point B to H is C.
Note(1): Point 0 is base of tolerance among perpendicular flat face S, V, H and B.

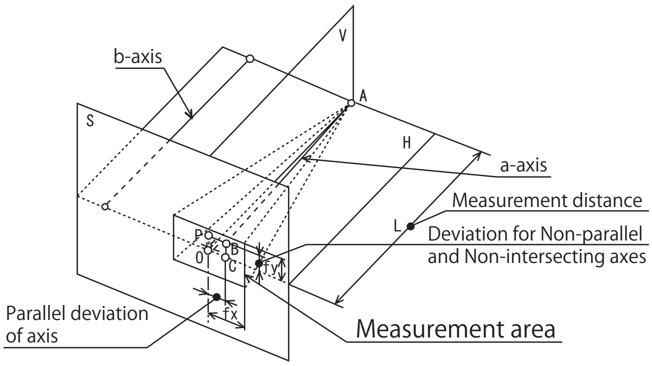


Fig. 1 Deviations for Parallel axis, Non-parallel and Non-intersecting axes.

(3) Refer to Fig.1, **deviation for Non-parallel and Non-intersecting axes**: Distance between points 0 and D where D is orthogonal projection of point B to V, referring to above (2).

3. Allowable value

Allowable value for parallel accuracy of gear axis is met with System of accuracy N10 to N12 in JIS B1702-1 (1998) as follows,

(1) Allowable value of parallel deviation for axis fx
Calculating fx for measuring span L of gear axis is as follows,

$$fx = \frac{L}{b} fx'$$

Hereby, L : Measuring span (mm)
b : Facewidth (mm), choose smaller dimension of Facewidth (mm) between pinion and gear.
fx': Refer to Table 1 (μm)

(2) Allowable value of deviation for Axes of Non-parallel and Non-intersecting fy.

Calculation fy for measuring span L of gear axis is as follows,

$$fy = \frac{L}{b} fy'$$

Hereby, L : Measuring span (mm)

b : Facewidth (mm), choose smaller dimension of Facewidth (mm) between pinion and gear.

fy' : Refer to Table 2 (μm)

Remark

Depending on purpose of usage and System of accuracy class, which is different from the gear, Allowable value of deviation of parallelism accuracy of axis can be used.

Table 2. Allowable values of parallel deviations fx' for axis per Facewidth

Unit: μm

Reference diameter d (mm)	Facewidth b (mm)	System of Accuracy												
		N0	N1	N2	N3	N4	N5	N6	N7	N8	N9	N10	N11	N12
5 ≦ d ≦ 20	4 ≦ b ≦ 10	1.1	1.5	2.2	3.1	4.3	6.0	8.5	12	17	24	35	49	69
	10 < b ≦ 20	1.2	1.7	2.4	3.4	4.9	7.0	9.5	14	19	28	39	55	78
	20 < b ≦ 40	1.4	2.0	2.8	3.9	5.5	8.0	11	16	22	31	45	63	89
20 < d ≦ 50	4 ≦ b ≦ 10	1.1	1.6	2.2	3.2	4.5	6.5	9.0	13	18	25	36	51	72
	10 < b ≦ 20	1.3	1.8	2.5	3.6	5.0	7.0	10	14	20	29	40	57	81
	20 < b ≦ 40	1.4	2.0	2.9	4.1	5.5	8.0	11	16	23	32	46	65	92
50 < d ≦ 125	4 ≦ b ≦ 10	1.2	1.7	2.4	3.3	4.7	6.5	9.5	13	19	27	38	53	76
	10 < b ≦ 20	1.3	1.9	2.6	3.7	5.5	7.5	11	15	21	30	42	60	84
	20 < b ≦ 40	1.5	2.1	3.0	4.2	6.0	8.5	12	17	24	34	48	68	95
	40 < b ≦ 80	1.7	2.5	3.5	4.9	7.0	10	14	20	28	39	56	79	111
125 < d ≦ 280	4 ≦ b ≦ 10	1.3	1.8	2.5	3.6	5.0	7.0	10	14	20	29	40	57	81
	10 < b ≦ 20	1.4	2.0	2.8	4.0	5.5	8.0	11	16	22	32	45	63	90
	20 < b ≦ 40	1.6	2.2	3.2	4.5	6.5	9.0	13	18	25	36	50	71	101
	40 < b ≦ 80	1.8	2.6	3.6	5.0	7.5	10	15	21	29	41	58	82	117
280 < d ≦ 560	10 < b ≦ 20	1.5	2.1	3.0	4.3	6.0	8.5	12	17	24	34	48	68	97
	20 < b ≦ 40	1.7	2.4	3.4	4.8	6.5	9.5	13	19	27	38	54	76	108
	40 < b ≦ 80	1.9	2.7	3.9	5.5	7.5	11	15	22	31	44	62	87	124
	80 < b ≦ 160	2.3	3.2	4.6	6.5	9.0	13	18	26	36	52	73	103	146

Table 3. Allowable values of Non-parallel and Non-intersecting deviations fy' for axis per Facewidth

Unit: μm

Reference diameter d (mm)	Facewidth b (mm)	System of Accuracy												
		N0	N1	N2	N3	N4	N5	N6	N7	N8	N9	N10	N11	N12
5 ≦ d ≦ 20	4 ≦ b ≦ 10	0.5	0.8	1.1	1.5	2.2	3.1	4.3	6.0	8.5	12	17	24	35
	10 < b ≦ 20	0.6	0.9	1.2	1.7	2.4	3.4	4.9	7.0	9.5	14	19	28	39
	20 < b ≦ 40	0.7	1.0	1.4	2.0	2.8	3.9	5.5	8.0	11	16	22	31	45
20 < d ≦ 50	4 ≦ b ≦ 10	0.6	0.8	1.1	1.6	2.2	3.2	4.5	6.5	9.0	13	18	25	36
	10 < b ≦ 20	0.6	0.9	1.3	1.8	2.5	3.6	5.0	7.0	10	14	20	29	40
	20 < b ≦ 40	0.7	1.0	1.4	2.0	2.9	4.1	5.5	8.0	11	16	23	32	46
50 < d ≦ r125	4 ≦ b ≦ 10	0.6	0.8	1.2	1.7	2.4	3.3	4.7	6.5	9.5	13	19	27	38
	10 < b ≦ 20	0.7	0.9	1.3	1.9	2.6	3.7	5.5	7.5	11	15	21	30	42
	20 < b ≦ 40	0.7	1.1	1.5	2.1	3.0	4.2	6.0	8.5	12	17	24	34	48
	40 < b ≦ 80	0.9	1.2	1.7	2.5	3.5	4.9	7.0	10	14	20	28	39	56
125 < d ≦ 280	4 ≦ b ≦ 10	0.6	0.9	1.3	1.8	2.5	3.5	5.0	7.0	10	14	20	29	40
	10 < b ≦ 20	0.7	1.0	1.4	2.0	2.8	4.0	5.5	8.0	11	16	22	32	45
	20 < b ≦ 40	0.8	1.1	1.6	2.2	3.2	4.5	6.5	9.0	13	18	25	36	50
	40 < b ≦ 80	0.9	1.3	1.8	2.6	3.6	5.0	7.5	10	15	21	29	41	58
280 < d ≦ 560	10 < b ≦ 20	0.8	1.1	1.5	2.1	3.0	4.3	6.0	8.5	12	17	24	34	48
	20 < b ≦ 40	0.8	1.2	1.7	2.4	3.4	4.8	6.5	9.5	13	19	27	38	54
	40 < b ≦ 80	1.0	1.4	1.9	2.7	3.9	5.5	7.5	11	15	22	31	44	62

6.4 Tooth bearings

Regardless of how accurate the gear itself may be, poor tooth bearing not only causes oscillation and noise but also have bad effect on gear's life span.
Refer to Fig. 2. Extracted **Tooth bearing on gear from JIS B 1741-1977 (old)**
JIS B1741 (old)「Tooth bearing on Gear」stipulates percentage of tooth bearing mark as follows.
As for Tooth trace direction, it is percentage (%) of mean value bc of Length of tooth bearing for Effective length of trace - b'. As for Tooth depth direction, it is percentage (%) of mean value lc of tooth bearing width for Working depth- h'.
Note∗ For edge of gear tooth with chamfering, Effective length of trace is after deducting chamfered area. For different Effective lengths of Tooth trace between Pinion and Gear, take the shorter side.

Tooth bearing percentage of Tooth trace direction $= \frac{b_c}{b'} \times 100(\%)$ Tooth bearing percentage for Tooth depth $= \frac{l_c}{h'} \times 100(\%)$

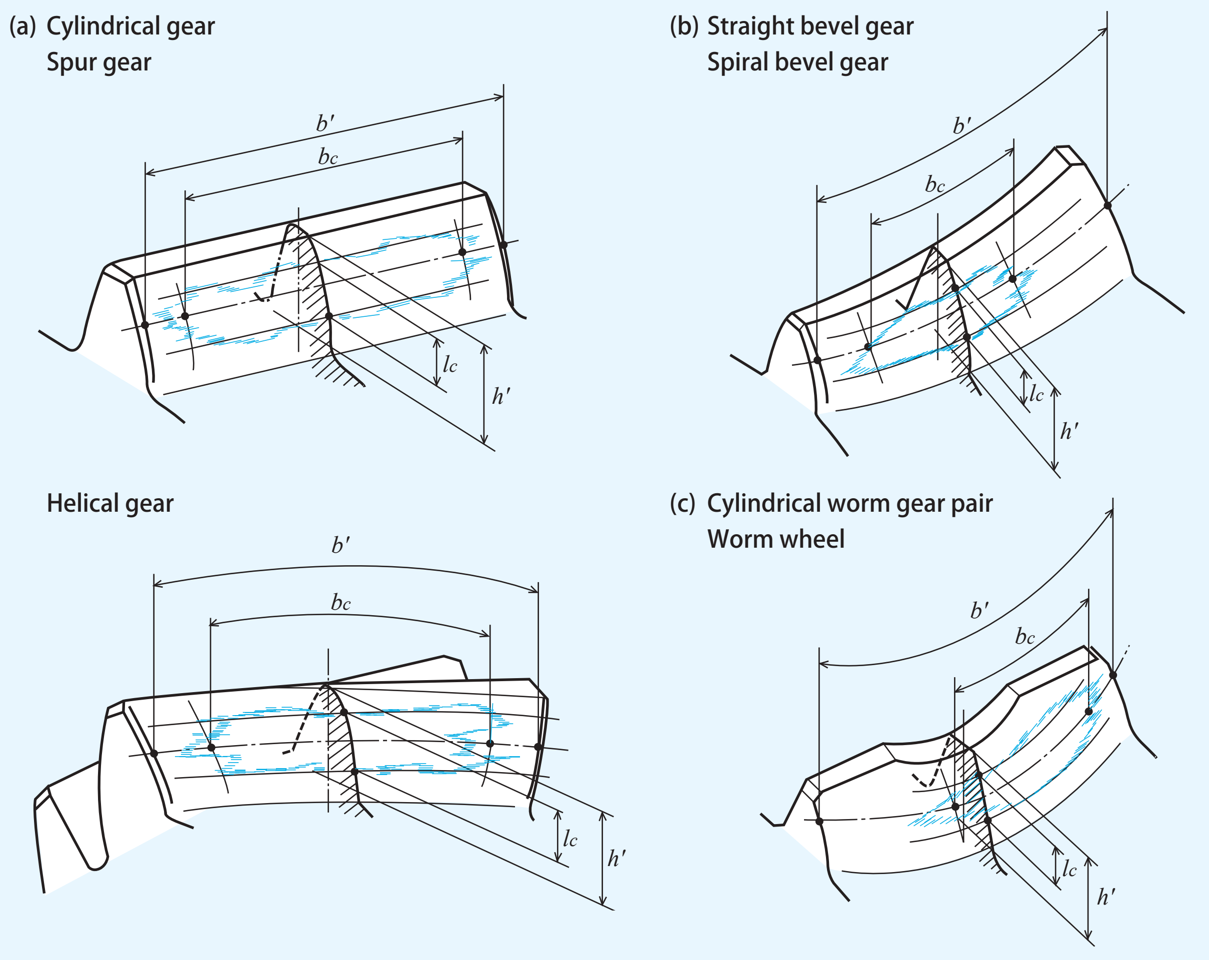


Fig. 2 Tooth bearing

Refer to Fig. 3 for Bevel gear with Crowning and empty load. It is desireable that centre of tooth bearing in Tooth trace direction is about 60% of Length of tooth trace from heel.

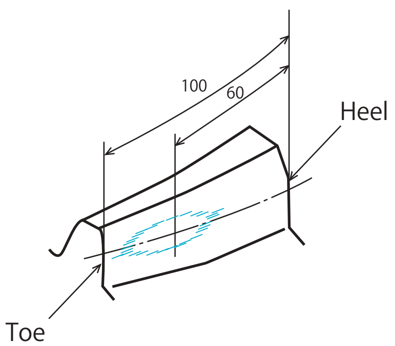


Fig. 3 Tooth bearing for Bevel gear with Crowning.

Percentage of tooth bearing for Worm gear pair is for Worm wheel engaged with Worm gear.
In general, Tooth bearing to inflow side of flank of Worm wheel is not desirable. It is desirable for Tooth bearing centre in Tooth trace direction to be biased towards outflow side to make clearance at inflow side. (Refer to Fig. 4)

Fig. 4 Tooth bearing for Worm wheel

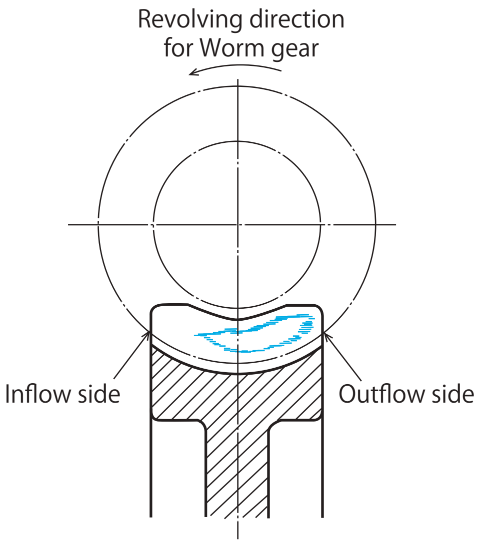


Fig. 5 Inflow clearance for Worm gear pair
{A few problems of lubricating oil for Worm gear pair and research work for machine. Volume 8, No. 4 (1956) written by Dr. Waguri and Dr. Ueno from Yokendo Co. Ltd.}

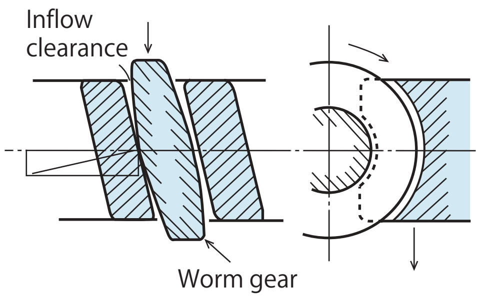


Fig. 6 Line of contact for Worm gear pair (2 number of threads) and Tooth bearing for standard Worm gear. Quoted literature is the same as Fig. 5.

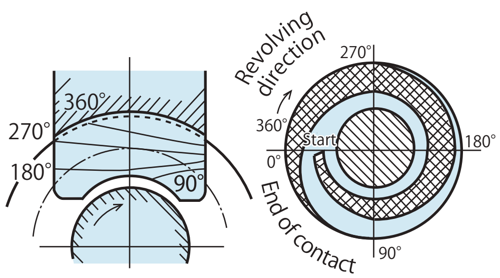
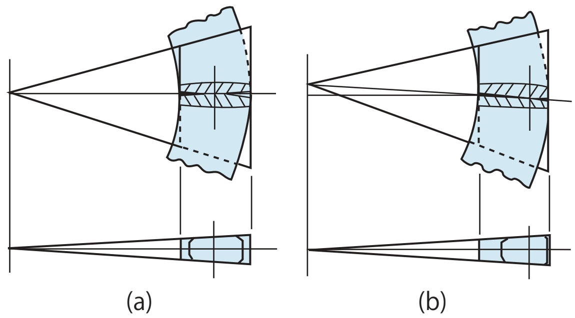


Fig. 7 Engagement for Bevel gear with Crowning
{Gleason Company, INSTALLATION OF BEVEL GEARS (1965)}

Coniflex® Bevel Gear
(Straight bevel gear with Crowning)
Ⓡ mark is Gleason Works trademark



Spiral bevel gear

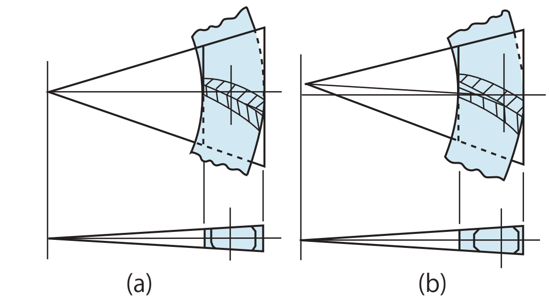
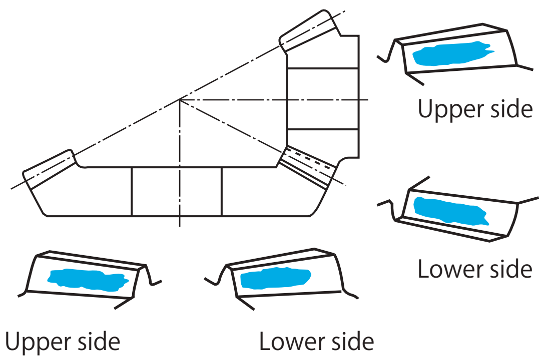


Fig. (a) shows proper assembly method, (b) is assembled off centre from location of Top cone. Please observe the difference in position for Tooth bearing.

Fig.8 Ideal tooth bearing for Bevel gear



Spiral bevel gear
(Pinion: Shape of teeth is left hand)

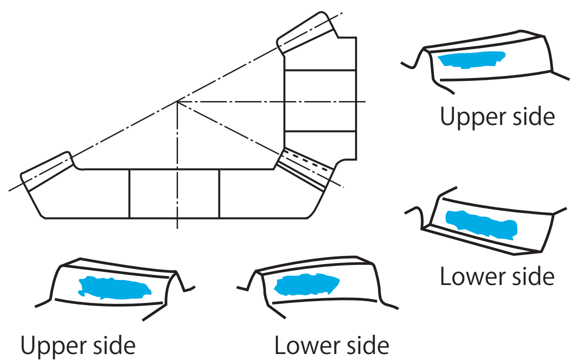


Table 4. Percentage of tooth bearing for Cylindrical gear (Spur and Helical gears)

Class	Percentage of tooth bearing	
	Tooth trace direction	Tooth depth direction
A	Above 70% of Effective length of Tooth trace	Above 40% of Effective length of Tooth profile
B	Above 50% of Effective length of Tooth trace	Above 30% of Effective length of Tooth profile
C	Above 35% of Effective length of Tooth trace	Above 20% of Effective length of Tooth profile

Table 5. Percentage of tooth bearing for Worm gear pair (Worm wheel)

Class	Percentage of tooth bearing	
	Tooth trace direction	Tooth depth direction
A	Above 50% of Effective length of Tooth trace	Above 40% of Effective length of Tooth profile
B	Above 35% of Effective length of Tooth trace	Above 30% of Effective length of Tooth profile
C	Above 20% of Effective length of Tooth trace	Above 20% of Effective length of Tooth profile

Table 6. Percentage of tooth bearing for Bevel gear

Class	Percentage of tooth bearing	
	Tooth trace direction	Tooth depth direction
A	Above 50% of effective length of Tooth trace	Above 40% of Effective length of Tooth profile
B	Above 35% of Effective length of Tooth trace	Above 30% of Effective length of Tooth profile
C	Above 25% of Effective length of Tooth trace	Above 20% of Effective length of Tooth profile

Table 7. Table for Tooth bearing classification and System of accuracy

Class	System of accuracy for Cylindrical gear	System of accuracy class for Bevel gear
	JIS B 1702-1960 (old)	JIS B 1704-1973
A	1, 2	1, 2
B	3, 4	3, 4
C	5, 6	5, 6

6.5 Lubricating oil for Gears

Purpose of using lubricating oil for longer life of gear is as follows,

1) Avoid metal contact (without oil film) to flank.

2) Reduce frictional heat from flank

In addition, better efficiency with less oscillation and noise can be expected.

Insufficient lubricating oil to flank can cause high oscillation and noise in a short time. Scuffing will occur with the increasing temperature, resulting in damage to the bearing. To prevent such problems, apply suitable lubricating oil to the gear is necessary. Proceed with proper method and amount to gear.

Method of lubricating oil

Classifications of lubricating oil to gears are as follows,

1) Grease lubricating method

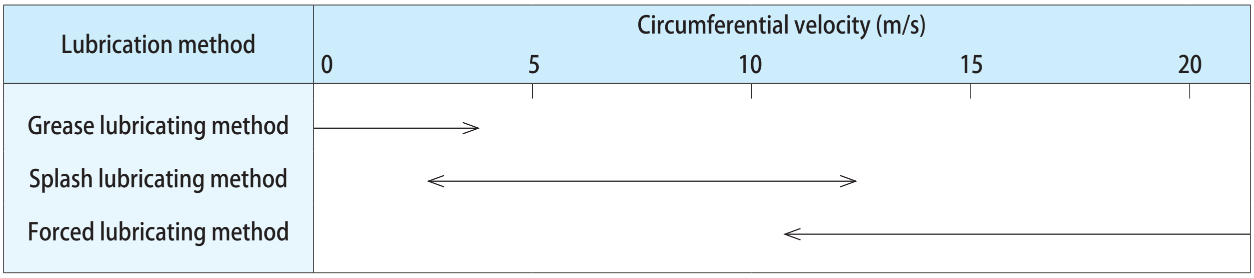
2) Splash lubricating method (Oil bath or Splash lubrication)

3) Forced lubricating method

Selection of Method of lubricating oil can be by types of gears, Circumferential velocity, surface pressure (load applied to gear), finishing condition of flank, hardness of material and combination of materials. However, Circumferential velocity is usually used.

Table 8 indicates guide for selecting gear's lubricating method by circumferential velocity.

(1) For Spur, Helical and Bevel gears



(2) For Worm gear pair and Hypoid gears

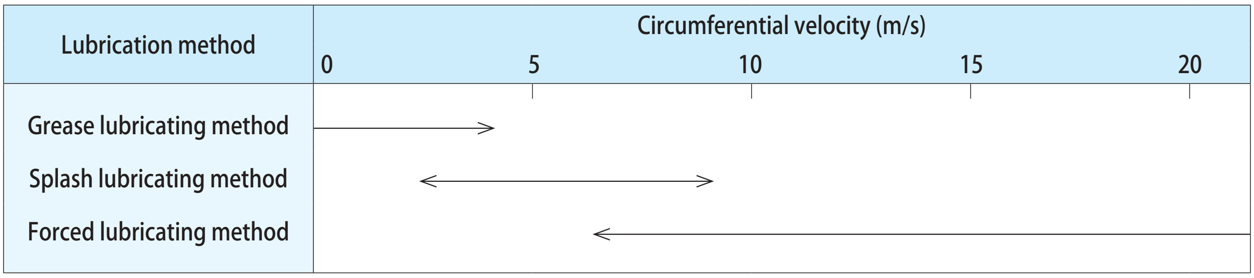


Table 8. Guide for selecting gear lubricating method by circumferential velocity.

Proper level of lubricating oil

(1) Splash lubricating method (Oil bath or Splash lubricating)

Amount of lubricating oil for soaking each type of gear is different. The mixer resistance and windage are increased when large amount of lubricating oil are used for soaking the gear. Table 9 shows the proper level of lubricating oil for soaking the gear.

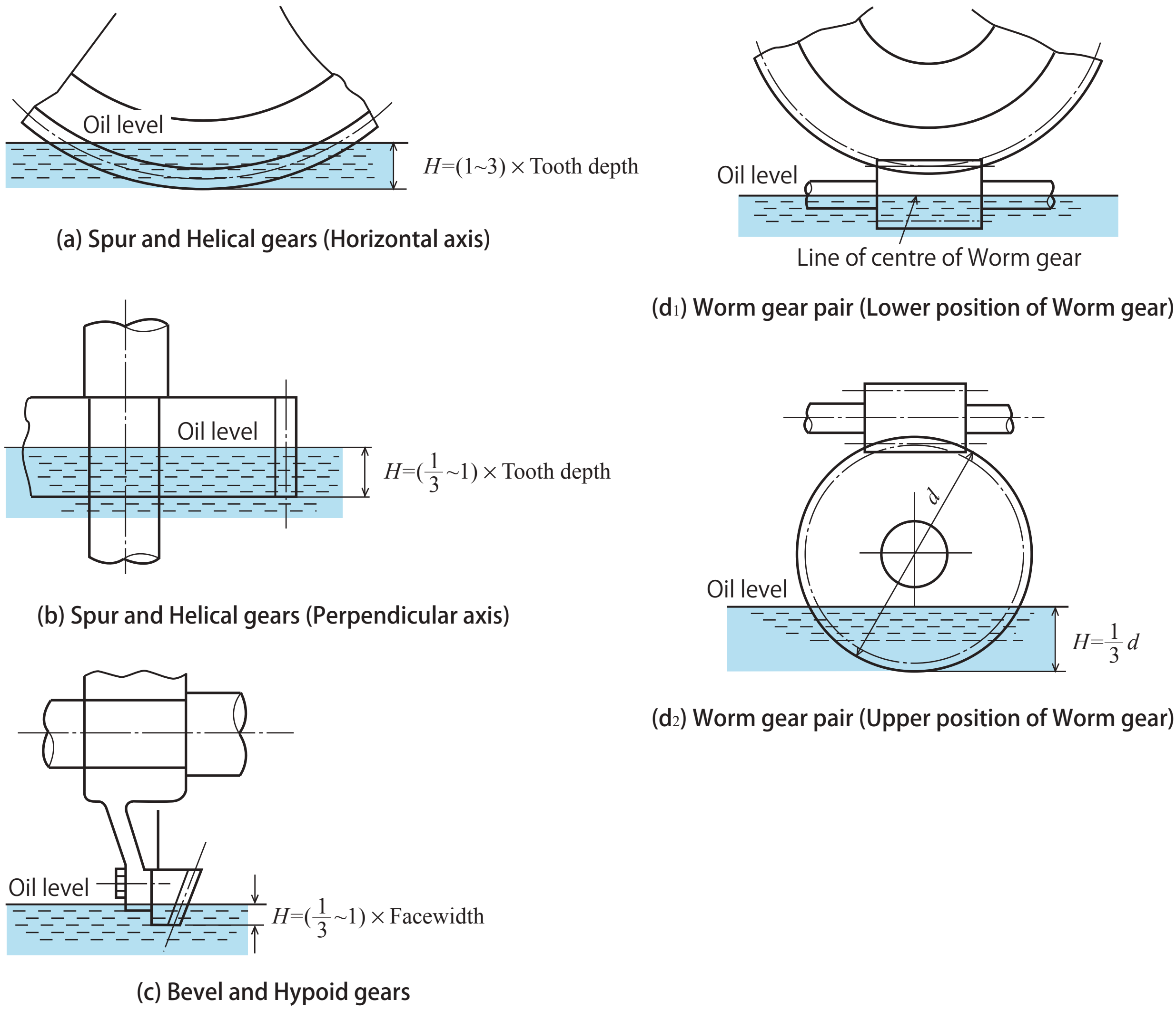


Fig. 9 Soaking level of gear in gearbox

(2) Forced lubricating method

In general, temperature of lubricating oil should not exceed 8°C when lubricating oil flows onto working area of gear. Criterion for facewidth per cm is 0.5*l*/min for low speed and 1*l*/min for high speed. Lubricating oil for high speed, use following empirical formula.

$$\text{Oil level}(l/\text{min}) = 0.6 + 2\times10^{-3} \cdot mv$$

Hereby

m : Module (mm)

v : Circumferential velocity (m/s) of Pitch circle

Spray before the starting area of gear engagement with lubricating oil perpendicular to flank. In rare instances for high speed, spray in the direction towards the end of the engagement.

To prevent temperature of oil from increasing, the collected oil should go through a cooling process using cooling equipment before being reused.

6.6 Lubricating oil

Requisite for lubricating oil are:
1) Coefficient of viscosity, 2) Wear resistance, 3) Coolness and 4) Stability. Selection of proper lubricating oil from grades and usage of the gears are recommended.

(1) The Coefficient of viscosity of lubricating oil for gear
Table 9 shows Coefficient of viscosity and grades of industrial gear oil (JIS K 2219).

Table 9. Grades of Industrial gear oil and Coefficient of viscosity

Grades			Kinematic viscosity cSt {mm²/s} (40°C)	Usage
For industrial use	Grade 1	ISO VG 32	28.8 < cSt ≦ 35.2	Used for sealed gearbox for general industries with light load.
		ISO VG 46	41.4 < cSt ≦ 50.6	
		ISO VG 68	61.2 < cSt ≦ 74.8	
		ISO VG100	90.0 < cSt ≦ 110	
		ISO VG150	135 < cSt ≦ 165	
		ISO VG220	198 < cSt ≦ 242	
		ISO VG320	288 < cSt ≦ 352	
		ISO VG460	414 < cSt ≦ 506	
	Grade 2	ISO VG 68	61.2 < cSt ≦ 74.8	Used for sealed gearbox for general and rolling mill industries with medium or heavy load.
		ISO VG100	90.0 < cSt ≦ 110	
		ISO VG150	135 < cSt ≦ 165	
		ISO VG220	198 < cSt ≦ 242	
		ISO VG320	288 < cSt ≦ 352	
		ISO VG460	414 < cSt ≦ 506	
		ISO VG680	612 < cSt ≦ 748	

(2) Coefficient of viscosity for gear lubricating oil from AGMA (American Gear Manufacturer Association)
Refer to Table 10. Shows the Coefficient of viscosity of lubricating oil for the gear from AGMA (American Gear Manufacturer Association).

Table 10. Lubricating gear oil No. and Coefficient of viscosity from AGMA

Lubricating oil number R&O(1) from AGMA	Kinematic viscosity cSt {mm²/s} (40°C)	ISO Coefficient of Viscosity grade	Lubricating oil No. EP(2) from AGMA
1	41.4 < cSt ≦ 50.6	46	
2	61.2 < cSt ≦ 74.8	68	2 EP
3	90 < cSt ≦ 110	100	3 EP
4	135 < cSt ≦ 165	150	4 EP
5	198 < cSt ≦ 242	220	5 EP
6	288 < cSt ≦ 352	320	6 EP
7 Comp(3)	414 < cSt ≦ 506	460	7 EP
8 Comp	612 < cSt ≦ 748	680	8 EP
8A Comp	900 < cSt ≦ 1,100	1000	8A EP

Note (1) R&O is an abbreviation for Rust and Oxidation Inhibited Gear Oils.
(2) EP is an abbreviation for Extreme Pressure Gear Lubricants.
(3) Comp: 3 % - 10 % of oils and fats or synthetic oils and fats are mixed.

(3) Selection of lubricating oil {AGMA 250.04 (1981)}

The General guide for selection of lubricating gear oil from AGMA 250.04 (1981-9) for Sealed gearbox. Table 11 is for Cylindrical and Bevel gears. Table 12 is for Worm gear pair.

Table 11. Recommended lubricating gear oil for sealed gearbox from AGMA (for Cylindrical and Bevel gears).

Types of train and axes condition	Capacity of gearbox		Surrounding temperature °C	
			-10 ～ 10	10 ～ 50
Parallel axis Reduction speed with single pair	Centre distance	Below 200mm	2-3	3-4
		200 mm - 500 mm	2-3	4-5
		Above 500mm	2-3	4-5
Parallel axis Speed reduction with 2 pairs		Below 200mm	2-3	3-4
		Above 200mm	3-4	4-5
Parallel axis Speed reduction with 3 pairs		Below 200mm	2-3	3-4
		200 mm - 500 mm	3-4	4-5
		Above 500mm	4-5	5-6
Planetary gearbox	Outer dimension of gearbox			
	Below 400 mm		2-3	3-4
	Above 400 mm		3-4	4-5
Straight and Spiral bevel gearboxes	Cone distance			
	Below 300 mm		2-3	4-5
	Above 300 mm		3-4	5-6
Geared motor			2-3	4-5
Gearbox for High speed			1	2

Table 12. Recommended lubricating gear oil for sealed gearbox from AGMA (Worm gear pair)

Types of Worm gear pair and Centre distance mm	Revolving velocity of Worm gear bellow (min^{-1})	Surrounding temperature °C -10 ～ 10	Surrounding temperature °C 10 ～ 50	Revolving velocity of Worm gear exceeds (min^{-1})	Surrounding temperature °C -10 ～ 10	Surrounding temperature °C 10 ～ 50
Cylindrical worm gear pair						
Below 150 mm	700	7 Comp, 7EP	8 Comp, 8EP	700	7 Comp, 7EP	8 Comp, 8EP
150mm - 300mm	450	"	"	450	"	7 Comp, 7EP
300mm - 450mm	300	"	"	300	"	"
450mm - 600mm	250	"	"	250	"	"
Above 600 mm	200	"	"	200	"	"
Enveloping worm gear pair						
Below 150 mm	700	8 Comp	8A Comp	700	8 Comp	8 Comp
150mm - 300mm	450	"	"	450	"	"
300mm - 450mm	300	"	"	300	"	"
450mm - 600mm	250	"	"	250	"	"
Above 600 mm	200	"	"	200	"	"

Industrial gear oil (For Extreme pressure type)

ISO viscosity grade ISO VG cst (40°C)	COSMO	NISSEKI (Shin Nippon Oil)	IDEMITSU	MITSUBISHI	JOMO	SHOWA SHELL	ESSO	MOBIL
Below 68	COSMO Gear SE68	BON NOCK AX68 BON NOCK M68	Daphne Super Gear Oil 68 Daphne Super Gear Oil LW 68 Daphne Alpha Gear 68	DIAMOND SUPER GEARLUBE SP 68	REDUCTUS 68 ES GEAR G68	OMALA OIL 68 G-C OIL 68SE	SPARTAN EP68	MOBILGEAR 626
100	COSMO Gear SE100 COSMO Gear MO68	BON NOCK AX100 BON NOCK M100	Daphne Super Gear Oil 100 Daphne Super Gear Oil LW 100 Daphne Alpha Gear 100	DIAMOND SUPER GEARLUBE SP 100	REDUCTUS 100 ES GEAR G100	OMALA OIL 100 G-C OIL 100SE	SPARTAN EP100	MOBILGEAR 627
150	COSMO Gear SE150 COSMO Gear MO150	BON NOCK AX150 BON NOCK M150	Daphne Super Gear Oil 150 Daphne Super Gear Oil LW 150 Daphne Alpha Gear 150	DIAMOND SUPER GEARLUBE SP 150	REDUCTUS 150 ES GEAR G150	OMALA OIL 150 G-C OIL 150SE	SPARTAN EP150	MOBILGEAR 629 MOBIL GLYGOYLE 22
220	COSMO Gear SE220 COSMO Gear MO220	BON NOCK AX220 BON NOCK M220	Daphne Super Gear Oil 220 Daphne Super Gear Oil LW 220 Daphne Alpha Gear 220	DIAMOND SUPER GEARLUBE SP 220	REDUCTUS 220 ES GEAR G220	OMALA OIL 220 G-C OIL 220SE	SPARTAN EP220	MOBILGEAR 630 SHC220 MOBIL GLYGOYLE 30
320	COSMO Gear SE320 COSMO Gear MO320	BON NOCK AX320 BON NOCK M320	Daphne Super Gear Oil 320 Daphne Alpha Gear 220	DIAMOND SUPER GEARLUBE SP 320	REDUCTUS 320 ES GEAR G320	OMALA OIL 320 G-C OIL 320SE	SPARTAN EP320	MOBILGEAR SHC320
460	COSMO Gear SE460	BON NOCK AX460 BON NOCK M460	Daphne Super Gear Oil 460	DIAMOND SUPER GEARLUBE SP 460	REDUCTUS 460 ES GEAR G460	OMALA OIL 460 G-C OIL 460SE	SPARTAN EP460	MOBILGEAR SHC460 MOBIL GLYGOYLE 80
600	COSMO Gear SE680	BON NOCK AX680 BON NOCK M680	Daphne Super Gear Oil 680	DIAMOND SUPER GEARLUBE SP 680	REDUCTUS 680 ES GEAR G680	OMALA OIL 680	SPARTAN EP680	MOBILGEAR 636 MOBILGEAR SHC680
Above 1000	COSMO Gear SE4600	BON NOCK M1800 BON NOCK M3800	Daphne Super Gear Oil 1500 Daphne Super Gear Oil 4600	DIAMOND SUPER GEARLUBE SP 1800				

Industrial gear oil (For Worm gear)

ISO viscosity grade ISO VG cst (40°C)	COSMO	NISSEKI (Shin Nippon Oil)	IDEMITSU	MITSUBISHI	JOMO	SHOWA SHELL	ESSO	MOBIL
220	COSMO Gear W220	BON NOCK EX220 BON NOCK M220	Super Gear Oil 220	DIAMOND WORM GEARLUBE 220 (N)	REDUCTUS 220	TIVELA OIL SB220EP VITREA OIL220	SPARTAN EP220	MOBILGEAR 630
320	COSMO Gear W320	BON NOCK EX320 BON NOCK M320	Super Gear Oil 320	DIAMOND WORM GEARLUBE 380 (N)	REDUCTUS 320	VITREA OIL320	SPARTAN EP320	MOBILGEAR 632
460	COSMO Gear W460	BON NOCK EX460 BON NOCK M460	Daphne Worm Gear Oil 460 Super Gear Oil 460		REDUCTUS 460	TIVELA OIL SD460EP VITREA OIL460	SPARTAN EP460 CYLESSO TK460	MOBILGEAR 600W MOBILGEAR 634 SUPER CYLINDER OIL

Chapter 7. Oscillation and Noise level for Gear

7.1 Cause and solution for noise and oscillation

During operation of machine, make sure that gearing sound can be heard. 500 to 5,000 Hz is comfortable sound frequency for humans. Even if it is not loud, depending on the frequency component or the environment where the gears are used, such sound may feel unpleasant. Occurrence of noise is often blamed on the gear. However, noise problems are not solely from gear but may also include causes from designing error to lubrication. Refer to Fig. 1 for cause and solution.

Refer to Fig. 1 to reduce the noise level by following solutions.

1) Improve the accuracies of gear and gear assembly. → (Preventing at source)

2) For gear, axis and gearbox, provide suitable material and design to reduce noise. → (Reduce the cause of noise level)
(avoid resonance and quick attenuation)

3) Provide a sealed type of gearbox to shut in the noise. → (Shield and cover)

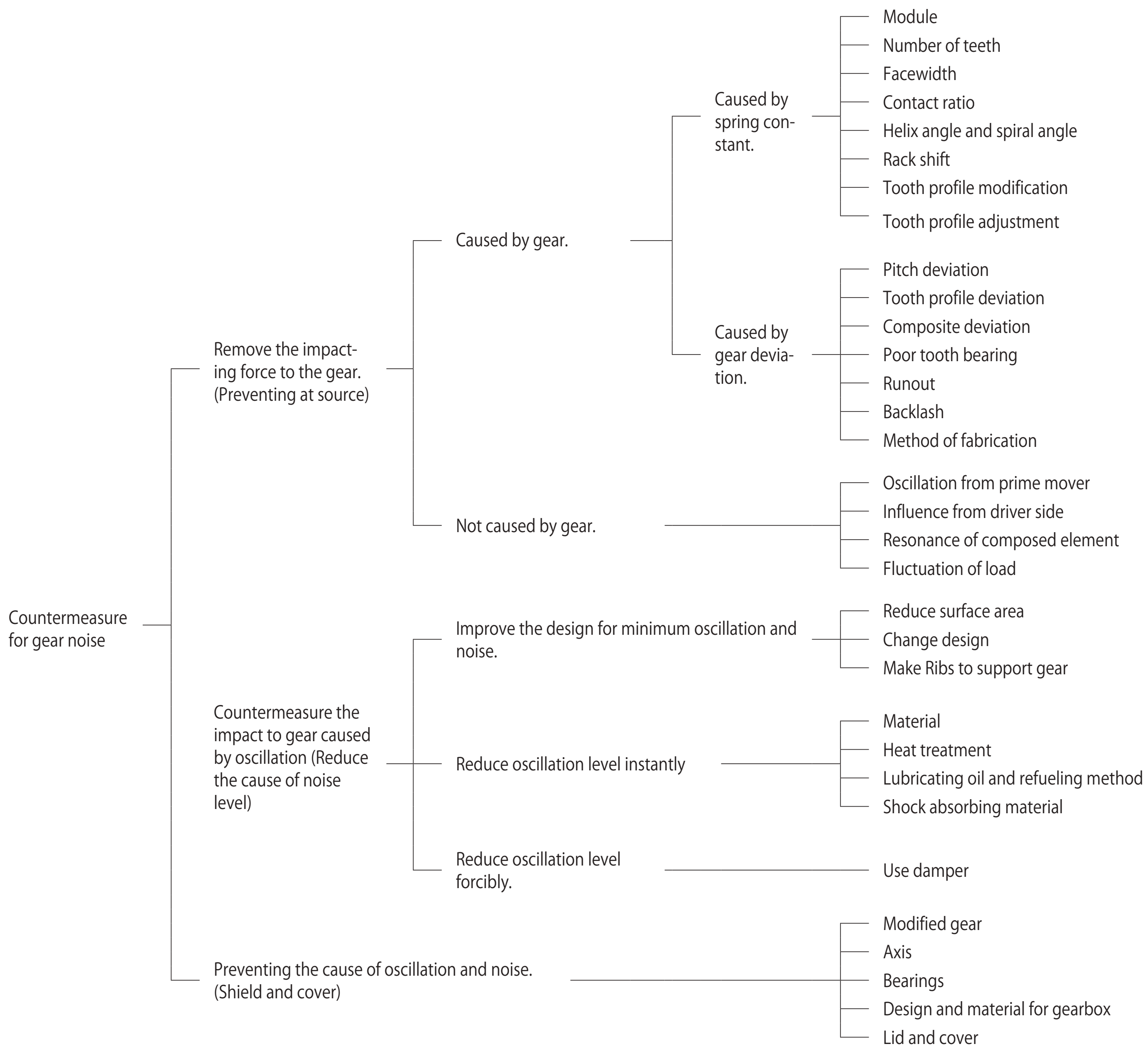
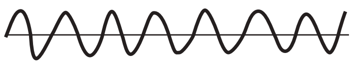
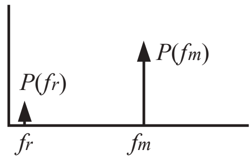
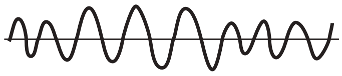
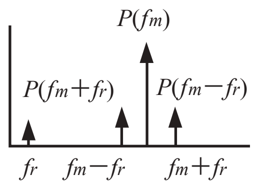
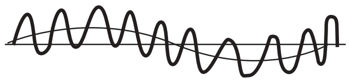
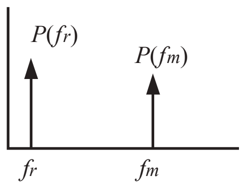
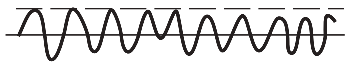
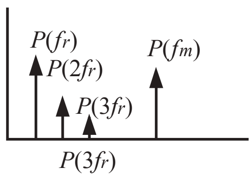
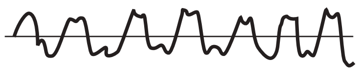
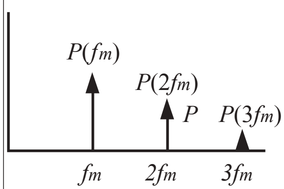
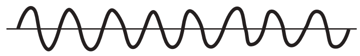
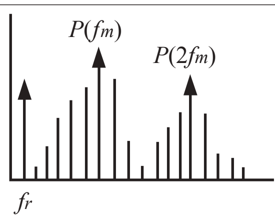


Fig. 1 Cause and solution for gear noise

7.2 Analyze the cause of noise by frequency constituent (Low frequency zone)

When gear causes noise and oscillation, analysis at low frequency zone will show the frequency constituent as seen in Fig. 2. Therefore deviation for the cause of noise can be found.
For cases with unusual localized noise from gear, an analysis at high frequency zone will be accurate but its description is omitted here.

Fig. 2 Oscillation from gear (Low frequency)

Condition of Gear	Time domain	Frequency domain
Normal		
Misalignment of gear axis		
Offcentre		
Part Abnormality		
Wear		
Pitch deviation	 Amp.Mod.+Freg-Mod	

f_m : Engaging frequency

$f_m = z \times \frac{n}{60}$

z : Number of teeth

f_r : Revolution frequency

$f_r = \frac{n}{60}$

n : Revolution per minute

Chapter 8 Gear damage

Table 1 shows causes for gear damage and its countermeasures.

Table 1. Causes of gear damage and its countermeasure

Technical Know How - Design for gear strength extracted from JSME (Japan Society of Mechanical Engineers), and others.

Damage	Details of damage	Condition of flank	Cause	Solution
1 Breakage	a. Overload breakage	Found crystallization on surface of breakage flank similar to surface of cast iron.	Overload. Poor tooth contact. Misusage.	Find the cause of overload. Comprehend usage conditions.
	b. Fatigue breakage	Flank breakage has discoloration but less damage than overload breakage.	Error in actual load calculation. Unsuitable shape of Dedendum.	Improve gear data for Tooth thickness. Exchange material and heat treatment.
	c. Shearing breakage	Found large plastic deformation at breakage surface.	Overload. Defective and unsuitable material.	Practice warm up and test run. Exchange material and heat treatment.
	d. Impact breakage	Found crystallization on the surface of breakage flank similar to surface of cast iron.	Oscillation of bearings from impulse load.	Improve axis and bearing stability to ease impulse load.
2 Wear	a. Abrasive wear	Found small scratches or grooves at sliding direction of flank.	Metallic dust from worn gear and bearing.	Purify lubricating oil. Exchange material and heat treatment
	b. Scratching	Found rough scratches at sliding direction. Scratches are larger and deeper than Abrasive wear.	Larger particles than ‘a’ and waste objects.	Countermeasure is the same as 'a'.
	c. Corrosive wear	Found rough pockmarks at flank.	Unsuitable lubricating oil. Oxidation of flank. Water contamination.	Exchange type of lubricating oil. Improve countermeasures for moisture and waterproof.
	d. Fretting	Found surface damaged with rust and oxidation by chemical change.	Relative reciprocation motion from minute oscillation at surface of contact.	Decrease oscillation. Improve surface hardening.
	e. Burning	Found discoloration and loss of hardness due to high temperature from excessive wear.	Inferior lubricating oil. Overload. Excessive speed. Increased temperature.	Exchange type of lubricating oil. Improve method of lubrication.
	f. Normal wear	This wear is within expectation of gear’s lifespan.	As expected and unavoidable.	
	g. Moderate wear	Found flank with over excessive mark cannot engage normally.	Unsuitable lubricating oil. Oxidation of flank. Water contamination.	Countermeasure is the same as 'a'. Practice warm up and test run.
3 Plastic deformation	a. Rippling	Found marks of corrugation or scale on contact area at flank.	Extreme sliding load. Deterioration of material or lubricating oil.	Improve gear strength and usage condition. Use extreme additive oil.
	b. Rolling	Found polish mark at flank and curled Tooth tip.	Heavy load. Insufficient robustness, or hardness of material.	Use shock absorber for design of driver side. Improve material and heat treatment.
	c. Peeling	Commonly, interpreted as rolling.	Commonly, interpreted as rolling.	Same as above. Practice warm up and test run. Improve assembly accuracy.
	d. Plastic flow	Same as above and usually found in soft materials.	Same as above. Insufficient robustness or hardness of material.	Improve strength of material. Practice warm up and test run.
	e. Collapsing tooth	All teeth collapsed.	Same as above. Dedendum stress exceeds elastic limit.	Same as above. Re-calculate conditions of load and gear data.
4 Fatigue of flank	a. Pitting	Found small pockmarks just under pitch line.	Metal fatigue from repeated stress.	Improve gear strength, material and heat treatment.
	b. Spalling	Detached large pieces of metal fragments from flank.	Metal fatigue under surface from repeated stress.	Same as above. Practice warm up and test run. Use extreme active oil.
	c. Case crush	Found extensive range of hardness layer detached from flank.	Same as above. Extreme residual stress of core.	Same as above. Improve flank and core hardness. Design radiator for gear.
5 Thermal damage	a. Scoring	Found many scratches and fusion marks at sliding direction of flank.	Extreme load. Metal contact from lack of oil film.	Emphasize on warm up and test run. Exchange the lubricating oil.
	b. Sand burning	Found extreme cohesion and fusion at final form of scoring.	Same as above. Inferior lubricating oil and accuracy. Increased temperature.	Same as above. Re-examine heat treatment. Improve gear accuracy.
6 Others	a. Damage and wear from interference	Found large scratches, exfoliation and falling apart at area of Tooth tip and Dedendum.	Inferiors design. Insufficient backlash.	Provide proper backlash to gears. Improve gear accuracy.
	b. Damage and wear from waste object	Found several detached tooth in various conditions.	Waste objects from inside and outside of equipment.	Improve assembly method. Remove waste objects. Seal for dust proof.
	c. Rust and corrosion	Found rust and corrosion on flank.	Chemical changes of lubricating oil, intrusion of impurities and water.	Add anti-corrosion agent to lubricating oil. Prevent intrusion of water and acids.

Chapter 9 Calculations for types of gear

9.1 Calculation for Standard spur gear

1. Finishing method		4. Module	$m =$	7. Addendum	$h_a = m =$
2. Number of teeth of Pinion	$z_1 =$	5. Reference pressure angle	$\alpha =$	8. Dedendum	$h_f = m + c =$
3. Number of teeth of Gear	$z_2 =$	6. Bottom clearance	$c =$	9. Tooth depth	$h = h_a + h_f =$

Gear terms	Pinion 1	Gear 2
10. Centre distance	$a = \dfrac{m(z_1 + z_2)}{2} = \dfrac{d_1 + d_2}{2}$	
11. Reference diameter	$d_1 = z_1 m$	$d_2 = z_2 m$
12. Tip (Outside) diameter	$d_{a1} = d_1 + 2h_a = m(z_1 + 2)$	$d_{a2} = d_2 + 2h_a = m(z_2 + 2)$
13. Sector span of teeth	$z_{m1} = \dfrac{\alpha z_1}{180} + 0.5$	$z_{m2} = \dfrac{\alpha z_2}{180} + 0.5$
14. Sector span	$W_1 = m\cos\alpha\{\pi(z_{m1} - 0.5) + z_1 \mathrm{inv}\alpha\}$	$W_2 = m\cos\alpha\{\pi(z_{m2} - 0.5) + z_2 \mathrm{inv}\alpha\}$
15. Base diameter	$d_{b1} = d_1 \cos\alpha$	$d_{b2} = d_2 \cos\alpha$
16. Circular pitch	$p = \dfrac{\pi d_1}{z_1} = \pi m$	$p = \dfrac{\pi d_2}{z_2} = \pi m$
17. Circular thickness	$s = \dfrac{p}{2} = \dfrac{\pi m}{2}$	
18. Base pitch	$p_b = \dfrac{\pi d_{b1}}{z_1} = \dfrac{\pi d_1 \cos\alpha}{z_1} = \dfrac{\pi d_2 \cos\alpha}{z_2} = \pi m \cos\alpha$	
19. Working depth	$h_1 = h_{a1} + h_{a2}$	
20. Transverse contact ratio	$\varepsilon_a = \dfrac{\sqrt{\left(\dfrac{d_{a1}}{2}\right)^2 - \left(\dfrac{d_{b1}}{2}\right)^2} + \sqrt{\left(\dfrac{d_{a2}}{2}\right)^2 - \left(\dfrac{d_{b2}}{2}\right)^2} - \dfrac{m(z_1 + z_2)}{2}\sin\alpha}{\pi m \cos\alpha}$	

9.2 Calculation for Standard Internal gear

1. Finishing method		4. Module	$m =$	7. Addendum	$h_a = m =$
2. Number of teeth of Pinion	$z_1 =$	5. Reference pressure angle	$\alpha =$	8. Dedendum	$h_f = m + c =$
3. Number of teeth of Gear	$z_2 =$	6. Bottom clearance	$c =$	9. Tooth depth	$h = h_a + h_f =$

Gear terms	Pinion 1	Gear 2
10. Centre distance	$a = \dfrac{m(z_2 - z_1)}{2} = \dfrac{d_2 - d_1}{2}$	
11. Reference diameter	$d_1 = z_1 m$	$d_2 = z_2 m$
12. Tip (Outside) diameter	$d_{a1} = d_1 - 2h_a = m(z_1 + 2)$	$d_a = d_2 + 2h_a = m(z_2 - 2)$
13. Root diameter	$d_{f1} = d_1 - 2h_f$	$d_{f2} = d_2 + 2h_f$
14. Sector span of teeth	$z_{m1} = \dfrac{\alpha z_1}{180} + 0.5$	$z_{m2} = \dfrac{\alpha z_2}{180} + 0.5$
15. Sector span	$W_1 = m\cos\alpha\{\pi(z_{m1} - 0.5) + z_1 \mathrm{inv}\alpha\}$	$W_2 = m\cos\alpha\{\pi(z_{m2} - 0.5) + z_2 \mathrm{inv}\alpha\}$
16. Transverse contact ratio	$\varepsilon_a = \dfrac{\sqrt{\left(\dfrac{d_{a1}}{2}\right)^2 - \left(\dfrac{d_{b1}}{2}\right)^2} - \sqrt{\left(\dfrac{d_{a2}}{2}\right)^2 - \left(\dfrac{d_{b2}}{2}\right)^2} + \dfrac{m(z_2 - z_1)}{2}\sin\alpha}{\pi m \cos\alpha}$	

Refer to 4.2 Method of Over balls or Rollers (page 63-64).

9.3 Calculation for the Normal standard helical gear

1. Finishing method
2. Number of teeth of Pinion $z_1 =$
3. Number of teeth of Gear $z_2 =$
4. Normal module $m_n =$
5. Normal pressure angle $\alpha_n =$
6. Reference cylinder angle $\beta =$
7. Bottom clearance $c =$
8. Addendum $h_a = m =$
9. Dedendum $h_f = m + c =$
10. Tooth depth $h = h_a + h_f =$

Gear terms	Pinion 1	Gear 2
11. Centre distance	$a = \dfrac{m_n(z_1+z_2)}{2\cos\beta} = \dfrac{d_1+d_2}{2}$	
12. Lead	$p_{z1} = \dfrac{\pi z_1 m_n}{\sin\beta}$	$p_{z2} = \dfrac{\pi z_2 m_n}{\sin\beta}$
13. Reference diameter	$d_1 = \dfrac{z_1 m_n}{\cos\beta}$	$d_2 = \dfrac{z_2 m_n}{\cos\beta}$
14. Tip (Outside) diameter	$d_{a1} = d_1 + 2h_a = d_1 + 2m_n$	$d_{a2} = d_2 + 2h_a = d_2 + 2m_n$
15. Sector span of teeth	$z_{m1} = \dfrac{\alpha_n z_{v1}}{180} + 0.5$ (Refer to Gear terms 19)	$z_{m2} = \dfrac{\alpha_n z_{v2}}{180} + 0.5$
16. Sector span	$W_1 = m_n\cos\alpha_n\{\pi(z_{m1}-0.5) + z_1\text{inv}\,\alpha_t\}$	$W_2 = m_n\cos\alpha_n\{\pi(z_{m2}-0.5) + z_2\text{inv}\,\alpha_t\}$
17. Transverse module	$m_t = \dfrac{d_1}{z_1} = \dfrac{m_n}{\cos\beta}$	Note: May use vocabulary of gear instead of pinion. Change z_1 to z_2 if calculating for gear.
18. Normal module	$m_n = m_t\cos\beta = \dfrac{d_1\cos\beta}{z_1}$	Note: May use vocabulary of gear instead of pinion. Change z_1 to z_2 if calculating for gear.
19. Virtual number of teeth of spur gear	$z_{v1} = \dfrac{z_1}{\cos^3\beta}$	$z_{v2} = \dfrac{z_2}{\cos^3\beta}$
20. Transverse pitch	$p_t = \dfrac{p_n}{\cos\beta}$	
21. Normal pitch	$p_n = p_t\cos\beta = \pi m_n$	
22. Transverse base pitch	$p_{bt} = \dfrac{\pi d_1\cos\alpha_t}{z_1} = \dfrac{\pi m_n\cos\alpha_n}{\cos\beta_b}$	Note: May use vocabulary of gear instead of pinion. Change z_1 to z_2 if calculating for gear.
23. Normal base pitch	$p_{bn} = \dfrac{\pi d_{b1}\cos\beta}{z_1} = \dfrac{\pi d_1\cos\alpha_t\cos\beta}{z_1}$	Note: May use vocabulary of gear instead of pinion. Change z_1 to z_2 if calculating for gear.
24. Transverse circular thickness	$s = \dfrac{p_t}{2} = \dfrac{\pi m_n}{2\cos\beta}$	
25. Normal circular thickness	$s_n = \dfrac{p_t}{2} = \dfrac{\pi m_n}{2} = s_t\cos\beta$	
26. Base diameter	$d_{b1} = d_1\cos\alpha_t = \dfrac{z_1 m_n\cos\alpha_\omega}{\cos\beta_{b1}}$	$d_{b2} = d_2\cos\alpha_t = \dfrac{z_2 m_n\cos\alpha_\omega}{\cos\beta_{b2}}$
27. Transverse pressure angle	$\alpha_t = \tan^{-1}\left(\dfrac{\tan\alpha_n}{\cos\beta}\right)$ or $\tan\alpha_t = \dfrac{\tan\alpha_n}{\cos\beta}$	
28. Normal pressure angle	$\alpha_n = \tan^{-1}(\tan\alpha_t\cos\beta)$ or $\tan\alpha_n = \tan\alpha_t\cos\beta$	
29. Reference cylinder helix angle	$\beta = \tan^{-1}\left(\dfrac{\pi d_1}{p_{z1}}\right)$	Note: May use vocabulary of gear instead of pinion. Change z_1 to z_2 if calculating for gear.
30. Base cylinder helix angle	$\beta_b = \tan^{-1}\left(\dfrac{\pi d_{b1}}{p_{z1}}\right)$	Note: May use vocabulary of gear instead of pinion. Change z_1 to z_2 if calculating for gear.
31. Contact ratio	$\varepsilon_a = \dfrac{\sqrt{\left(\frac{d_{a1}}{2}\right)^2 - \left(\frac{d_{b1}}{2}\right)^2} + \sqrt{\left(\frac{d_{a2}}{2}\right)^2 - \left(\frac{d_{b2}}{2}\right)^2} - \frac{m_t(z_1+z_2)}{2}\sin\alpha_t}{\pi m_t\cos\alpha_t}$	Note: This does not apply to Crossed helical gear (Screw gear).

9.4 Calculation for Crossed helical gear (Screw gear)

Crossed helical gear can use the same calculation formula of Normal standard helical gear taking careful consideration that Reference pitch cylindrical helix angle β and Transverse pressure angle α_t are different between Pinion and Gear (Except item 31 in section 9.3)

Gear terms	Same direction of Helix between both gears	Different Helix directions between both gears
1. Shaft angle	$\Sigma = \beta_1 + \beta_2$	$\Sigma = \beta_1 - \beta_2$ or $= \beta_2 - \beta_1$

9.5 Calculation for Worm gear pair

1. Finishing method of Worm gear pair
2. Finishing method of Worm wheel
3. Number of thread/s for Worm gear $z_1 =$
4. Number of teeth for Worm wheel $z_2 =$
5. Module $m_n =$
6. Reference pressure angle $\alpha =$
7. Reference diameter of Worm gear $d_1 =$
8. Bottom clearance $c =$
9. Addendum $h_a = m_n \quad h_a = m_x$
10. Dedendum $h_f = h_a + c =$
11. Tooth depth $h = h_a + h_f =$

Gear terms	Worm gear	Worm wheel 2
12. Lead	$p_z = z_1 p_x = z_1 \pi m_x = \pi d_1 \tan\gamma$	
13. Reference cylinder lead angle	$\gamma = \tan^{-1}\left(\frac{p_z}{\pi d_1}\right) = \tan^{-1}\left(\frac{z_1}{Q}\right) = \sin^{-1}\left(\frac{z_1 m_n}{d_1}\right)$	
14. Reference diameter	$d_1 = m_x Q = \frac{p_z}{\pi \tan\gamma}$	$d_2 = z_2 m_x = \frac{z_2 m_n}{\cos\gamma}$
15. Centre distance	$\alpha = \frac{d_1 + d_2}{2} = \frac{(Q + z_2)m_x}{2}$ (Standard worm wheel) $\alpha = \left(\frac{Q + z_2}{2} + x\right)m_x$ (Rack shifted worm wheel)	
16. Axial module	$m_x = \frac{m_n}{\cos\gamma} = \frac{p_x}{\pi}$	
17. Normal module	$m_n = m_x \cos\gamma = \frac{p_x \cos\gamma}{\pi}$	
18. Axial pitch	$p_x = p_t = \pi m_x = \frac{p_z}{z_1} = \frac{\pi m_n}{\cos\gamma}$	$p_t = p_x = \frac{\pi d_2}{z_2} = \frac{p_n}{\cos\gamma}$
19. Normal pitch	$p_n = p_x \cos\gamma$	$p_n = \pi m_n = p_t \cos\gamma$
20. Axial pressure angle	$\alpha_x = \tan^{-1}\left(\frac{\tan\alpha_n}{\cos\gamma}\right)$	
21. Normal pressure angle	$\alpha_n = \tan^{-1}(\tan\alpha_x \cos\gamma)$	
22. Rack shift coefficient	$x_1 = 0$	$x_2 = \frac{a - 0.5(d_1 + d_2)}{m_x}$
23. Gorge radius		$r_t = 0.5d_1 - h_a = a - \frac{d_r}{2}$
24. Throat diameter		$d_T = (z_2 + 2x_2)m_x + 2h_a$
25. Tip (Outside) diameter	$d_{a1} = d_1 + 2h_a$	1 $d_{a2} = d_2 + (2x_2 + 3.5)m_x$ 2 $d_{a2} = d_r + (d_1 - 2m_x)\left(1 - \cos\frac{\phi}{2}\right)$
26. Facewidth	$b_1 = 4.5\pi m_x$ Or $= p_x\left(4.5 + \frac{2 \bullet z_2}{100}\right)$	$b_2 = m_x\sqrt{7Q - 12.25}$ Or $= 2\sqrt{(d_1 + h_a)h_a} + 0.5p_x$
27. Diameter quotient	$Q = \frac{d_1}{m_x}$	

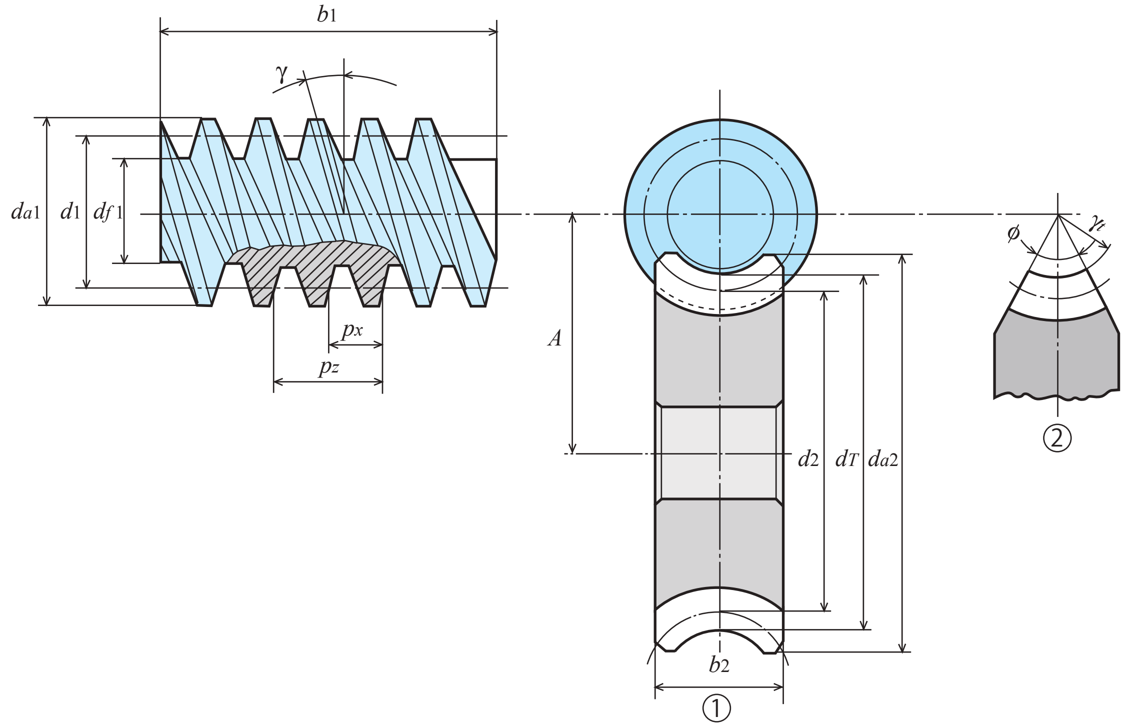


Fig. 1 Worm gear pair

9.6 Calculation for Gleason system Straight bevel gear

1. Number of teeth for pinion	$z_1 =$	5. Working depth	$h' = 2.000m =$
2. Number of teeth for gear	$z_2 =$	6. Tooth depth	$h = 2.188m + 0.05 =$
3. Module	$m =$	7. Reference pressure angle	$\alpha = 20°$
4. Facewidth	$b =$	8. Shaft angle	$\sum = 90°$

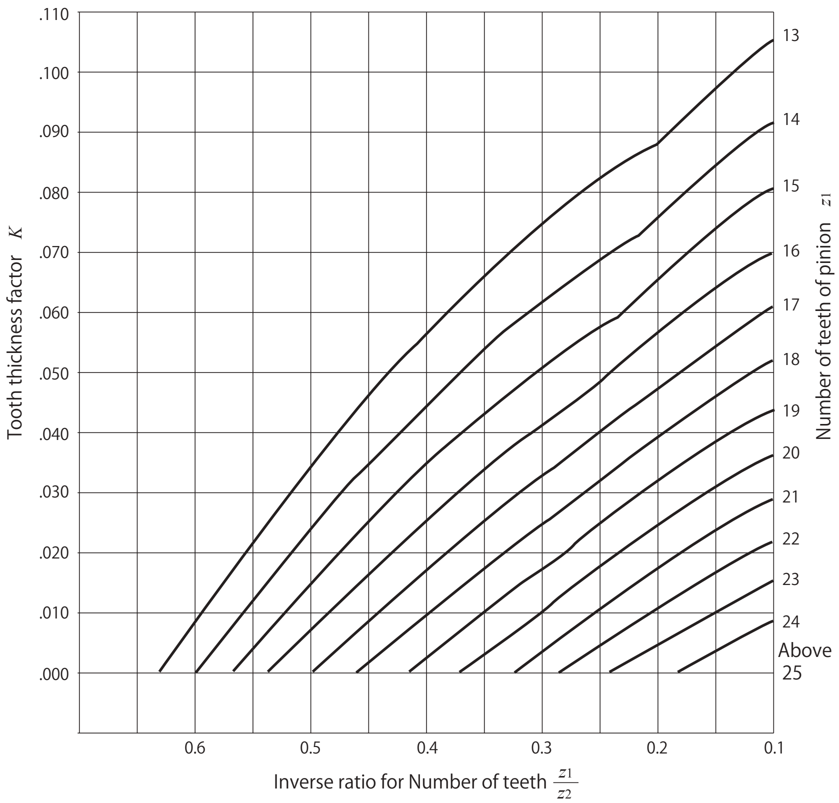
Gear terms	Pinion 1	Gear 2
9. Reference diameter	$d_1 = z_1 m$	$d_2 = z_2 m$
10. Reference pitch angle	$\delta_1 = \tan^{-1}\frac{z_1}{z_2}$	$\delta_2 = 90° - \delta_1$
11. Cone distance (outer)	$R_e = \frac{d_2}{2\sin\delta_2}$	
12. Circular pitch	$p = \pi m = 3.1416m$	
13. Addendum	$h_{a1} = h' - h_{a2}$	$h_{a2} = 0.540m + \frac{0.460m}{\left(\frac{z_2}{z_1}\right)^2}$
14. Dedendum[1]	$h_{f1} = 2.188m - h_{a1}$	$h_{f2} = 2.188m - h_{a2}$
15. Bottom clearance	$c = h - h'$ (Parallel bottom clearance)	
16. Dedendum angle[2]	$\theta_{f1} = \tan^{-1}\frac{h_{f1}}{R_e}$	$\theta_{f2} = \tan^{-1}\frac{h_{f2}}{R_e}$
17. Tip angle	$\delta_{a1} = \delta_1 + \theta_{f2}$	$\delta_{a2} = \delta_2 + \theta_{f1}$
18. Root angle	$\delta_{f1} = \delta_1 - \theta_{f1}$	$\delta_{f2} = \delta_2 - \theta_{f2}$
19. Tip (Outside) diameter (heel)	$d_{a1} = d_1 + 2h_{a1}\cos\delta_1$	$d_{a2} = d_2 + 2h_{a2}\cos\delta_2$
20. Pitch apex to crown	$X_1 = \frac{d_2}{2} - h_{a1}\sin\delta_1$	$X_2 = \frac{d_1}{2} - h_{a2}\sin\delta_2$
21. Circular thickness	$s_1 = p - s_2$	$s_2 = \frac{p}{2} - (h_{a1} - h_{a2})\tan\alpha - K \bullet m$[3]
22. Backlash	$j_n =$ Refer to Backlash for Bevel gear in JIS B 1705.	
23. Chordal tooth thickness	$\bar{s} = s_1 - \frac{(s_1)^3}{6(d_1)^2}$	$\bar{s} = s_2 \frac{(s_2)^3}{6(d_2)^2}$
24. Chordal height	$\bar{h}_1 = h_{a1} + \frac{(s_1)^2\cos\delta_1}{4d_1}$	$\bar{h}_2 = h_{a2} + \frac{(s_2)^2\cos\delta_2}{4d_2}$
25. Axial facewidth	$X_{b1} = \frac{b\cos\delta_{a1}}{\cos\theta_{f2}}$	$X_{b2} = \frac{b\cos\delta_{a2}}{\cos\theta_{f1}}$
26. Tip (Inside) diameter (toe)	$d_{i1} = d_{a1} - \frac{2b\sin\delta_{a1}}{\cos\theta_{f2}}$	$d_{i2} = d_{a2} - \frac{2b\sin\delta_{a2}}{\cos\theta_{f1}}$
27. Material angle	$\theta_{x1} = 90° - \theta_{f2}$	$\theta_{x2} = 90° - \theta_{f1}$
28. Material angle	$\theta_{y1} = 90° - \delta_1$	$\theta_{y2} = 90° - \delta_2$

Calculation for Gleason system Angular straight bevel gear

Calculation for Gleason system Angular straight bevel gear

Gear terms	Pinion 1	Gear 2
10. Pitch angle	Refer to next page for Standard angular straight bevel gear	
13. Addendum	$h_{a1} = h' - h_{a1}$	$h_{a1} = 0.54m + \dfrac{0.46m}{\left(\dfrac{z_2 \cos\delta_1}{z_1 \cos\delta_2}\right)}$
20. Pitch apex to crown	$X_1 = R_e \cos\delta_1 - h_{a1}\sin\delta_1$	$X_2 = R_e \cos\delta_2 - h_{a2}\sin\delta_2$

Note (1) Actual dedendum is 0.05 mm longer than calculated value. (2) Dedendum angle θ_a is equivalent to Dedendum angle θ_f for Mating gear. (3) Obtain factor K from Fig. 2



Note) $u = \dfrac{z_1}{z_2}$ when $z_2 = 1.5$, $z_1 = 1.0$ or above 25, $K = 0$.

Fig. 2 Tooth thickness factor K

Table 1. Minimum number of teeth to prevent Undercut

α=20°		α=14.5°	
No. of teeth of pinion z_1	No. of teeth of gear z_2	No. of teeth of pinion z_1	No. of teeth of gear z_2
z_1	z_2	z_1	z_2
13	30	24	57
14	20	25	40
15	17	26	35
16	16	27	31
		28	29
		29	29

9.7 Calculation for Standard straight bevel gear

1. Number of teeth of pinion z_1 $z_1 =$
2. Number of teeth of gear z_2 $z_2 =$
3. Module $m =$
4. Reference pressure angle $\alpha =$
5. Facewidth $b =$
6. Bottom clearance $c = 0.25m$
7. Addendum $h_a = m =$
8. Dedendum $h_f = 1.25m =$
9. Tooth depth $h = 2.25m =$
10. Shaft angle $\sum = 90°$

Gear terms	Pinion 1	Gear 2
11. Reference diameter	$d_1 = z_1 m$	$d_2 = z_2 m$
12. Reference pitch angle	$\delta_1 = \tan^{-1}\frac{z_1}{z_2}$	$\delta_2 = 90° - \delta_1$
13. Cone distance	$R_n = \frac{d_2}{2\sin\delta_2}$	
14. Addendum angle	$\theta_a = \tan^{-1}\frac{h_a}{R_e}$	
15. Dedendum angle	$\theta_f = \tan^{-1}\frac{h_f}{R_e}$	
16. Tip angle	$\delta_{a1} = \delta_1 + \theta_a$	$\delta_{a2} = \delta_2 + \theta_a$
17. Root angle	$\delta_{f1} = \delta_1 - \theta_f$	$\delta_{r2} = \delta_2 - \theta_f$
18. Tip (Outside) diameter (heel)	$d_a = d_1 + 2h_a\cos\delta_1$	$d_a = d_2 + 2h_a\cos\delta_2$
19. Tip (Inside) diameter (toe)	$d_{i1} = d_{a1} - \frac{2b\sin\delta_{a1}}{\cos\theta_a}$	$d_{i2} = d_{a2} - \frac{2b\sin\delta_{a2}}{\cos\theta_a}$
20. Material angle	$\theta_{x1} = 90° - \theta_a = \theta_{x2}$	$\theta_{x2} = 90° - \theta_a = \theta_{x1}$
21. Material angle	$\theta_{y1} = 90° - \delta_1 = \delta_2$	$\theta_{y2} = 90° - \delta_2 = \delta_1$
22. Pitch apex to crown	$X_1 = \frac{d_2}{2} - h_a\sin\delta_1$	$X_2 = \frac{d_1}{2} - h_a\sin\delta_2$
23. Axial facewidth	$X_{b1} = \frac{b\cos\delta_{a1}}{\cos\theta_a}$	$X_{b2} = \frac{b\cos\delta_{a2}}{\cos\theta_a}$
24. Chordal tooth thickness	$\bar{s}_1 = z_{v1} m\sin\theta_{v1} \fallingdotseq s - \frac{s^3}{6d_1^2}$	$\bar{s}_2 = z_{v2} m\sin\theta_{v2} \fallingdotseq s - \frac{s^3}{6d_2^2}$
25. Chordal height	$\bar{h}_1 = m + R_{v1}(1 - \cos\theta_{v1}) \fallingdotseq m + \frac{s^2\cos\delta_1}{4d_1}$	$\bar{h}_2 = m + R_{v2}(1 - \cos\theta_{v2}) \fallingdotseq m + \frac{s^2\cos\delta_2}{4d_2}$

Note ∗ Table of Chordal tooth thickness can be used assuming Standard spur gear with Number of teeth $Zv = z/\cos\delta$.

Standard angular straight bevel gear

Calculation for Standard angular straight bevel gear is the same as 9.7 except (12) and (22)

Gear terms	Pinion 1	Gear 2
12. Pitch angle	If shaft angle Σ is smaller than 90° $\delta_1 = \tan^{-1} \dfrac{\sin\Sigma}{\frac{z_2}{z_1} + \cos\Sigma}$ If shaft angle Σ is greater than 90° $\delta_1 = \tan^{-1} \dfrac{\sin(180° - \Sigma)}{\frac{z_2}{z_1} - \cos(180° - \Sigma)}$	$\delta_2 = \Sigma - \delta_1$
22. Pitch apex to crown	$X_1 = R_e\cos\delta_1 - h_a\sin\delta_1$	$X_2 = R_e\cos\delta_2 - h_a\sin\delta_2$

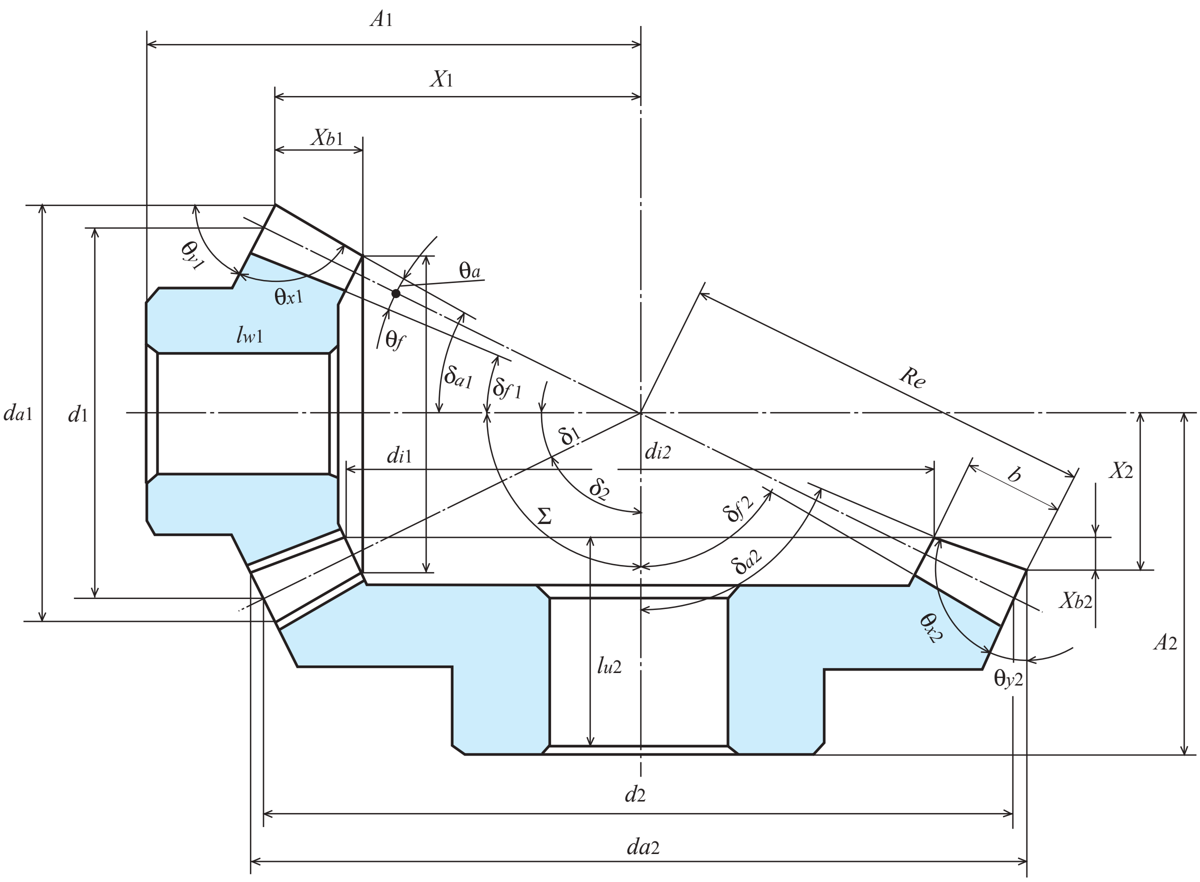


Fig. 3 Bevel gear

9.8 Calculation for Gleason system spiral bevel gear

1. Number of teeth of pinion z_1 $z_1 =$
2. Number of teeth of gear z_2 $z_2 =$
3. Module $m =$
4. Facewidth $b =$
5. Working depth $h' = 1.700m =$
6. Tooth depth $h = 1.888m =$
7. Pressure angle $\alpha =$ (standard is 20°)
8. Shaft angle $\Sigma = 90°$

Gear terms	Pinion 1	Gear 2
9. Reference diameter	$d_1 = z_1 m$	$d_2 = z_2 m$
10. Pitch angle	$\delta_1 = \tan^{-1}\dfrac{z_1}{z_2}$	$\delta_2 = 90° - \delta_1$
11. Cone distance	$R_e = \dfrac{d_2}{2\sin\delta_2}$	
12. Circular pitch	$p = \pi m = 3.14159m$	
13. Addendum	$h_{a1} = h' - h_{a2}$	$h_{a2} = 0.460m + \dfrac{0.390m}{\left(\dfrac{z_2}{z_1}\right)^2}$
14. Dedendum	$h_{f1} = h - h_{a1}$	$h_{f2} = h - h_{a2}$
15. Bottom clearance	$c = h - h'$ (Parallel bottom clearance)	
16. Dedendum angle[1]	$\theta_{f1} = \tan^{-1}\dfrac{h_{f1}}{R_e}$	$\theta_{f2} = \tan^{-1}\dfrac{h_{f2}}{R_e}$
17. Tip angle	$\delta_{a1} = \delta_1 + \theta_{f2}$	$\delta_{a2} = \delta_2 + \theta_{f1}$
18. Root angle	$\delta_{f1} = \delta_1 - \theta_{f1}$	$\delta_{f2} = \delta_2 - \theta_{f2}$
19. Tip (Outside) diameter (heel)	$d_{a1} = d_1 + 2h_{a1}\cos\delta_1$	$d_{a2} = d_2 + 2h_{a2}\cos\delta_2$
20. Pitch apex to crown	$X_1 = \dfrac{d_2}{2} - h_{a1}\sin\delta_1$	$X_2 = \dfrac{d_1}{2} - h_{a2}\sin\delta_2$
21. Tooth thickness[2]	$s_1 = p - s_2$	$s_2 = \dfrac{p}{2} - (h_{a1} - h_{a2})\dfrac{\tan\alpha_n}{\cos\beta} - Km$[3]
22. Backlash	$j_n =$ Refer to Gleason Works for backlash recommendation or Backlash for Bevel gear in JIS B 1705.	
23. Spiral angle	$\beta =$ (35° is standard)	
24. Shape of teeth		
25. Driving gear		
26. Revolving direction		
27. Axial facewidth	$X_{b1} = \dfrac{b\cos\delta_{a1}}{\cos\theta_{f2}}$	$X_{b2} = \dfrac{b\cos\delta_{a2}}{\cos\theta_{f1}}$
28. Tip (Inside) diameter (toe)	$d_{i1} = d_{a1} - \dfrac{2b\sin\delta_{a1}}{\cos\delta_{f2}}$	$d_{i2} = d_{a2} - \dfrac{2b\sin\delta_{a2}}{\cos\delta_{f1}}$
29. Material angle	$\theta_{x1} = 90° - \theta_{f2}$	$\theta_{x2} = 90° - \theta_{f1}$
30. Material angle	$\theta_{y1} = 90° - \delta_1$	$\theta_{y2} = 90° - \delta_2$

Note (1) Addendum angle θ_a is equivalent to Dedendum angle θ_f of Mating gear.
(2) Gear cutting by methods of Spread Blade and Single Side may use calculation formula from drawing. There are different calculations depending on gear cutting methods when using Gear tooth vernier calipers to calculate dimension of Sector span. Therefore designed Tooth thickness is necessary for reference.
(3) Obtain Factor K from Fig. 2 of page 112.

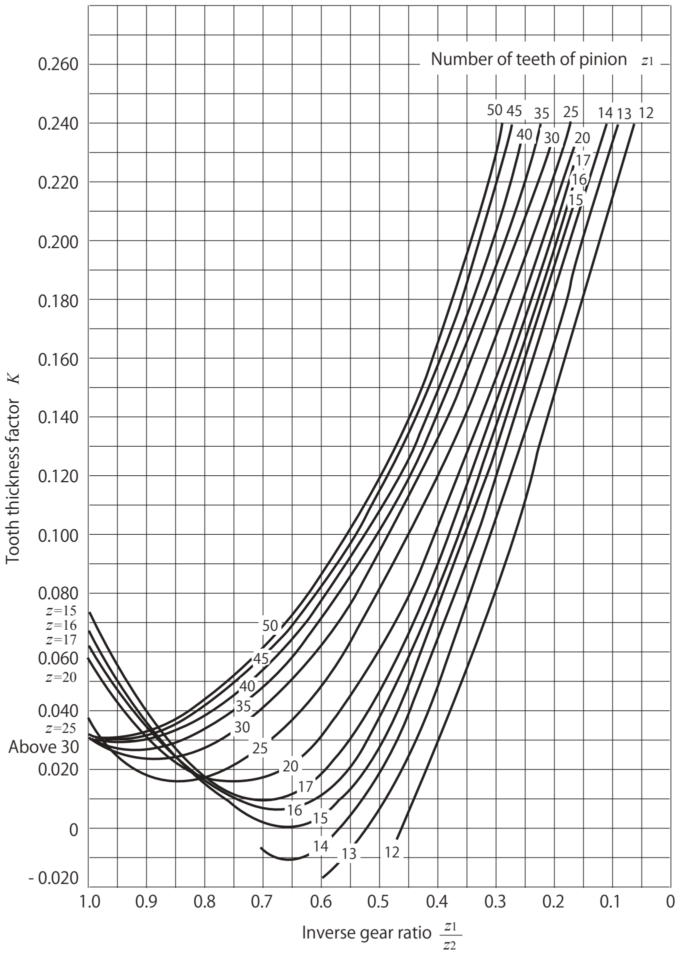


Fig. 4 Tooth thickness factor K

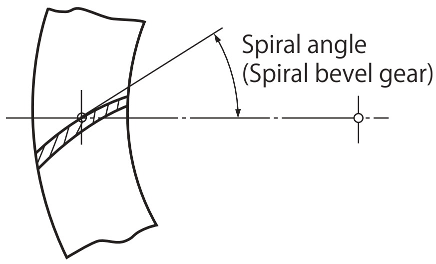


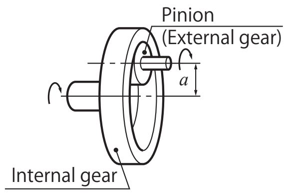
Fig. 5 Spiral angle for Spiral bevel gear
(Mean spiral angle)

Table 2. Minimum number of teeth to prevent Undercut

α=20°		α=16°		α=14.5°	
No. of teeth of pinion z_1	No. of teeth of gear z_2	No. of teeth of pinion z_1	No. of teeth of gear z_2	No. of teeth of pinion z_1	No. of teeth of gear z_2
z_1	z_2	z_1	z_2	z_1	z_2
12	26	16	59	19	70
13	22	17	45	20	60
14	20	18	36	21	42
15	19	19	31	22	40
16	18	20	29	23	36
17	17	21	27	24	33
		22	26	25	32
		23	25	26	30
		24	24	27	29
				28	28

9.9 Calculation for Planetary gear mechanism

1. Engagement between Internal gear and pinion (External gear)



Centre distance 'a' for the Internal gear trains are shorter than the External gear trains. Internal gear train operates in the same gear direction. Calculation formulas for transfer ratio 'u' are as follows.

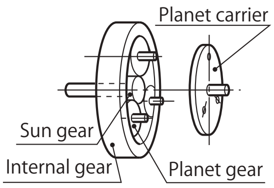
a) When pinion is driver

$$u = \frac{\text{No. of teeth of Pinion}}{\text{No. of teeth of Internal gear}} \text{(speed reduction)}$$

b) When Internal gear is driver

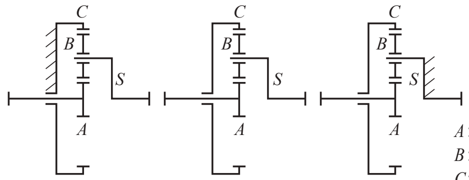
$$u = \frac{\text{No. of teeth of Internal gear}}{\text{No. of teeth of Pinion}} \text{(increase speed)}$$

2. Planetary gear mechanism



Most mechanism of Planetary gear comes with compact design and high reductive gear ratio consisting of Sun, Planet, Internal gears and Planet carrier.

Basic gear axis for Planetary gear train mechanism (2K-H)



(a) (Types of planetary) (b) (Types of solar) (c) (Types of star)

A : Sun gear
B : Planet gear
C : Internal gear
S : Planet carrier

Types of mechanism	Fixed member	Input	Output	Formula of gear ratio	Ratio range
(a) Types of planetary	Internal gear	Sun gear	Planet carrier	$\dfrac{1}{\frac{zC}{zA}+1}$	1/3 - 1/12
(b) Types of solar	Sun gear	Internal gear	Planet carrier	$\dfrac{1}{\frac{zA}{zC}+1}$	1/1.2 - 1/1.7
(c) Types of star	Planet carrier	Sun gear	Internal gear	$\dfrac{1}{-\frac{zC}{zA}}$	1/2 - 1/11

- z: No. of teeth. A & C: Sun and Internal gear
- '-' symbol indicates output revolving direction

Interference of Internal gear

Interference will occur when design provides insufficient Number of teeth between Internal and Planet gears (External gear) during assembly. Please refer to the causes and types of interference as follows.

Interference	Phenomenon	Cause	Interference	Phenomenon	Cause
Involute interference	Unworkable conditions when a Tooth tip of Internal gear cuts into Dedendum of pinion during operations.	Insufficient No. of teeth for pinion	Trimming interference	During assembling, pinion can be assembled to axial direction but not to radius direction.	Same as trochoid interference
Trochoid interference	Tip of pinion after engaging with Sun gear interferes to Tooth tip of Internal gear causing unworkable conditions.	Difference in No. of teeth between Internal and Planet is insufficient.	Fillet interference	Tooth tip of pinion touched Dedendum fillet of Internal gear causing unworkable condition.	Insufficient No. of teeth for pinion. (insufficient Tooth depth of pinion).

Relationship among the gears in a Planetary gear mechanism

When designing Planet gear, please achieve following conditions.

① No. of teeth of Internal gears = (No. of teeth of Sun gear + 2) × No. of teeth of Planet gear.

② $\dfrac{\text{No. of teeth of Internal gear} + \text{No. of teeth of Sun gear}}{\text{The number of planet gears}} = \text{Should be integer number}$

③ Prevent the Tip interference among Planetry gears.

$$m(Z_B + 2) < m(Z_A + Z_B)\ \sin\frac{\pi}{n}\ (n: \text{The number of Planet gears})$$

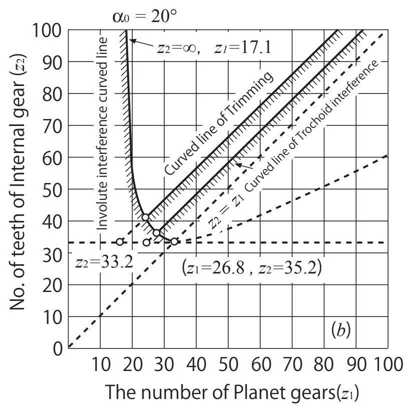


Fig. 6 Interference of Internal gear

Range of Number of teeth for pinion and KG-Internal gears

No. of teeth of Internal gear	Range of No. of teeth for pinion	No. of teeth of Internal gear	Range of No. of teeth for pinion
60	21 - 44	96	19 - 80
72	20 - 56	100	19 - 84
80	20 - 64	108	19 - 92
84	20 - 68	120	19 - 104
90	19 - 74		

9.10 Calculation for types of Gear

Calculation for Standard spur gear

Full depth tooth

Description / Tooth profile	Vocabulary	Pinion	Gear
Module	m	1.5	
No. of teeth	Z	20	60
Reference pressure angle	α	20°	
Rack shift coefficient	x	0.0000	0.0000
Addendum	h_a	1.500	1.500
Dedendum	h_f	1.875	1.875
Tooth depth	h	3.375	
Bottom clearance	c	0.375	
Reference diameter	d	30.000	90.000
Tip (Outside) diameter	d_a	33.000	93.000
Root Diameter	d_f	26.250	86.250
Base diameter	d_b	28.191	84.572
Base helix angle	β_b	0° 0′ 0″	
Centre distance	a	60.0000	
Working pressure angle	α_w	20° 0′ 0″	
Intermeshing PCD	d_w	30.000	90.000
Chordal height	$\bar{h}$	1.546	1.515
Chordal tooth thickness	$\bar{s}$	2.354	2.356
Sector span of teeth	z_m	(3)	(7)
Sector span	W	11.491	30.044
Over balls or Rollers	d_p	2.500	2.500
Over balls or Rollers dimension	d_m	33.268	93.309

Calculation for Standard internal gear

Full depth tooth

Description / Tooth profile	Vocabulary	Pinion	Gear
Module	m	1	
No. of teeth	z	20	100
Pressure angle	α	20°	
Addendum	h_a	1.000	1.000
Dedendum	h_f	1.250	1.250
Tooth depth	h	2.250	2.250
Bottom clearance	c	0.250	0.250
Reference diameter	d	20.000	100.000
Tip (Outside) diameter	d_a	22.000	98.000
Root diameter	d_f	17.500	102.500
Centre distance	α	40.000	
Transverse contact ratio	ε_a	1.860	
Sector span of teeth	z_m	(3)	(12)
Sector span	W	7.660	35.350

Calculation for Normal standard helical gear (based on Centre distance)

Full depth tooth

Tooth profile / Description	Vocabulary	Pinion	Gear
Module	m_n	2	
No. of teeth	z	30	60
Reference pressure angle	α	20°	
Reference cylinder helix angle	β	20° 0′ 0″	
Direction of helix		Right	Left
Rack shift coefficient	x_n	0.00000	0.0000
Addendum	h_a	2.000	2.000
Dedendum	h_f	2.500	2.500
Tooth depth	h	4.500	
Bottom clearance	c	0.500	
Reference diameter	d	63.851	127.701
Tip (Outside) diameter	d_a	67.851	131.701
Root diameter	d_f	58.851	122.701
Base diameter	d_b	59.540	119.081
Base helix angle	β_b	18° 44′ 50″	
Centre distance	a	95.7760	
Working pressure angle	α_w	21° 10′ 22″	
Intermeshing PCD	d_w	63.851	127.701
Chordal height	$\bar{h}$	2.034	2.017
Chordal tooth thickness	$\bar{s}$	3.141	3.141
Sector span of teeth	z_m	(5)	(9)
Sector span	W	27.572	52.193
Over balls or Rollers	d_p	3.500	3.500
Over balls or Rollers dimension	d_m	68.844	132.743

Calculation for Normal standard crossed helical gear (Screw gear)

Full depth tooth

Tooth profile / Description	Vocabulary	Pinion	Gear
Module	m_n	2	
No. of teeth	z	13	26
Reference pressure angle	α	20°	
Reference cylindrer helix angle	β	45° 0′ 0″	45° 0′ 0″
Direction of helix		Right	Left
Rack shift coefficient	x_n	0.0000	0.0000
Addendum	h_a	2.000	2.000
Dedendum	h_f	2.500	2.500
Tooth depth	h	4.500	
Bottom clearance	c	0.500	
Reference diameter	d	36.770	73.539
Tip (Outside) diameter	d_a	40.770	77.539
Root diameter	d_f	31.770	68.539
Base diameter	d_b	32.693	65.386
Base helix angle	β_b	41°38′ 28″	41° 38′ 28″
Centre distance	a	55.1543	
Normal working pressure angle	α_{wn}	20° 0′ 0″	20° 0′ 0″
Transverse working pressure angle	α_{wt}	27° 14′ 11″	27° 14′ 11″
Intermeshing PCD	d_w	36.770	73.539
Shaft angle	Σ	90° 0′ 0″	
Chordal height	$\bar{h}$	2.034	2.017
Chordal tooth thickness	$\bar{s}$	3.141	3.141
Sector span of teeth	z_m	(5)	(9)
Sector span	W	27.531	52.110
Over balls or Rollers	d_p	3.500	3,500
Over balls or Rollers dimension	d_m	41.487	78.583

Calculation for Normal worm gear and Worm wheel

Full depth tooth

Description / Tooth profile	Vocabulary	Worm gear	Worm wheel
Module	m_n	1.5 (m_a=1.5027)	
Reference pressure angle	α	20° (α_a=20° 2′ 0″)	
Addendum	h_a	1.500	
Dedendum	h_f	1.875	
Tooth depth	h	3.375	
Lead	p_z	4.7209	
Reference pitch	P	4.7209	
Reference cylinder lead angle	γ	3° 26′ 23″	
Centre distance	a	42.500	
Type of Worm wheel		******	Type I
No. of thread / No. of teeth	z_w/z_2	1	40
Reference diameter	d	25.000	60.108
Outside diameter	d_a	28.000	65.260
Root diameter	d_f	21.250	56.250
Rack shift coefficient	x	******	-0.0360
Diameter quotient	Q	16.6366	******
Throat diameter	d_T	******	63.000
Gorge radius	r_t	******	11.00
Facewidth	b	30.00	15.00
Chordal height	$\bar{h}$	1.500	1.468
Chordal tooth thickness	$\bar{s}$	2.356	2.316

Calculation for Standard straight bevel gear

Full depth tooth

Description / Tooth profile	Vocabulary	Pinion	Gear
Module	m	1.5	
No. of teeth	z	20	40
Reference pressure angle	α	20°	
Facewidth	b	10	
Addendum	h_a	1.500	
Dedendum	h_f	1.875	
Tooth depth	h	3.375	
Bottom clearance	c	0.375	
Shaft angle	Σ	90° 0′ 0″	
Cone distance	R_e	33.541	
Reference diameter	d	30.000	60.000
Pitch angle	δ	26° 33′ 54″	63° 26′ 6″
Addendum angle	θ_a	2° 33′ 38″	
Dedendum angle	θ_f	3° 11′ 59″	
Tip angle	δ_a	29° 7′ 32″	65° 59′ 44″
Root angle	δ_f	23° 21′ 56″	60° 14′ 7″
Outer tip diameter	d_a	32.683	61.342
Inner tip diameter	d_i	22.939	43.053
Pitch apex to crown	X	29.329	13.658
Axial facewidth	X_b	8.744	4.072
Tooth thickness	S	2.356	
Tooth angle		190.71(min)	
Angle of material	θ_x	87° 26′ 22″	87° 26′ 22″
Angle of material	θ_y	63° 26′ 6″	26° 33′ 54″
Chordal tooth thickness	$\bar{s}$	2.354	2.356
Chordal height	$\bar{h}$	1.541	1.510
Virtual number of teeth of spur gear(1)	z_v	22.361	89.443

Note(1) old gear terms adopted.

Calculation for Gleason system Straight bevel gear

Full depth tooth

Tooth profile \ Description	Vocabulary	Pinion		Gear
Module	m		1.5	
No. of teeth	z	20		40
Reference pressure angle	α		20°	
Facewidth	b		10	
Shaft angle	Σ		90° 0′ 0″	
Working depth	h_w		3.000	
Tooth depth	h		3.332	
Cone distance	R_e		33.5410	
Reference diameter	d	30.000		60.000
Pitch angle	δ	26° 33′ 54″		63° 26′ 6″
Addendum	h_a	2.018		0.983
Dedendum	h_f	1.265		2.300
Bottom clearance	c		0.332	
Addendum angle	θ_a	3° 55′ 19″		2° 9′ 33″
Dedendum angle	θ_f	2° 9′ 33″		3° 55′ 19″
Tip angle	δ_a	30° 29′ 13″		65° 35′ 38″
Root angle	δ_f	24° 24′ 22″		59° 30′ 47″
Outer tip diameter	d_a	33.609		60.879
Inner tip diameter	d_i	23.438		42.653
Pitch apex to crown	X	29.098		14.121
Axial facewidth	X_b	8.638		4.135
Tooth thickness	S	2.733		1.979
Tooth angle		187.2 (min)		187.2 (min)
Material angle	θ_x	86° 4′ 41″		87° 50′ 27″
Material angle	θ_y	63° 26′ 6″		26° 33′ 54″
Chordal height	$\bar{s}$	2.729		1.979
Chordal addendum	$\bar{h}$	2.073		0.990
Virtual No. of tooth of spur gear	z_v	22.361		89.443

Calculation for Gleason system Spiral bevel gear

Full depth tooth

Tooth profile \ Description	Vocabulary	Pinion		Gear
Module	m		1.5	
No. of teeth	z	20		40
Reference pressure angle	α		20°	
Facewidth	b		10	
Reference cylinder spiral angle	β		35° 0′ 0″	
Hand of spiral		左ねじれ		右ねじれ
Shaft angle	Σ		90° 0′ 0″	
Working depth	h_w		2.550	
Tooth depth	h		2.832	
Cone distance	R_e		33.541	
Reference diameter	d	30.000		60.000
Pitch angle	δ	26° 33′ 54″		63° 26′ 6″
Addendum	h_a	1.714		0.836
Dedendum	h_f	1.118		1.996
Addendum angle	θ_a	3° 24′ 19″		1° 54′ 34″
Dedendum angle	θ_f	1° 54′ 34″		3° 24′ 19″
Tip angle	δ_a	29° 58′ 13″		65° 20′ 40″
Root angle	δ_f	24° 39′ 20″		60° 1′ 47″
Pitch apex to crown	d_a	33.066		60.748
Inner tip diameter	d_i	23.057		42.561
Outer cone distance	X	29.234		14.252
Axial facewidth	X_b	8.678		4.174
Tooth thickness	S	2.834		1.878
Angle of material	θ_x	86° 35′ 41″		88° 5′ 26″
Angle of material	θ_y	63° 26′ 6″		26° 33′ 54″
Virtual number of teeth of spur gear(1)	z_v	40.681		162.724

Note(1) old gear term adopted.

9.11 Gear efficiency

(Reference for gears only)

Types of gear		Efficiency of gear
Spur gear		97 - 99%
Helical gear		97 - 99%
Bevel gear		96 - 99%
Worm gear	Single thread	45 - 55% *
	Double thread	55 - 65% *

*Above efficiency values are for KG STOCK GEARS only

Chapter 10 Calculation for Gear strength

10.1 Calculation of strength for Spur and Helical gears

There are calculations for Tooth bending strength (hereby called Bending strength), Surface durability and Scoring when considering gear strength. These are from ISO, JGMA, AGMA, DIN, BS and JSME.
KG had developed and marketed KG-CALMET for easy searching of suitable KG STOCK GEARS by entering gear data, tooth strength (Bending strength and Surface durability), profile generation, condition of engagement, Number of teeth and transfer torque.
Now to introduce the selected calculation formula for Bending strength and Surface durability from formula extracted from JGMA (Japan Gear Manufacture Association Standard) as follows.

Calculation formula of Bending strength for Spur and Helical gears JGMA 401-01 (1974).
Calculation formula of Surface durability for Spur and Helical gears JGMA 402-01 (1975).

1. Application range (common)

1.1 This standard is applied to Spur, Helical, Double helical and Internal gears that uses general industrial machinery transfer power.

Module : 1.5 to 25.0 mm
Reference pitch diameter : 25 to 3,200 mm
Circumferential velocity : Below 25m/s
Revolving velocity : Below 3,600 m^{-1}
Tooth profile of Spur and normal type of Helical gears as stipulated in JIS B 1701 (Involute tooth profile and dimensions). Also applicable to gears with Normal reference pressure angle of 22.5°and 25°
Accuracy : Accuracy classes 1 to 6 stipulated in JIS B 1702 (Accuracy for Spur and Helical gear).

1.2.1 This standard stipulates calculation for Bending allowable load and when determining designated dimension based on Tooth root bending stress.

1.2.2 This standard stipulates calculation for Tooth surface allowable load for a gear with designated dimension and for calculating specifications based on flank stress.

2 Definition

2.1 Bending strength
Bending allowable load for gear is Allowable tangential load on the Reference pitch circle based on Allowable tooth root bending stress of gears when transferring power during operation.

2.2 Surface durability
Surface durability is stipulated as capacity of load it can withstand and still provide necessary strength and enough safety for gear against progressive pitting. Therefore, meaning of Allowable flank load is Allowable tangential load on the Reference pitch circle determined in accordance to Surface durability of its gears when transferring power during operations.

3. Basic formula (common)

In regards to calculating Gear strength, the conversion formulas related to calculating Tangential load on Reference pitch circle, Nominal power and Nominal torque are as follows.

3.1 Nominal tangential load on Reference pitch circle F_t(kgf)

$$F_t = \frac{102P}{\upsilon} = \frac{1.95\times10^6 P}{dn} \quad \cdots\cdots(1)$$

Hereby
P : Nominal power (kW)
υ : Circumferential velocity (m/s) on the Reference pitch circle
d : Reference pitch diameter (mm)
n : Revolving velocity (min^{-1})

$$\upsilon = \frac{dn}{19100} \quad \cdots\cdots(2)$$

Or

$$F_t = \frac{2000T}{d} \quad \cdots\cdots(3)$$

Hereby
T : Nominal torque (kgf・m)

3.2 Nominal power (kW)

$$P = \frac{F_t\upsilon}{102} = \frac{10^{-6}}{1.95}F_t dn \quad \cdots\cdots(4)$$

3.3 Nominal torque (kgf・m)

$$T = \frac{F_t d}{2000} \quad \cdots\cdots (5)$$

Or

$$T = \frac{974P}{n} \quad \cdots\cdots (6)$$

4. Calculation formula for Strength

4.1 Bending strength

Nominal tangential load on the Reference pitch circle is necessary as reference for calculating Bending strength. Therefore, Nominal tangential load on the Reference pitch circle should be equal or below Allowable tangential load on the Reference pitch circle, which is derived from calculating Allowable tooth root bending stress. Therefore,

$$F_t \leqq F_{t\lim} \quad \cdots\cdots (7)$$

Hereby

F_t : Nominal tangential load on the Reference pitch circle (kgf)

$F_{t\lim}$: Calculate Allowable tangential load (kgf) on the Reference pitch circle by selecting the smaller value from either pinion or gear.

On the other hand, Tooth root stress calculated from Nominal tangential load on the Reference pitch circle should be equal or below Allowable tooth root bending stress.

Therefore

$$\sigma_F \leqq \sigma_{F\lim} \quad \cdots\cdots (8)$$

Hereby

σ_F : Dedendum stress calculated from Nominal tangential load on Reference pitch circle (kgf/mm^2)

$\sigma_{F\lim}$: Allowable tooth root bending stress (kgf/mm^2)

4.1.1 Calculation for Allowable tangential load on the Reference pitch circle is as follow.

$$F_{t\lim} = \sigma_{F\lim} \frac{m_n b}{Y_F Y_\varepsilon Y_\beta} \left(\frac{K_L K_{FX}}{K_V K_O} \right) \frac{1}{S_F} \quad \cdots\cdots (9)$$

Hereby

m_n : Normal module (mm)

b : Facewidth (mm)

Y_F : Form factor

Y_ε : Load distribution factor

Y_β : Helix angle factor

K_L : Life factor

K_{FX}: Dimension factor for Tooth root stress

K_v : Dynamic factor

K_O : Overload factor

S_F : Safety factor for Tooth root bending damage

4.1.2 Calculation for Tooth root bending stress is as follow.

$$\sigma_F = F_t \frac{Y_F Y_\varepsilon Y_\beta}{m_n b} \left(\frac{K_V K_O}{K_L K_{FX}} \right) S_F \quad \cdots\cdots (10)$$

4.2 Calculation for Surface durability

Nominal tangential load on the Reference pitch circle is necessary as reference for calculating Surface strength. Therefore, Nominal tangential load on the Reference pitch circle should be equal or below Allowable tangential load on the Reference pitch circle, which is derived from calculating Allowable Hertz stress. Therefore,

$$F_t \leqq F_{t\lim} \quad \cdots\cdots (11)$$

Hereby F_t : Nominal tangential load on the Reference pitch circle (kgf)

$F_{t\lim}$: Calculate Allowable tangential load (kgf) on the Reference pitch circle by selecting the smaller value (kgf) from either pinion or gear.

On the other hand, Hertz stress from Nominal tangential load on the Reference pitch circle should be equal or below Allowable hertz stress.

Therefore

$$\sigma_H \leqq \sigma_{H\lim} \quad \cdots\cdots (12)$$

Hereby

σ_H : Hertz stress calculated from Nominal tangential load on Reference pitch circle (kgf/mm^2)

$\sigma_{H\lim}$: Allowable hertz stress ((kgf/mm^2)

4.2.1 Calculation for Allowable tangential load on the Reference pitch circle is as follow.

$$F_{t\lim} = \sigma_{H\lim}{}^2 d_1 b_H \frac{u}{u \pm 1} \left(\frac{K_{HL} Z_L Z_R Z_V Z_W Z_{HX}}{Z_H Z_M Z_\varepsilon Z_\beta} \right)^2 \times \frac{1}{K_{H\beta} K_V K_O} \frac{1}{S_H{}^2} \quad \cdots\cdots (13)$$

+/- : '+' indicate the engagement with both External gears. '-' for engagement with External and Internal gears.

Hereby

d_1 : Reference pitch diameter for pinion (mm)

b_H : Effective facewidth for Surface durability (mm)

u : Gear ratio

Z_H : Zone factor

Z_M : Elasticity factor

Z_ε : Contact ratio factor

Z_β : Helix angle factor

K_{HL} : Life factor for Surface durability

Z_L : Lubricating oil factor

Z_R : Roughness factor

Z_V : Lubricating speed factor

Z_W : Work hardening factor

K_{HX}: Dimension factor for Surface durability

$K_{H\beta}$: Face load factor for Contact stress

K_V : Dynamic factor

K_O : Overload factor

S_H : Safety factor for Surface durability

4.2.2 Calculation for Hertz stress is as follows.

$$\sigma_H = \sqrt{\frac{F_t}{d_1 b_H} \frac{u \pm 1}{u}} \frac{Z_H Z_M Z_\varepsilon Z_\beta}{K_{HL} Z_L Z_R Z_V Z_W K_{HX}} \times \sqrt{K_{H\beta} K_V K_O}\, S_H \quad \cdots\cdots (14)$$

+/- : '+' indicate the engagement with both External gears. '-' for engagement with External and Internal gears.

5. Calculation formula for types of factor

5.1 How to obtain the types of factor using the calculation formula of Bending strength.

The following stipulates types of factor from calculation formula of Bending strength in previous paragraph.

5.1.1 Facewidth b

When Facewidths differs, assume wider Facewidth to be b_w and smaller Facewidth to be b_s. $b_w - b_s \leqq m_n$, use actual Facewidth for calculations.

When $b_w - b_s > m_n$, b_s is used in formula b_s + mn to calculation of Facewidth.

5.1.2 Form factor Y_F

Refer to Fig. 1 to find Form factor.

For Virtual number of teeth of spur gear for Helical gear, use following calculation formula.

$$z_v = \frac{z}{\cos^3\beta} \quad \cdots\cdots (15)$$

For Form factor for Tooth profile excluding Fig. 1 please refer to this original standard.

5.1.3 Load distribution factor Y_ε

Calculating Load distribution factor using following formula.

$$Y_\varepsilon = \frac{1}{\varepsilon_\alpha} \quad \cdots\cdots (16)$$

Hereby

ε_α : Transverse contact ratio

Calculation formulas of Transverse contact ratio are as follows,

Spur gear : $$\varepsilon_\alpha = \frac{\sqrt{r_{a1}^2 - r_{b1}^2} + \sqrt{r_{a2}^2 - r_{b2}^2} - a\sin\alpha_\omega}{m\pi\cos\alpha_0} \quad \cdots\cdots (17)$$

Helical gear : $$\varepsilon_\alpha = \frac{\sqrt{r_{a1}^2 - r_{b1}^2} + \sqrt{r_{a2}^2 - r_{b2}^2} - a\sin\alpha_{\omega t}}{m_t\pi\cos\alpha_t} \quad \cdots\cdots (18)$$

$$\cos^2\beta_b = 1 - \sin^2\beta \bullet \cos^2\alpha_n \quad \cdots\cdots (19)$$

Hereby

γ_a : Tip (Outside) radius (mm)

γ_b : Base radius (mm)

a : Centre distance (mm)

α_w : Working pressure angle (°)

α_{wt} : Transverse contact pressure angle (°)

α : Reference pressure angle (°)

α_n : Normal pressure angle (°)

α_t : Transverse reference pressure angle (°)

β : Reference pitch cylindrical helix angle (°)

β_b : Base cylinder helix angle (°)

Subscript

1 : Pinion

2 : Gear

Remark 1. Table 1 shows the Transverse contact ratio ε_α for Standard spur gear with Reference pressure angle 20°.

Remark 2. Use following formula to calculate approximate value of Y_ε for Helical gear.

$$\frac{\cos^2\beta_b}{\varepsilon_\alpha} = \frac{1}{\varepsilon_{\alpha n}} \quad \cdots\cdots (20)$$

However, obtain Transverse contact ratio $\varepsilon_{\alpha n}$ for Virtual spur gear from Table 1 by using Virtual number of teeth of spur gear z_{v1} and z_{v2}.

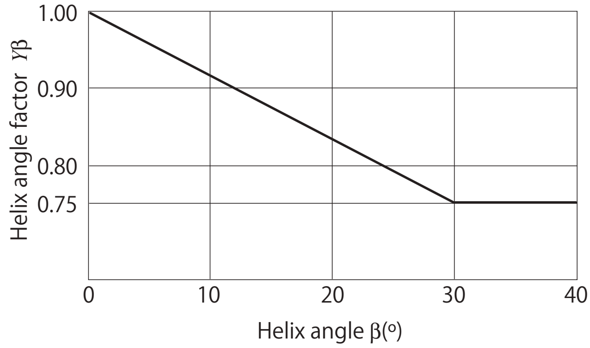
5.1.4 Helix angle factor Y_β

Calculate helix angle factor using following formula.

For $0° \leqq \beta \leqq 30°$: $Y_\beta = 1 - \frac{\beta}{120}$ ⋯⋯ (21)

For $\beta \geqq 30°\beta$: $Y_\beta = 0.75$ ⋯⋯ (22)

Fig. 2 Helix angle factor



5.1.5 Life factor K_L

Refer to Table 2 to obtain Life factor.

Table 2. Life factor K_L

Number of repeated	Hardness (1)(2) HB120 - 220	Hardness (2) Above HB221	Carburizing gear
Below 10,000	1.4	1.5	1.5
Approx. 100,000	1.2	1.4	1.5
Approx. 10^6	1.1	1.1	1.1
Above 10^7	1.0	1.0	1.0

Note (1) Steel casted gears to use this Table

Note (2) Core hardness is used for Induction hardened gear.

Meaning of repeated rotations is number of repeat during life span of gears. If uncertain, K_L =1.0.

Fig. 1 Graph for Form factor (Part 1 in Table 3)

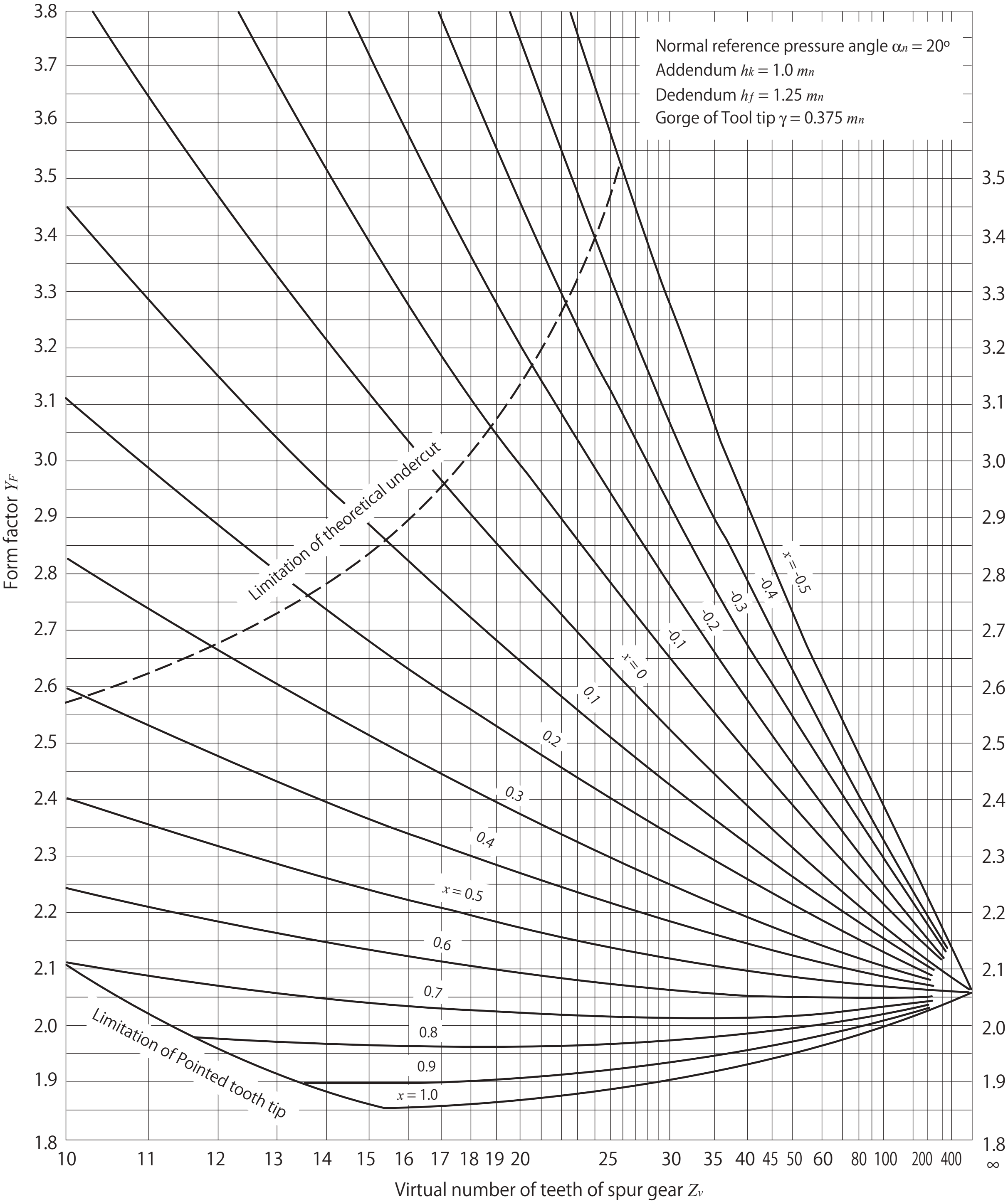
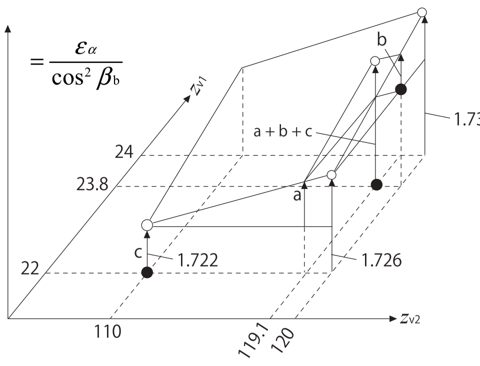


Table 1. Transverse contact ratio ε_α for Standard spur gear

$$\varepsilon_\alpha = \frac{\sqrt{r_{a1}^2 - r_{b1}^2} + \sqrt{r_{a2}^2 - r_{b2}^2} - a \bullet \sin\alpha_\omega}{m\pi\cos\alpha}$$

	17	18	19	20	21	22	24	25	26	28	30	32	34	36	38	40	42	45	48	50	52	55	58	60	65	70	75	80	90	100	110	120	130	140	150	160	170	180	190	200
17	1.515																																							
18	1.522	1.529																																						
19	1.530	1.537	1.544																																					
20	1.536	1.543	1.550	1.557																																				
21	1.542	1.549	1.556	1.563	1.569																																			
22	1.548	1.555	1.562	1.569	1.575	1.581																																		
24	1.558	1.566	1.573	1.579	1.586	1.591	1.602																																	
25	1.563	1.571	1.578	1.584	1.590	1.596	1.607	1.612																																
26	1.568	1.575	1.582	1.589	1.595	1.601	1.611	1.616	1.621																															
28	1.576	1.584	1.591	1.597	1.604	1.609	1.620	1.625	1.629	1.638																														
30	1.584	1.592	1.599	1.605	1.611	1.617	1.628	1.633	1.637	1.646	1.654																													
32	1.591	1.599	1.606	1.612	1.618	1.624	1.635	1.640	1.644	1.653	1.661	1.668																												
34	1.598	1.605	1.612	1.619	1.625	1.631	1.641	1.646	1.651	1.659	1.667	1.674	1.681																											
36	1.604	1.611	1.618	1.625	1.631	1.637	1.647	1.652	1.657	1.665	1.673	1.680	1.687	1.692																										
38	1.609	1.617	1.624	1.630	1.636	1.642	1.653	1.658	1.662	1.671	1.678	1.686	1.692	1.698	1.703																									
40	1.614	1.622	1.629	1.635	1.641	1.647	1.658	1.663	1.667	1.676	1.684	1.691	1.697	1.703	1.708	1.714																								
42	1.619	1.626	1.633	1.640	1.646	1.652	1.662	1.667	1.672	1.680	1.688	1.695	1.702	1.708	1.713	1.718	1.723																							
45	1.625	1.633	1.640	1.646	1.652	1.658	1.669	1.674	1.678	1.687	1.695	1.702	1.708	1.714	1.720	1.725	1.729	1.736																						
48	1.631	1.639	1.646	1.652	1.658	1.664	1.675	1.680	1.684	1.693	1.701	1.708	1.714	1.720	1.725	1.731	1.735	1.742	1.748																					
50	1.635	1.642	1.649	1.656	1.662	1.668	1.678	1.683	1.688	1.696	1.704	1.711	1.718	1.724	1.729	1.734	1.739	1.745	1.751	1.755																				
52	1.638	1.646	1.653	1.659	1.665	1.671	1.682	1.687	1.691	1.700	1.707	1.715	1.721	1.727	1.732	1.737	1.742	1.749	1.754	1.758	1.761																			
55	1.643	1.650	1.657	1.664	1.670	1.676	1.686	1.691	1.696	1.704	1.712	1.719	1.726	1.732	1.737	1.742	1.747	1.753	1.759	1.763	1.766	1.771																		
58	1.647	1.655	1.662	1.668	1.674	1.680	1.691	1.696	1.700	1.709	1.716	1.724	1.730	1.736	1.741	1.746	1.751	1.758	1.763	1.767	1.770	1.775	1.779																	
60	1.650	1.657	1.664	1.671	1.677	1.683	1.693	1.698	1.703	1.711	1.719	1.726	1.733	1.739	1.744	1.749	1.754	1.760	1.766	1.770	1.773	1.778	1.782	1.785																
65	1.656	1.663	1.670	1.677	1.683	1.689	1.699	1.704	1.709	1.717	1.725	1.732	1.739	1.745	1.750	1.755	1.760	1.766	1.772	1.776	1.779	1.784	1.788	1.791	1.797															
70	1.661	1.669	1.676	1.682	1.688	1.694	1.705	1.710	1.714	1.723	1.731	1.738	1.744	1.750	1.756	1.761	1.765	1.772	1.778	1.781	1.785	1.789	1.794	1.796	1.802	1.808														
75	1.666	1.673	1.680	1.687	1.693	1.699	1.710	1.714	1.719	1.728	1.735	1.742	1.749	1.755	1.760	1.765	1.770	1.777	1.782	1.786	1.789	1.794	1.798	1.801	1.807	1.812	1.817													
80	1.670	1.678	1.685	1.691	1.697	1.703	1.714	1.719	1.723	1.732	1.740	1.747	1.753	1.759	1.765	1.770	1.774	1.781	1.787	1.790	1.794	1.798	1.803	1.805	1.811	1.817	1.821	1.826												
90	1.678	1.685	1.692	1.699	1.705	1.711	1.721	1.726	1.731	1.739	1.747	1.754	1.760	1.766	1.772	1.777	1.782	1.788	1.794	1.798	1.801	1.806	1.810	1.813	1.819	1.824	1.829	1.833	1.840											
100	1.684	1.691	1.698	1.705	1.711	1.717	1.727	1.732	1.737	1.745	1.753	1.760	1.767	1.773	1.778	1.783	1.788	1.794	1.800	1.804	1.807	1.812	1.816	1.819	1.825	1.830	1.835	1.839	1.846	1.853										
110	1.689	1.696	1.703	1.710	1.716	1.722	1.732	1.737	1.742	1.750	1.758	1.765	1.772	1.778	1.783	1.788	1.793	1.799	1.805	1.809	1.812	1.817	1.821	1.824	1.830	1.835	1.840	1.844	1.852	1.858	1.863									
120	1.693	1.701	1.708	1.714	1.720	1.726	1.737	1.742	1.746	1.755	1.762	1.770	1.776	1.782	1.787	1.792	1.797	1.804	1.809	1.813	1.816	1.821	1.825	1.828	1.834	1.840	1.844	1.849	1.856	1.862	1.867	1.871								
130	1.697	1.704	1.711	1.718	1.724	1.730	1.740	1.745	1.750	1.758	1.766	1.773	1.780	1.786	1.791	1.796	1.801	1.807	1.813	1.817	1.820	1.825	1.829	1.832	1.838	1.843	1.848	1.852	1.860	1.866	1.871	1.875	1.879							
140	1.700	1.708	1.715	1.721	1.727	1.733	1.744	1.749	1.753	1.762	1.769	1.777	1.783	1.789	1.794	1.799	1.804	1.811	1.816	1.820	1.823	1.828	1.832	1.835	1.841	1.847	1.851	1.856	1.863	1.869	1.874	1.878	1.882	1.885						
150	1.703	1.710	1.717	1.724	1.730	1.736	1.747	1.751	1.756	1.765	1.772	1.779	1.786	1.792	1.797	1.802	1.807	1.813	1.819	1.823	1.826	1.831	1.835	1.838	1.844	1.849	1.854	1.858	1.866	1.872	1.877	1.881	1.885	1.888	1.891					
160	1.706	1.713	1.720	1.727	1.733	1.738	1.749	1.754	1.759	1.767	1.775	1.782	1.788	1.794	1.800	1.805	1.810	1.816	1.822	1.825	1.829	1.834	1.838	1.840	1.847	1.852	1.857	1.861	1.868	1.874	1.879	1.884	1.888	1.891	1.894	1.896				
170	1.708	1.715	1.722	1.729	1.735	1.741	1.751	1.756	1.761	1.769	1.777	1.784	1.791	1.797	1.802	1.807	1.812	1.818	1.824	1.828	1.831	1.836	1.840	1.843	1.849	1.854	1.859	1.863	1.871	1.877	1.882	1.886	1.890	1.893	1.896	1.898	1.901			
180	1.710	1.717	1.724	1.731	1.737	1.743	1.753	1.758	1.763	1.771	1.779	1.786	1.793	1.799	1.804	1.809	1.814	1.820	1.826	1.830	1.833	1.838	1.842	1.845	1.851	1.856	1.861	1.865	1.873	1.879	1.884	1.888	1.892	1.895	1.898	1.901	1.903	1.905		
190	1.712	1.719	1.726	1.733	1.739	1.745	1.755	1.760	1.765	1.773	1.781	1.788	1.795	1.800	1.806	1.811	1.816	1.822	1.828	1.832	1.835	1.840	1.844	1.847	1.853	1.858	1.863	1.867	1.874	1.881	1.886	1.890	1.894	1.897	1.900	1.902	1.905	1.907	1.909	
200	1.713	1.721	1.728	1.734	1.741	1.746	1.757	1.762	1.766	1.775	1.783	1.790	1.796	1.802	1.808	1.813	1.817	1.824	1.830	1.833	1.837	1.841	1.846	1.848	1.854	1.860	1.865	1.869	1.876	1.882	1.887	1.892	1.895	1.899	1.902	1.904	1.906	1.908	1.910	1.912

$$= \frac{\varepsilon_\alpha}{\cos^2\beta_b}$$



Calculation of transverse contact ratio for Helical gear

Example: Calculate $\varepsilon\alpha$ of Helical gear with $z_1 = 20, z_2 = 100, \beta = 19°22''$

Firstly calculate $Z_{v1} = 23.8$, $Z_{v2} = 119.1$ from formula (18), secondly calculate $\varepsilon\alpha = 1.722$ to compare between engagements of 22 and 110 where the difference is 0.004.
Obtain $\varepsilon\alpha = 1.726$ to compare between engagements of 22 and 120 where the difference is 0.004.
Obtain $\varepsilon\alpha = 1.737$ to compare between engagements of 24 and 120 where the difference is 0.011.

$$a = 0.004 \times \frac{119.1 - 110}{120 - 110} = 0.00364$$

$$b = 0.011 \times \frac{23.8 - 22}{24 - 22} = 0.0099$$

Transverse contact ratio $\varepsilon\alpha$ between $c = 22$ and 110 is 1.722

$$a + b + c = 1.73554$$

Calculate $\cos^2\beta b = 0.900286478$ from formula 21
Therefore,
$\varepsilon\alpha = 1.73554 \times 0.90286478 = 1.567$ (theoretical value is 1.568).

5.1.6 Dimension factor K_{FX} for Tooth root stress

With increased Tooth profile, Bending strength is influenced. At the moment, due to insufficient data Dimension factor will be 1.0.

5.1.7 Dynamic factor K_V

Obtain Dynamic factor from Table 3 using gear accuracy and Circumferential speed on the Reference pitch circle.

Table 3. Dynamic factor K_V

System of accuracy from JIS B 1702		Circumferential speed on the Reference pitch circle (m/s)						
Tooth profile		Below 1	Above 1.0 to below 3.0	Above 3.0 to below 5.0	Above 5.0 to below 8.0	Above 8.0 to below 12.0	Above 12.0 to below 18.0	Above 18.0 to below 25.0
Normal	Modified							
	1	-	-	1.0	1.0	1.1	1.2	1.3
1	2	-	1.0	1.05	1.1	1.2	1.3	1.5
2	3	1.0	1.1	1.15	1.2	1.3	1.5	-
3	4	1.0	1.2	1.3	1.4	1.5	-	-
4	-	1.0	1.3	1.4	1.5	-	-	-
5	-	1.1	1.4	1.5	-	-	-	-
6	-	1.2	1.5	-	-	-	-	-

5.1.8 Overload factor K_0

Obtain Overload factor using following formula.

$$K_O = \frac{\text{Actual tangential load}}{\text{Nominal tangential load } (F_t)} \quad \cdots\cdots\cdots (23)$$

Use Table 4 to obtain Actual tangential load if uncertain of value.

Table 4. Overload factor K_0

Impact from motor side	Impact from load		
	Flat load	Average impact	Heavy impact
Flat load (Electric, turbine, hydraulic motors)	1.0	1.25	1.75
Light impact (Multi cylinder engine)	1.25	1.5	2.0
Average impact(Single cylinder engine)	1.5	1.75	2.25

Note: If the impact from load is unknown, refer to Table 5.

5.1.9 Safety factor S_F for damage from Tooth root bending

Fixed value of Safety factor for damage from Tooth root bending is difficult to be determined due to various internal and external factors. Minimum factor of 1.2 is necessary.

5.1.10 Allowable Tooth root bending stress $\sigma_{F\text{lim}}$

Refer to Tables 9 and 10 for Allowable tooth root bending stress for gear with fixed load direction. For intermediate Hardness values in the tables shown, it is our recommendation to use interpolation values. When load direction is bi-directional, value of Allowable tooth root bending stress $\sigma_{F\text{lim}}$ will be 2/3 of values in the table. For exmple, an idler gear or gear which alternates bi-directionally and for equal loads on either right or left teeth.

Value of hardness or core hardness uses centre of Tooth root.

Table 5. Classification of load for Driven machine

Name of Driven machine	Range	Name of Driven machine	Range	Name of Driven machine	Range
Agitator	M	Elevator	U	Petroleum refinery machinery	M
Blower	U	Extruder	U	Paper mill machinery	M
Brewing and Distillation apparatus	U	Fan (electric fan)	U	Timber mill machinery	H
Vehicles	M	Fan (for industries)	M	Pump	M
Clarifier	U	Feeder	M	Rubber machinery (medium load)	M
Sorting Machine	M	Feeder (to and fro motion)	H	Rubber machinery (heavy load)	H
Ceramics industry machine (medium load)	M	Food machinery	M	Water treatment machine (light load)	U
Ceramics industry machine (heavy load)	H	Hammer mill	H	Water treatment machine (medium load)	M
Compressor	M	Hoist	M	Screen (fluid)	U
Conveyer (uniform load)	U	Machine tools (main drive)	M	Screen (gravel)	M
Conveyer (uniform load / heavy load)	M	Machine tools (supplementary drive)	U	Sugar plant machinery	M
Crane	U	Metalwork machinery	H	Textile machinery	M
Crusher	H	Rotary mill	M	Iron mill machinery (hot rolling)	H
Dredger (Medium load)	M	Tumbler	H	Iron mill machinery (cold rolling)	U
Dredger (heavy load)	H	Mixer	M		

Note U: Uniform load, M: Medium impact, H: Heavy impact

5.2 How to obtain each factor based on calculation for Surface durability

Factors using calculation formulas based on Surface durability as mentioned above is defined below.

5.2.1 Effective facewidth for Surface durability b_H (mm)

Obtain Effective facewidth for Surface durability from (a) and (b).

(a) For different Facewidth between pinion and gear, select the narrower Facewidth as Effective facewidth.

(b) For Facewidth with end relief at both ends, Effective facewidth is the narrower of the Facewidth deducted by such end relief areas.

5.2.2 Zone factor Z_H

Calculation of Zone factor is as follows.

$$Z_H = \sqrt{\frac{2\cos\beta_b\cos\alpha_{wt}}{\cos^2\alpha_t\sin\alpha_{wt}}} = \frac{1}{\cos\alpha_t}\sqrt{\frac{2\cos\beta_b}{\tan\alpha_{wt}}} \quad \cdots\cdots\cdots\cdots (24)$$

Hereby

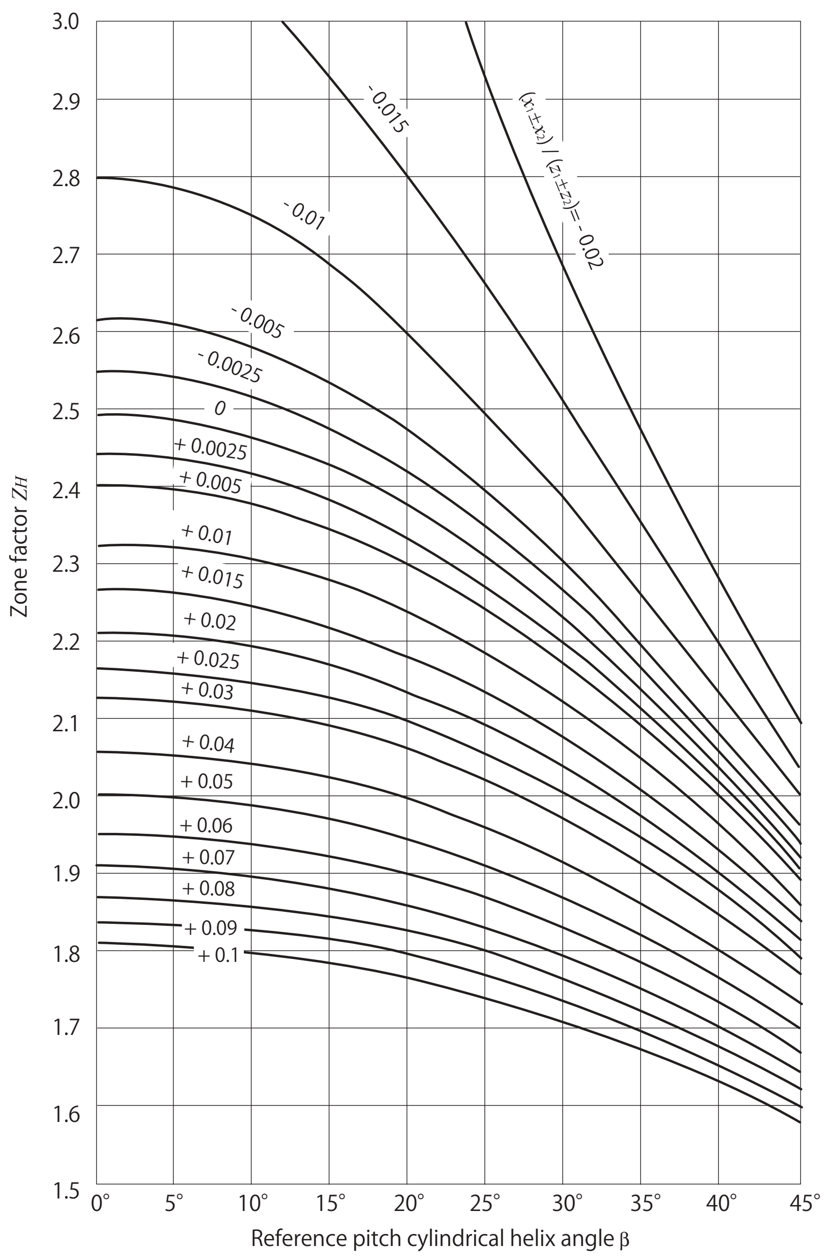
β_b : Base cylinder helix angle (°)

α_{wt} : Transverse contact pressure angle (°)

α_t : Transverse reference pressure angle (°)

(a) Obtain Zone factor from Fig. 3 with Normal reference pressure angle of 20° defined in JIS.

Fig. 3 Zone factor Z_H (Normal reference pressure angle α_n = 20°)



In Fig. 3, x : Rack shift coefficient (Normal rack shift coefficient for Helical gear and Super-script) 1 is Pinion and 2 is Gear.)

z : Number of teeth

β : Reference pitch cylindrical helix angle (°)

(b) Factors from above formula and figure are defined as follows.

$$\beta_b = \tan^{-1}(\tan\beta \cos\alpha_t) \quad \cdots\cdots (25)$$

$$\mathrm{inv}\,\alpha_{wt} = 2\tan\alpha_n\left(\frac{x_1 \pm x_2}{z_1 \pm z_2}\right) + \mathrm{inv}\,\alpha_t \quad \cdots\cdots (26)$$

$$\alpha_t = \tan^{-1}(\tan\alpha_n / \cos\beta) \quad \cdots\cdots (27)$$

(c) Zone factor is based upon Curvature radius of flank at Pitch point. Therefore this factor is used for calculating Allowable load for flank. Due to Relative curvature radius at the worst load point is slightly smaller than that at Pitch point, such Zone factor cannot be use. These are Spur gear or Helical gear with extremely small Overlap ratio ($\varepsilon_\beta <$ about 0.5) with below minimum number of teeth ($z \leqq$ about 23) and small Rack shift coefficient. For such cases, please refer to 4.2.2 to check for Hertz stress at the worst load point.

5.2.3 Elasticity factor Z_M

Calculation of Elasticity factor is as follows.

$$Z_M = \sqrt{\frac{1}{\pi\left(\frac{1-\nu_1^2}{E_1} + \frac{1-\nu 2^2}{E_2}\right)}} \quad \cdots\cdots (28)$$

Hereby

ν : Poisson's ratio

E : Modulus of direct elasticity (Young's modulus) (kgf/mm²)

For Z_M, refer to Table 6 for combinations of main gear materials.

5.2.4 Contact ratio factor Z_ε

Obtain Contact ratio factor using following formula (refer to Fig 4).

Spur gear : $Z_\varepsilon = 1.0 \quad \cdots\cdots (29)$

Fig. 4 Contact ratio factor

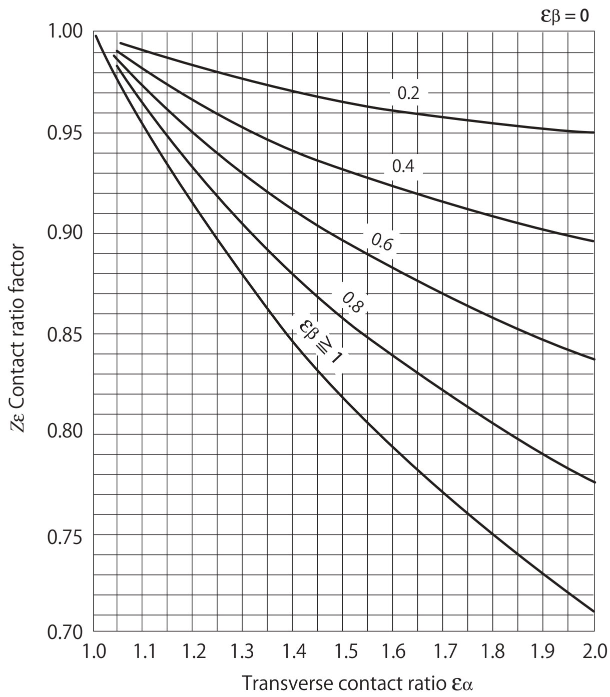


Table 6. Elasticity factor Z_M

Gear Materials	Gear Vocabularies	Gear Modulus of direct elasticity E kgf/mm²	Gear Poisson's ratio ν	Mating gear Materials	Mating gear Vocabularies	Mating gear Modulus of direct elasticity E kgf/mm²	Mating gear Poisson's ratio ν	Elasticity factor Z_M (kgf/mm²)$^{0.5}$
Structural steel	*(1)	21000	0.3	Structural steel	*(1)	21000	0.3	60.6
				Casting steel	SC	20500		60.2
				Spheroidal graphite iron	FCD	17600		57.9
				Gray iron casting	FC	12000		51.7
Casting steel	SC	20500		Casting steel	SC	20500		59.9
				Spheroidal graphite iron	FCD	17600		57.6
				Gray iron casting	FC	12000		51.5
Spheroidal graphite iron	FCD	17600		Spheroidal graphite iron	FCD	17600		55.5
				Gray iron	FC	12000		50.0
Gray iron casting	FC	12000		Gray iron casting	FC	12000		45.8

Note(1) *Structural steel to be S ～ C, SNC, SNCM, SCr, SCM.

Helical gear : in case $Z_\varepsilon = \sqrt{1-\varepsilon_\beta+\dfrac{\varepsilon_\beta}{\varepsilon_\alpha}}$ $\varepsilon_\beta \leqq 1$ ········ (30)

: in case $Z_\varepsilon = \sqrt{\dfrac{1}{\varepsilon_\alpha}}$ $\varepsilon_\beta > 1$ ····················· (31)

Hereby

ε_α : Transverse contact ratio (refer to Clause 5.1.3 and Reference 1)

ε_β : Overlap ratio

5.2.5 Helix angle factor Z_β

Helix angle factor for Surface durability is difficult to accurately stipulate due to insufficient data. Calculation formula will be

$Z_\beta = 1.0$ ·· (32)

5.2.6 Life factor for Surface durability K_{HL}

Obtain Life factor for Surface durability from Table 7

Table 7. Life factor of Surface durability

Number of repeated	Life factor for Surface durability
Below 10,000	1.5
About 100,000	1.3
About 10^6	1.15
Above 10^7	1.0

Remark 1. 'Repeated' is number of times of engaged rotation during life span.

Remark 2. Normally idler gear makes 2 engagements per rotation. However for engagements between different flanks for 1 rotation, it should be counted as 1 engagement.

Remark 3. For reversible rotation or similar conditions, number of rotation is from larger load applied to either flank.

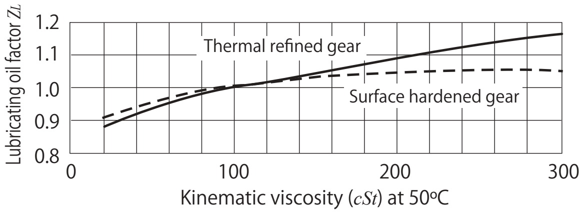
If number of times is uncountable, life factor to be

$K_{HL} = 1.0$ ·· (33)

5.2.7 Lubricating oil factor Z_L

For the 2 types of gear stated below, obtain Lubricating oil factor from Fig. 5 based on Kinematic viscosity (cSt) at 50°C.

Fig. 5 Lubricating oil factor



(1) Thermal refined gear [1]: Use solid line in Fig. 5.

(2) Surface hardened gear: Use broken line in Fig. 5.

Note (1) Thermal refined gear includes gear with quenching, tempering and normalizing.

Remark: Casting steel gear is equivalent to thermal refined gear.

5.2.8 Roughness factor Z_R

Find Roughness factor based on average roughness of flank $R_{maxm}(\mu m)$ from Fig. 6 for 2 types of gears. Use the following formula to obtain the average of maximum height of profile roughness of flank R_{maxm} from R_{max1}, R_{max2} and centre distance a(mm). (Meaning of R_{max1}, R_{max2} is Maximum height if profile roughness of flank inclusive of the effects of warm up and test run.)

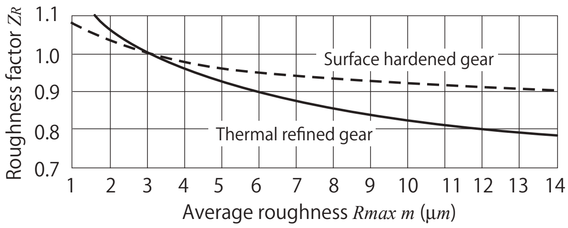
$$R_{maxm} = \frac{R_{max1}+R_{max2}}{2}\sqrt[3]{\frac{100}{a}}(\mu m) \quad \cdots\cdots (34)$$

(1) Thermal refined gear [1]: Use solid line in Fig. 6.

(2) Surface hardened gear: Use broken line in Fig. 6.

Refer to 5.2.7 for Note (1) and Remark

Fig. 6 Roughness factor



5.2.9 Lubricating speed factor Z_V

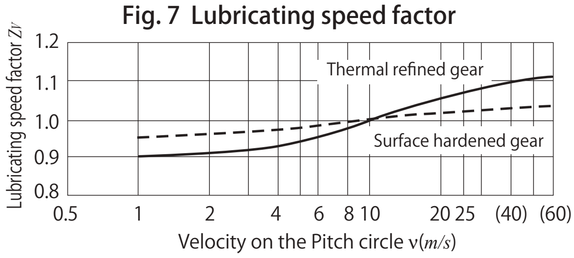
Find Lubricating speed factor based on maximum height of profile roughness of flank $R_{maxm}(\mu m)$ from Fig. 7 using either pinion or gears

(1) Thermal refined gear (1): Use solid line in Fig. 7.

(2) Surface hardened gear: Use broken line in Fig. 7.

Refer to 5.2.7 for Note (1) and Remark

Fig. 7 Lubricating speed factor



5.2.10 Work hardening factor Z_W

Hardness ratio factor is applied to engagement between gear and pinion(1) which is hardened ground. Calculation for Work hardening factor Z_w is as follow. (Refer to Fig. 8)

$$Z_W = 1.2 - \frac{HB_2 - 130}{1700} \quad \cdots\cdots (35)$$

Hereby

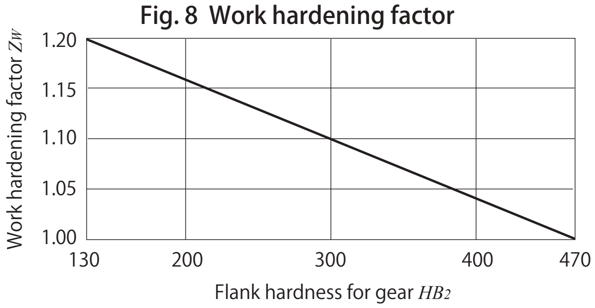
HB_2 : Hardness of gear flank (indicated by Brinell hardness)

However

Gear with conditions that cannot match above (35) and $130 \leqq HB_2 \leqq 470$, Pinion to be

$Z_W = 1.0$ ·· (36)

Fig. 8 Work hardening factor



Note (1) Flank roughness of pinion is $R_{MAX1} \leqq 6\mu$m when engaged with stipulated gear.

5.2.11 Dimension factor K_{HX} for Surface durability

If Tooth profile and gear size increases, Surface durability also increases but has a tendency to increase disproportionately. Due to insufficient data at the moment Dimension factor

$$K_{HX} = 1.0 \quad \cdots\cdots (37)$$

5.2.12 Face load factor for contact stress $K_{H\beta}$

Obtain Face load factor for contact stress for Surface durability using following formula.

(a) If unable to estimate tooth contact conditions when load is applied to gear. Obtain Tooth trace load distribution factor from ratio (b/d_1) between Facewidth b and Reference diameter d_1 of pinion and from method of gear support from Table 8.

(b) Satisfactory tooth contact when load is applied to gear.

Tooth trace load distribution factor $K_{H\beta}$ for Surface durability depends on level of modification compared to used load (reference value). When calculating modifications on Tooth trace for following cases, analyse all causes that influence Tooth bearing when load is applied. Apply modifications of Proper Tooth trace for gear, Helix angle, Axial parallelism. Warm up and test run is performed and confirm Tooth bearing is secured during operation.

$$K_{H\beta}=1.0 \sim 1.2 \quad \cdots\cdots (38)$$

5.2.13 Dynamic factor K_V (common)

Obtain Dynamic factor based on gear accuracy and Circumferential speed on the Reference pitch circle from 5.1.7 of Table 3.

5.2.14 Overload factor K_o (common)

Obtain the overload factor from 5.1.8 - Table 4.

5.2.15 Safety factor for flank damage (Pitting) S_H

A minimum Safety factor for flank damage (Pitting) value of 1.15 is necessary even though it is difficult to find fixed value of internal and external factors.

Table 8. Face load factor for contact stress

$\frac{b}{d_1}$	Supporting method			
	Support on both end			Unbalanced support
	Balanced to both bearings	Bearing is on one side and stiffness of axis is increased.	Bearing is on one side and less stiffness of axis.	
0.2	1.0	1.0	1.1	1.2
0.4	1.0	1.1	1.3	1.45
0.6	1.05	1.2	1.5	1.65
0.8	1.1	1.3	1.7	1.85
1.0	1.2	1.45	1.85	2.0
1.2	1.3	1.6	2.0	2.15
1.4	1.4	1.8	2.1	-
1.6	1.5	2.05	2.2	-
1.8	1.8	-	-	-
2.0	2.1	-	-	-

Remark 1. b is Effective facewidth for Spur and Helical gears. For Double helical gear, b is length of facewidth inclusive of cutter groove at centre of gear.

Remark 2. Tooth contact has to be satisfactory without load.

Remark 3. Inapplicable to Idler gear and pinion (Idler) engaged with gears.

5.2.16 Allowable hertz stress $\sigma_{H\lim}$

Refer to Tables 9 ～ 12 to find the Allowable hertz stress. For values not listed, use interpolation. Meaning of flank's hardness is hardness near Pitch circle.

Table 9. Gear without surface hardening

Materials (Arrow marks are for references only)		Hardness of flank HB	Hardness of flank HV	Lower limit of tensile strength kgf/mm² (reference)	σFlim kgf/mm²	σHlim kgf/mm²
Casting steel	SC37 SC42 SC46 SC49 SCC3			37	10.4	34
				42	12.0	35
				46	13.2	36
				49	14.2	37
				55	15.8	39
				60	17.2	40
Carbon steel for structural use with Normalizing	S25C S35C S43C S48C S53C S58C	120	126	39	13.8	41.5
		130	136	42	14.8	42.5
		140	147	45	15.8	44
		150	157	48	16.5	45
		160	167	51	17.6	46.5
		170	178	55	18.4	47.5
		180	189	58	19.0	49
		190	200	61	19.5	50
		200	210	64	20	51.5
		210	221	68	20.5	52.5
		220	231	71	21	54
		230	242	74	21.5	55
		240	252	77	22	56.5
		250	263	81	22.5	57.5
Carbon steel for structural use with Quenching and Tempering	S35C S43C S48C S53C S58C	160	167	51	18.2	51
		170	178	55	19.4	52.5
		180	189	58	20.2	54
		190	200	61	21	55.5
		200	210	64	22	57
		210	221	68	23	58.5
		220	231	71	23.5	60
		230	242	74	24	61
		240	252	77	24.5	62,5
		250	263	81	25	64
		260	273	84	25.5	65.5
		270	284	87	26	67
		280	295	90	26	68.5
		290	305	93	26.5	70
		300	316	97		71
		310	327	100		72.5
		320	337	103		74
		330	347	106		75.5
		340	358	110		77
		350	369	113		78.5
Alloy steel for structural use with Carburizing, Quenching and Tempering	SMn443 SNC836 SCM435 SCM440 SNCM439	220	231	71	25	70
		230	242	74	26	71.5
		240	252	77	27.5	73
		250	263	81	28.5	74.5
		260	273	84	29.5	76
		270	284	87	31	77.5
		280	295	90	32	79
		290	305	93	33	81
		300	316	97	34	82.5
		310	327	100	35	84
		320	337	103	36.5	85.5
		330	347	106	37.5	87
		340	358	110	39	88.5
		350	369	113	40	90
		360	380	117	41	92
		370	391	121		93.5
		380	402	126		95
		390	413	130		96.5
		400	424	135		98

Table 9. Gear with High frequency induction hardening (continued)

Materials (Arrow marks are for references only)		Conditions of Heat treatment before High-frequency induction hardening	Core hardness HB	Core hardness HV	Flank hardness [1] HV	$\sigma F\text{lim}$[2] kgf/mm²	$\sigma H\text{lim}$ kgf/mm²
Carbon steel for structural use	S43C, S48C	Normalizing	160	167	Above 550	21	
			180	189	〃	21	
			220	231	〃	21.5	
			240	252	〃	22	
	S48C, S43C	Induction hardening and Tempering	200	210	Above 550	23	
			210	221	〃	23.5	
			220	231	〃	24	
			230	242	〃	24.5	
			240	252	〃	25	
			250	263	〃	25	
Alloy steel for structural use	SMn443, SCM440, SNCM439, SCM435, SNC836	Induction hardening and tempering	230	242	Above 550	27	
			240	252	〃	28	
			250	263	〃	29	
			260	273	〃	30	
			270	284	〃	31	
			280	295	〃	32	
			290	305	〃	33	
			300	316	〃	34	
			310	327	〃	35	
			320	337	〃	36.5	
Carbon steel for structural use	S43C S48C	Normalizing			420		77
					440		80
					460		82
					480		85
					500		87
					520		90
					540		92
					560		93.5
					580		95
					Above 600		96
		Induction hardening and Tempering			500		96
					520		99
					540		101
					560		103
					580		105
					600		106.5
					620		107.5
					640		108.5
					660		109
					Above 680		109.5
Alloy steel for structural use	SMn443 SCM435 SCM440 SNC836 SNCM439	Induction hardening and Tempering			500		109
					520		112
					540		115
					560		117
					580		119
					600		121
					620		123
					640		124
					660		125
					Above 680		126

Note(1) When flank hardness is low, use $\sigma F\text{lim}$ value which is equivalent to gear without hardened surface.

Note(2) When gear has defects such as quenching cracks, insufficient hardening depth and uneven hardness, precaution is necessary as values of $\sigma F\text{lim}$ may become significantly lower compared with Tables 9 and 10. Values in Tables 9 and 10 are shown for full quenching at bottomland. Assuming insufficient quenching at bottomland, value will be 75% from Table 9 and 10.

Table 10. Gear with case hardening

Materials (Arrow marks are references only)		Effective carburizing depth [2]	Core hardness [1]		Flank hardness	$\sigma_{F\lim}$ [2]	$\sigma_{H\lim}$
			HB	HV	HV	kgf/mm^2	kgf/mm^2
Carbon steel for machine structural use	S15C S15CK		140	147		18.2	
			150	157		19.6	
			160	167		21	
			170	178		22	
			180	189		23	
			190	200		24	
		Relatively shallow depth (A)			580		115
					600		117
					620		118
					640		119
					660		120
					680		120
					700		120
					720		119
					740		118
					760		117
					780		115
					800		113
Alloy steel for machine structural use	SCM415 SCM415 SCM420 SNC815 SNCM420		220	231		34	
			230	242		36	
			240	252		38	
			250	263		39	
			260	273		41	
			270	284		42.5	
			280	295		44	
			290	305		45	
			300	316		46	
			310	327		47	
			320	337		48	
			330	347		49	
			340	358		50	
			350	369		51	
			360	380		51.5	
			370	390		52	
	SCM415(21) SCM420(22) SNC415(21) SNC815(22) SNCM420(23)	Relatively shallow depth (B)			580		131
					600		134
					620		137
					640		138
					660		138
					680		138
					700		138
					720		137
					740		136
					760		134
					780		132
					800		130
		Relatively deeper than above B			580		156
					600		160
					620		164
					640		166
					660		166
					680		166
					700		164
					720		161
					740		158
					760		154
					780		150
					800		146

Note (1) Relatively shallow effective case depth refers to below A and relatively deeper depth refers to B or more. Meaning of Effective case depth is hardness of up to HV513 (HRC50). Depth for Ground gear is after process.

Module		1.5	2	3	4	5	6	8	10	15	20	25
Effective depth	A	0.2	0.2	0.3	0.4	0.5	0.6	0.7	0.9	1.2	1.5	1.8
	B	0.3	0.3	0.5	0.7	0.8	0.9	1.1	1.4	2.0	2.5	3.4

Remark: Especially in engagement between gears, we recommend providing bigger Safety factor S_H., starting point of Maximum inner shear-stress force at inner gear tooth from surface pressure of flank is deeper than the depth of Case hardening which affects effectiveness of Carburizing depth.

Table 11. Nitriding gear [1]

Material		Flank hardness (reference)	σHlim kgf/mm²	
Nitriding steel	SACM 645 and others	Above HV 650	Normal	120
			Sustained period of Nitriding treatment	130 - 140

Note (1) Applicable to gear with proper Nitriding depth and hardened surface for improving Surface durability. We recommend providing a larger safety factor than usual when Surface hardness is lower than above table. Starting point of Maximum shear-stress force at inner gear tooth is deeper than depth of Nitriding.

Table 12. Nitro-carburizing [1]

Material	Nitriding period (h)	σHlim kgf/mm²		
		Relative curvature radius (mm)[2]		
		Below 10	10 - 20	Above 20
Carbon steel and Alloy steel for structural use	2	100	90	80
	4	110	100	90
	6	120	110	100

Note (1) Applicable to Salt bath and Gas Nitro-carburizing gears.

(2) Use Fig. 9 to obtain Relative curvature radius

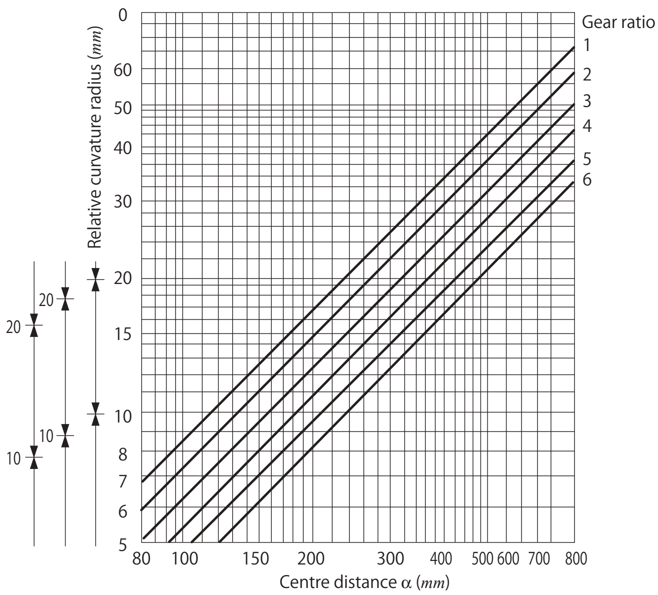
Remark. Use properly adjusted gear material for core.

Table 13. Nitriding gear [1]

Material	Flank hardness (reference)	Core hardness		σFlim
		HB	HV	kgf/mm²
Alloy steel for structural use without Nitriding steel	Above HV650	220	231	30
		240	252	33
		260	273	36
		280	295	38
		300	316	40
		320	337	42
		340	358	44
		360	380	46
Nitriding steel SACM645	Above HV650	220	231	32
		240	252	35
		260	273	38
		280	295	41
		300	316	44

Note (1) Applicable to gear with proper Nitriding depth for improving Surface durability. However Nitriding layer is extremely thin from Nitro-carburizing, use $\sigma_{f\,\mathrm{lim}}$ value of the gear without hardened surface.

Fig. 9 Relative curvature radius



10.2 Calculation for Bevel gear strength

Calculation formula of Bending strength for Bevel gear JGMA 403-01 (1976)
Calculation formula of Surface durability (Pitting resistance) for Bevel gear JGMA 404-01 (1977)

1. Application range (common)

1.1 This standard applies to Bevel gears (1) for power transfer used in the general industrial machinery with the following range.

Outer transverse module	: 1.5 ～ 25 mm
Outer pitch diameter	: Below 1,600 mm (For Straight bevel gear) Below 1,000 mm (For Spiral bevel gear)
Outer circumferential velocity	: Below 25 m/s
Revolving velocity	: Below 3,600 min^{-1}
Shaft angle	: 90°
Mean spiral angle	: Below 35°

Facewidth
For Maximum Facewidth, choose the smaller value from either 0.3 times of Cone distance or 10 times of Outer transverse module. However for Zerol® Bevel gear, it is 0.25 times of Outer cone distance.
Ⓡ mark is Gleason Works Trademark.

Tooth profile
Normal reference pressure angles are 20°, 22.5° and 25°.

Accuracy
Accuracy of Bevel gear is defined in JIS B1704 class 1 to 6.

Note (1) This standard is for Straight, Spiral and Zerol bevel gears.

1.2.1. Use this standard for calculation of Bending of Bevel gear for Allowable load as defined above in 1.1 and to determine gear dimensions based on Tooth root bending stress.

1.2.2 This standard used for calculation of tooth flank of allowable load for Straight, Spiral bevel gears and determines gear dimension based on Hertz stress of tooth flank.

2 Definition

2.1 Bending strength
Bending allowable load of Bevel gear is stipulated as Nominal allowable tangential load on the Mean pitch circle based on Allowable tooth root bending stress for each gear when transferring power during operation.

2.2 Surface durability
Surface durability of Bevel gear is stipulated as load capacity that is necessary to provide sufficient safety to the gear against progressive pitting.
Therefore, Allowable load on Bevel gear flank is stipulated as Allowable tangential load on the Mean pitch circle based on Surface durability for each gear when transferring power during operation.

3. Basic formula

For calculating gear strength, conversion formulas are related to calculating Nominal tangential load on the Reference pitch circle. Nominal power and torque are as follows.

3.1 Nominal tangential load on the Mean pitch circle F_{tm}(kgf)

$$F_{tm}=\frac{102P}{v_m}=\frac{1.95\times10^6 P}{d_m n} \quad \cdots\cdots(1)$$

Hereby
- P : Nominal power (kW)
- v_m : Circumferential velocity (m/s) on the Mean pitch circle
- d_m : Mean pitch diameter (mm)
- n : Revolving velocity (min^{-1})

$$v_m=\frac{d_m n}{19100} \quad \cdots\cdots(2)$$

$$d_m=d-b\sin\delta \quad \cdots\cdots(3)$$

Hereby
- d : Pitch diameter (mm)
- δ : Pitch angle (°)

Or $F_{tm}=\dfrac{2000T}{d_m}$ $\cdots\cdots(4)$

Hereby
- T : Nominal torque (kgf・m)

3.2 Nominal power P (kW)

$$P=\frac{F_{tm}v_m}{102}=5.13\times10^{-7}F_{tm}d_m n \quad \cdots\cdots(5)$$

3.3 Nominal torque T (kg・m)

$$T=\frac{F_{tm}d_m}{2000} \quad \cdots\cdots (6)$$

Or $$T=\frac{974P}{n} \quad \cdots\cdots (7)$$

4. Calculation formula for gear strength

4.1 Calculation for Bending strength

When calculating Bending strength, use Nominal tangential load on the Mean pitch circle as reference. Therefore Nominal tangential load on the Mean pitch circle should be equal or less than Allowable tangential load on the Mean pitch circle calculated by Allowable tooth root stress. That is to say,

$$F_{tm} \leqq F_{tm\lim} \quad \cdots\cdots (8)$$

Hereby

F_{tm} : Nominal tangential load on the Mean pitch circle (kgf)

$F_{tm\lim}$: Nominal allowable tangential load (kgf) on the Mean pitch circle is selected from its smaller value from either pinion or gear.

On the other hand, Tooth root stress obtained from Nominal tangential load on the Mean pitch circle should be equal or lesser than Allowable Tooth root bending stress.

Therefore

$$\sigma_F \leqq \sigma_{F\lim} \quad \cdots\cdots (9)$$

Hereby

σ_F : Tooth root stress (kgf/mm^2) from Nominal tangential load on the Mean pitch circle.

$\sigma_{F\lim}$: Allowable Tooth root bending stress (kgf/mm^2)

4.1.1 Calculation for Allowable tangential load on the Mean pitch circle is as follow.

$$F_{tm\lim}=0.85\cos\beta_m\sigma_{F\lim}mb\frac{R_e-0.5b}{R_e}\frac{1}{Y_FY_\varepsilon Y_\beta Y_C}\times\left(\frac{K_LK_{FX}}{K_MK_VK_O}\right)\frac{1}{K_R} \quad \cdots\cdots (10)$$

Hereby

β_m : Mean spiral angle (°)

m : Outer transverse module (mm)

b : Facewidth (mm)

R_e : Cone distance (mm)

Y_F : Form factor

Y_ε : Load distribution factor

Y_β : Spiral angle factor

Y_c : Cutter diameter influence factor

K_L : Life factor

K_{FX}: Dimension factor for Tooth root stress

K_M : Load distributed factor for Tooth trace

K_v : Dynamic factor

K_O : Overload factor

K_R : Reliability factor for Tooth root bending damage

4.1.2 Calculation for Tooth root bending stress is as follow.

$$\sigma_F=F_{tm}\frac{Y_FY_\varepsilon Y_\beta Y_C}{0.85\cos\beta_m mb}\frac{R_e}{R_e-0.5b}\left(\frac{K_MK_VK_O}{K_LK_{FX}}\right)K_R \quad \cdots\cdots (11)$$

4.2 Calculation for Tooth root strength

Nominal tangential load on the Mean pitch circle is necessary as reference for calculating Surface strength. Therefore, Nominal tangential load on the Mean pitch circle should be equal or below Allowable tangential load on the Mean pitch circle, which is derived from calculating Allowable Hertz stress. Therefore,

$$F_{tm} \leqq F_{tm\lim} \quad \cdots\cdots (12)$$

Hereby

F_{tm} : Nominal tangential load on the Mean pitch circle (kgf)

$F_{tm\lim}$: Calculate Allowable tangential load (kgf) on the Mean pitch circle by selecting the smaller Allowable tangential load (kgf) from either pinion or gear.

On the other hand, Hertz stress based on Nominal tangential load on the Mean pitch circle should be equal or less than Allowable hertz stress.

Therefore

$$\sigma_H \leqq \sigma_{H\lim} \quad \cdots\cdots (13)$$

Hereby

σ_H : Hertz stress (kgf/mm^2) from Nominal tangential load on the Mean pitch circle

$\sigma_{H\lim}$: Allowable hertz stress (kgf/mm^2)

4.2.1 Calculation for Allowable tangential load on the Mean pitch circle is as follow.

$$F_{tm\lim}=\left(\frac{\sigma_{H\lim}}{Z_M}\right)^2\frac{d_1}{\cos\delta_1}\frac{R_e-0.5b}{R_e}\bullet b\frac{u^2}{u^2+1}\left(\frac{K_{HL}Z_LZ_RZ_VZ_WK_{HX}}{Z_HZ_\varepsilon Z_\beta}\right)^2\frac{1}{K_{H\beta}K_VK_O}\frac{1}{C_R^2} \quad \cdots\cdots (14)$$

Hereby

d_1 : Outer pitch diameter for pinion (mm)

b : Facewidth (mm)

u : Gear ratio

R_e : Cone distance (mm)

Z_H : Zone factor

Z_M : Elasticity factor

Z_ε : Contact ratio factor

Z_β : Spiral angle factor for Surface durability

K_{HL} : Life factor for Surface Durability

Z_L : Lubricating oil factor

Z_R : Roughness factor

Z_V : Lubricating speed factor

Z_W : Work hardening factor

Z_{HX} : Dimension factor for Surface durability

$K_{H\beta}$: Face load for contact stress for Surface durability

K_V : Dynamic factor

K_O : Overload factor

C_R : Reliability factor for Surface durability

4.2.2 Calculation for Hertz stress is as follow.

$$\sigma_H = \sqrt{\frac{\cos\delta_1 F_{tm}}{d_1 b}\frac{u^2+1}{u^2}\frac{R_e}{R_e-0.5b}}\frac{Z_H Z_M Z_\varepsilon Z_\beta}{K_{HL} Z_L Z_R Z_V Z_W K_{HX}} \times$$

$$\sqrt{K_{H\beta} K_V K_O}\, C_R \quad \cdots\cdots (15)$$

5 Calculation method for factors

5.1 Calculation method for factors based on Bending (tooth root) strength of Bevel gear.

Factors used in calculation formulas for Bending (tooth root) strength as mentioned above are stipulated as follows.

5.1.1 Facewidth b

Facewidth b is stipulated as Facewidth on Pitch cone. For different Facewidth, use narrower side from either pinion or gear as Effective facewidth.

5.1.2 Form Y_F

Obtain Form factor from Fig. 1 and 2.

(a) Refer to Table 1, items 5 and 6 where Normal reference pressure angle is 20°.

Use Form factor graphs in Fig. 2 and 3 to obtain primary value of Y_{FO} (Value of Form factor by Rack shift). Then obtain Revision factor C using Horizontal rack shift from Fig. 1.

$$Y_F = C Y_{F0} \quad \cdots\cdots (16)$$

Calculate Y_F from formula $Y_F=CF_{Y0}$. However, Tooth profile with no Horizontal rack shift to be $Y_F=Y_{F0}$.

a.1 Refer to Table 1 for lists of Form factor chart.

Calculate Virtual number of teeth of spur gear Z_v and Rack shift coefficient x using following formula.

$$z_v = \frac{z}{\cos\delta \cos^3\beta_m} \quad \cdots\cdots (17)$$

Hereby

δ : Pitch angle (°)

$$x = \frac{h_a - h_{a0}}{m} \quad \cdots\cdots (18)$$

Hereby

h_a : Outer addendum (mm)

h_{ao} : Refer to Table 1 for Reference profile addendum (mm)

m : Outer transverse module (mm)

a. 2. For Bevel gear with tip of cutter with γ about 0.375 mm, constant 0.85 to be changed to 1.0 in the formulas for Allowable tangential load and Bending stress. (Refer to 4.1.1 of standard $\sigma_{F\lim}$).

a. 3. Calculate Horizontal rack shift coefficient K in Fig. 1 using the following formula.

$$K = \frac{1}{m}\left\{s - 0.5\pi m - \frac{2(h_a - h_{a0})\tan\alpha_n}{\cos\beta_m}\right\} \quad \cdots\cdots (19)$$

Hereby

s : Outer transverse circular thickness (mm)

h_a, h_{ao} and m : Same as formula (14).

However the above formula for K is inapplicable for an Isothermal full depth gear tooth.

Fig. 1 Revision factor base on Horizontal Rack shift

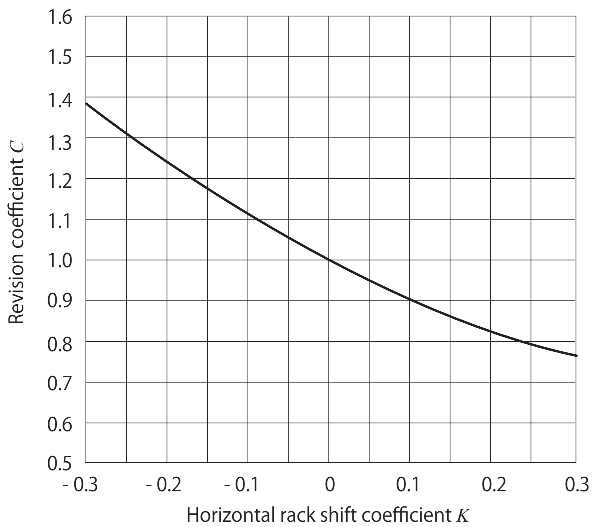


Table 1. Table for Form factor

Item No.	Transverse reference profile (Transverse tooth thickness : 0.5πm)						Mean spiral angle
	Normal reference pressure angle α_n	Tooth depth (heel) h	Addendum (heel) h_{a0}	Dedendum (heel) h_{f0}	Bottom clearance (heel) c	Cutter tip radius (normal) r	β_m
1	20°	1.888m	0.850m	1.038m	0.188m	0.12m	15°
2							20°
3							25°
4							30°
5							35°
6		2.188m	1.000m	1.188m			0°
7	22.5°	1.888m	0.850m	1.038m	0.188m	0.12m	35°
8		1.788m	0.800m	0.988m			0°
9	25°	1.888m	0.850m	1.038m	0.188m	0.12m	35°
10		1.788m	0.800m	0.988m			0°

Fig. 2 Form factor graph (No.6)

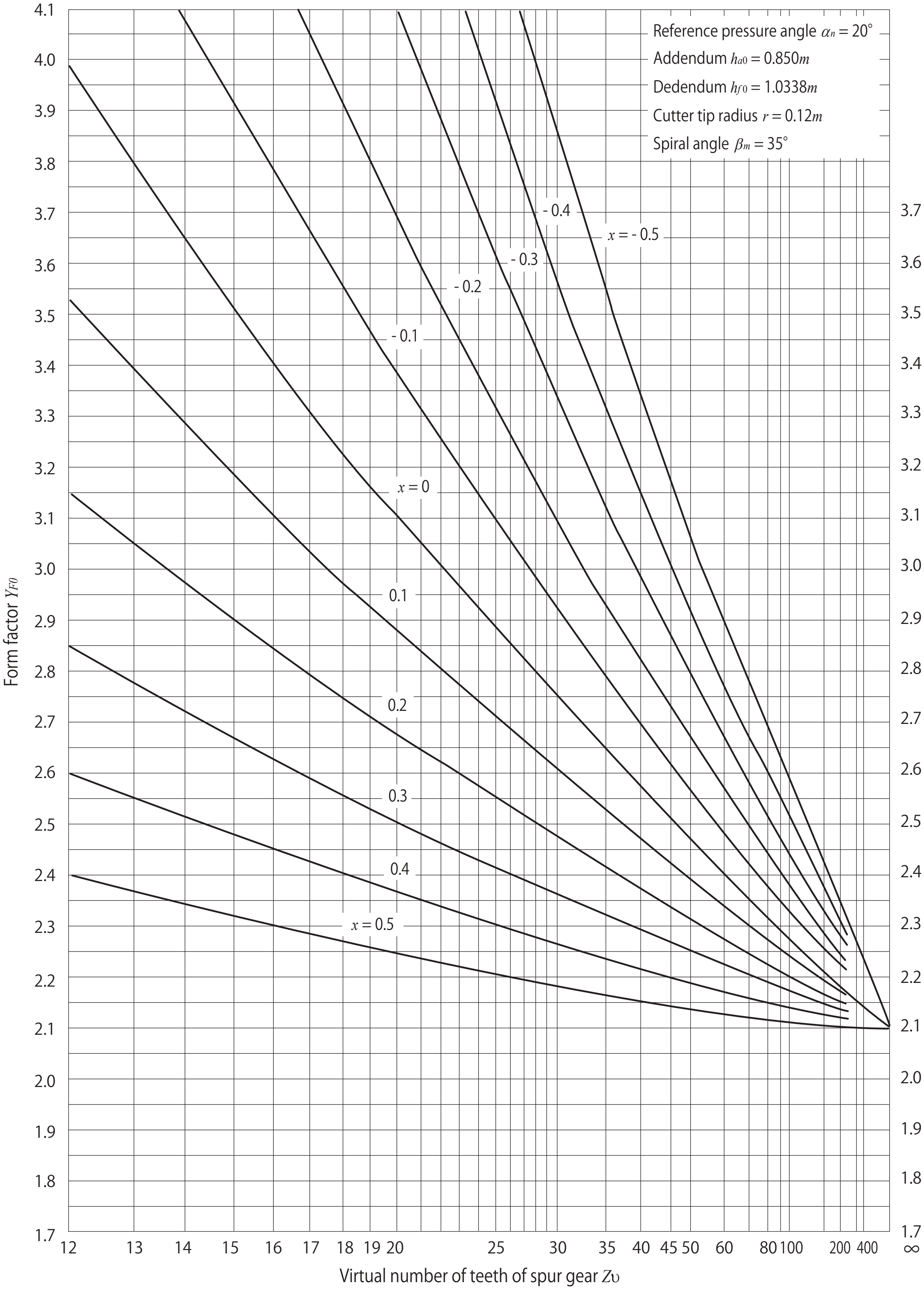
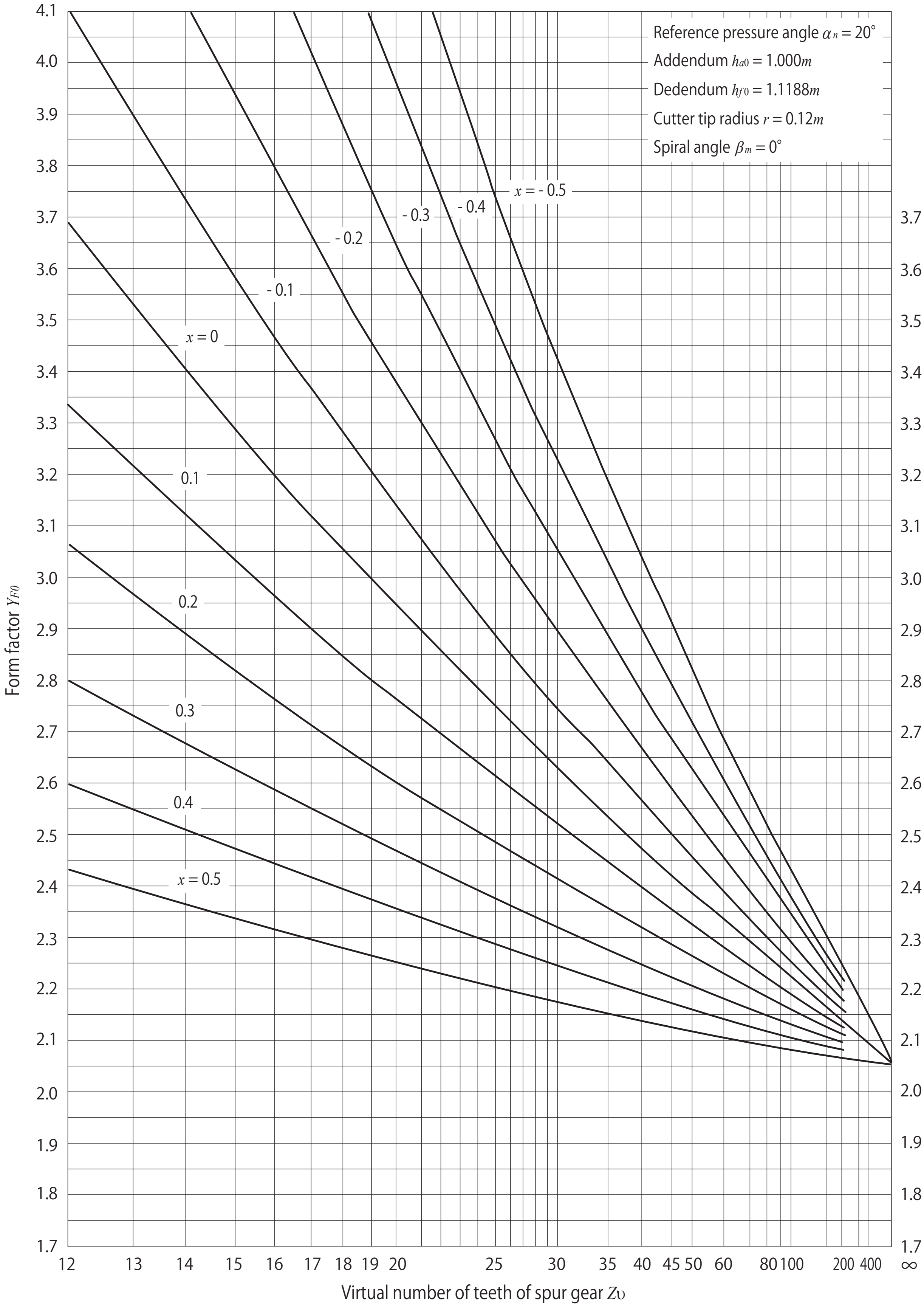


Fig. 3 Form factor graph (No.5)



5.1.3 Load distribution factor Y_ε

Calculation of Load distribution factor is as follows.

$$Y_\varepsilon = \frac{1}{\varepsilon_\alpha} \quad \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots (20)$$

Hereby

ε_α : Transverse contact ratio

(a) Obtain Transverse contact ratio using following formula (21-24). However use Straight bevel gear's calculation formula for Zerol Bevel gear.

Straight bevel gear

$$\varepsilon_\alpha = \frac{\sqrt{R_{ra1}^2 - R_{rb2}^2} + \sqrt{R_{ra2}^2 - R_{rb2}^2} - (R_{r1} + R_{r2})\sin\alpha}{m\pi\cos\alpha} \quad \cdots (21)$$

Use following summarized calculation formula (1) for gear ratio $u \geqq 2$

$$\varepsilon_\alpha = \frac{\sqrt{R_{ra1}^2 - R_{rb1}^2} + h_{a2}\operatorname{cosec}\alpha - R_{r1}\sin\alpha}{m\pi\cos\alpha} \quad \cdots\cdots\cdots (22)$$

Spiral bevel gear

$$\varepsilon_\alpha = \frac{\sqrt{R_{ra1}^2 - R_{rb1}^2} + \sqrt{R_{ra2}^2 - R_{rb2}^2} - (R_{r1} + R_{r2})\sin\alpha_t}{m\pi\cos\alpha_t} \quad \cdots (23)$$

Use following summarized calculation formula (1) for gear ratio $\upsilon \geqq 2$

$$\varepsilon_\alpha = \frac{\sqrt{R_{ra1}^2 - R_{rb1}^2} + h_{a2}\operatorname{cosec}\alpha_t - R_{r1}\sin\alpha_t}{m\pi\cos\alpha_t} \quad \cdots\cdots\cdots (24)$$

Note (1) Formulas (21) and (23) becomes complicated for Gear section thus Gear is assumed as Rack to show a summarized formula as follows.

Hereby (refer to Fig. 4)

$R_{\upsilon a}$: Tip diameter (mm) for Virtual spur gear on the Back cone = $R_\upsilon + h_a = \gamma\sec\delta + h_a$

$R_{\upsilon b}$: Base radius (mm) for Virtual spur gear on the Back cone

For Straight bevel gear = $R_{\upsilon\cos\alpha} = \gamma\sec\delta\cos\alpha$

For Spiral bevel gear = $R_{\upsilon\cos\alpha t} = \gamma\sec\delta\cos\alpha_t$

R_υ : Back cone distance (mm) = $\gamma\sec\delta$

γ : Radius of pitch circle (mm) = 0.5 zm

h_a : Outer addendum (mm)

α : Reference pressure angle (°)

α_t : Mean transverse pressure angle (°) = $\tan^{-1}(\tan\alpha_n / \cos\beta_m)$

α_n : Normal reference pressure angle (°)

β_m : Mean spiral angle (°)

δ : Pitch angle (°)

m : Outer transverse module (mm)

z : Number of teeth

Subscript

1 : Pinion

2 : Gear

(b) Refer to Fig. 5 to calculate Transverse contact ratio εa for Straight bevel gear with Reference pressure angle 20° or Spiral bevel gear with Normal pressure angle 20°. Use formula (16) to calculate Virtual number of teeth of spur gear Z_υ and the following formula for u.

Straight bevel gear : $u = \dfrac{h_a}{m}$ $\cdots\cdots\cdots\cdots\cdots\cdots$ (25)

Spiral bevel gear : $u = \dfrac{h_a}{m\cos\beta_m}$ $\cdots\cdots\cdots\cdots\cdots$ (26)

Hereby

h_a : Outer addendum (mm)

m : Outer transverse module (mm)

βm : Mean spiral angle (°)

From Fig. 5, calculate Transverse contact ratio ε_α using following formulas.

Straight bevel gear : $\varepsilon_\alpha = \varepsilon_1 + \varepsilon_2$

Spiral bevel gear : $\varepsilon_\alpha = K\varepsilon'_\alpha$

$\varepsilon'_\alpha = \varepsilon_1 + \varepsilon_2$

Hereby

ε_α : Transverse contact ratio for Straight bevel gear

ε'_α : Virtual spur gear transverse contact ratio for Spiral bevel gear

$\varepsilon_1, \varepsilon_2$: Obtain Virtual spur gear contact ratio from Pitch point to Tooth tip for pinion and gear from Fig. 5

k : Use Table 2 conversion factor for Virtual spur gear normal contact ratio to Transverse contact ratio for Spiral bevel gear.
$= \cos^2\alpha_n(\cos^2\beta_m + \tan^2\alpha_n)$

α_n : Normal reference pressure angle (°)

β_m : Mean spiral angle (°)

Fig. 4 Engagement of Virtual spur gear on the Back cone

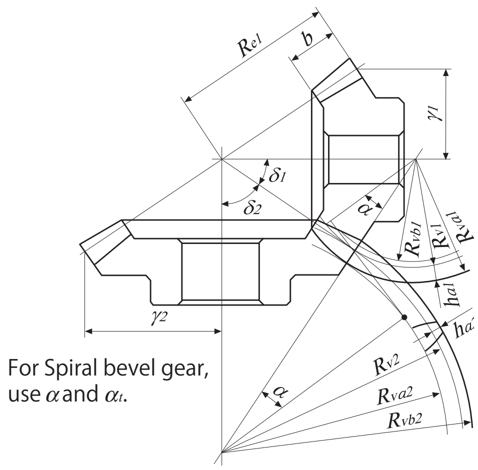
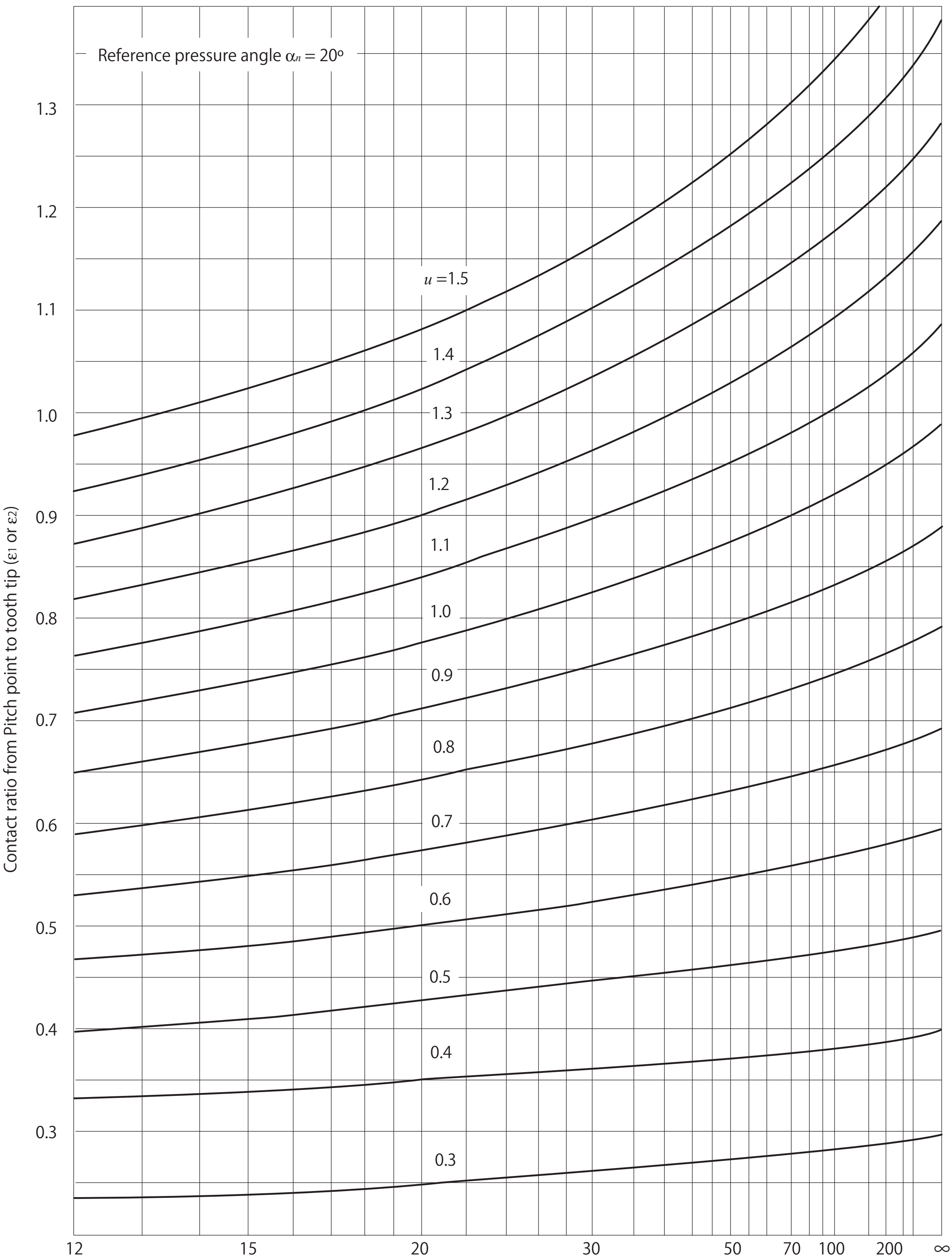


Table 2. Value of Conversion factor for Transverse contact ratio for Spiral bevel gear

Mean spiral angle β_m / Normal Reference pressure angle α_n	15°	20°	25°	30°	35°
20°	0.94085	0.89671	0.84229	0.77924	0.70949

Fig. 5 Table to obtain Contact ratio



Virtual number of teeth of Spur gear $\left(z_r = \dfrac{z}{\cos\delta \cos^3 \beta_m} \right)$

5.1.4 Spiral angle factor Y_β

Calculate Spiral angle factor using following formulas. (Refer to Table 3 and Fig. 6)

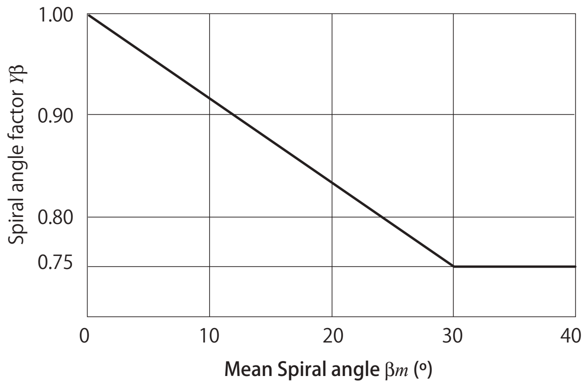
For $0° \leqq \beta_m \leqq 30°$: $Y_\beta = 1 - \frac{\beta_m}{120}$ .... (27)

For $\beta_m \geqq 30°$: $Y_\beta = 0.75$ .... (27)'

Table 3. Spiral angle factor

βm	15°	20°	25°	30°	35°
$Y\beta$	0.875	0.833	0.792	0.75	

Fig. 6 Spiral angle factor



5.1.5 Cutter diameter influence factor Y_C

Calculate Cutter diameter influence factor from Table 4 based on ratio cutter diameter for Length of tooth trace. If cutter diameter is unknown, Y_C=1.0. Length of tooth trace to be $b / \cos\beta_m$ (mm).

5.1.6 Life factor K_L

Refer to Table 2 of 5.1.5 under Spur gear.

5.1.7 Dimension factor for Tooth root factor K_{FX}

Obtain Dimension factor for Tooth root factor from transverse module in Table 5.

Table 5. Dimension factor for Tooth root factor K_{FX}

Outer transverse module m	Non surface hardening gear	Surface hardening gear
1.5 < d ≦ 5	1.0	1.0
5 < d ≦ 7	0.99	0.98
7 < d ≦ 9	0.98	0.96
9 < d ≦ 11	0.97	0.94
11 < d ≦ 13	0.96	0.92
13 < d ≦ 15	0.94	0.90
15 < d ≦ 17	0.93	0.88
17 < d ≦ 19	0.92	0.86
19 < d ≦ 22	0.90	0.83
22 < d ≦ 25	0.88	0.80

5.1.8 Tooth distributed factor for Tooth load K_M

Calculate load distribution factor for Tooth trace from Tables 6 and 7.

5.1.9 Dynamic load factor K_V

Using Gear accuracy and Circumferential speed on the Outer pitch circle from Table 8 to obtain Dynamic factor.

5.1.10 Overload factor K_O

Refer to formula (23) and Table 4 of 5.1.8 under Spur gear.

Table 4. Cutter diameter influence factor Y_C

Types	Cutter diameter			
	∞	6 times Length of tooth trace	5 times Length of tooth trace	4 times Length of tooth trace
Straight bevel gear	1.15	-	-	-
Spiral bevel gear Zerol Bevel gear	-	1.00	0.95	0.90

Table 6. Tooth trace load distribution factor K_M for Spiral bevel, Zerol bevel and Straight bevel gears (Crowning)

		Full support to both gears	Support to one side of gear	Support to both gears on one side
Stiffness of axis and gearbox	Especially strong	1.2	1.35	1.5
	Normal	1.4	1.6	1.8
	Weak	1.55	1.75	2.0

Table 7. Tooth trace load distributed factor K_M for Straight bevel gear without Crowning

		Full support to both gears	Support to one side of gear	Support to both gears on one side
Stiffness of axis and gearbox	Especially strong	1.05	1.15	1.35
	Normal	1.6	1.8	2.1
	Weak	2.2	2.5	2.8

Table 8. Dynamic factor K_V

System of accuracy from JIS B1704	Circumferential velocity (m/s)						
	Below 1	$1 < \upsilon \leqq 3$	$3 < \upsilon \leqq 5$	$5 < \upsilon \leqq 8$	$8 < \upsilon \leqq 12$	$12 < \upsilon \leqq 18$	$18 < \upsilon \leqq 25$
1	1.0	1.1	1.15	1.2	1.3	1.5	1.7
2	1.0	1.2	1.3	1.4	1.5	1.7	-
3	1.0	1.3	1.4	1.5	1.7	-	-
4	1.1	1.4	1.5	1.7	-	-	-
5	1.2	1.5	1.7	-	-	-	-
6	1.4	1.7	-	-	-	-	-

5.1.11 Reliability factor K_R

Reliability factor is as follows

(1) General cases $K_R = 1.2$

(2) Special cases

If clearly understood the usage conditions of impact from prime mover, driver side, stiffness of gearbox and axis for calculating Tooth bending strength. When determining numerical values of K_M, K_L, K_0 using ………………………… $K_R = 1.0$.

In situations opposite from above where numerical values of K_0 and K_M are uncertain (use K_L as 1.0 in this case). ………………………… $K_R = 1.4$

5.1.12 Allowable tooth root bending stress $\sigma_{F\lim}$

Refer to Tables 9, 10 and 13 of 5.1.10 under Spur gear.

5.2 How to calculate factors from calculation formula for Surface durability.

The following stipulates types of factor from calculation formula of Surface durability in previous paragraph.

5.2.1 Facewidth b (mm)

Facewidth b is stipulated to the Facewidth on Pitch cone. For different Facewidth between Pinion and Gear, select the narrower Effective facewidth.

5.2.2 Domain zone Z_H

Calculation of Domain zone is as follows.

$$Z_H = \sqrt{\frac{2\cos\beta_b}{\sin\alpha_t \cos\alpha_t}} \quad \cdots\cdots (28)$$

Hereby

β_b : $\tan^{-1}(\tan\beta_m \cos\alpha_t)$

α_t : Mean transverse pressure angle (°)

α_n : Normal reference pressure angle (°)

β_m : Mean spiral angle (°)

Obtain domain factor from Fig. 7 with Normal reference pressure angle 20°, 22.5°and 25°.

5.2.3 Elasticity factor Z_M

Refer to Table 6 of 5.2.3 under Spur gear

5.2.4 Contact ratio factor Z_ε

Obtain Contact ratio factor using following formula. Refer to Fig. 4 of 5.2.4 under Spur gear.

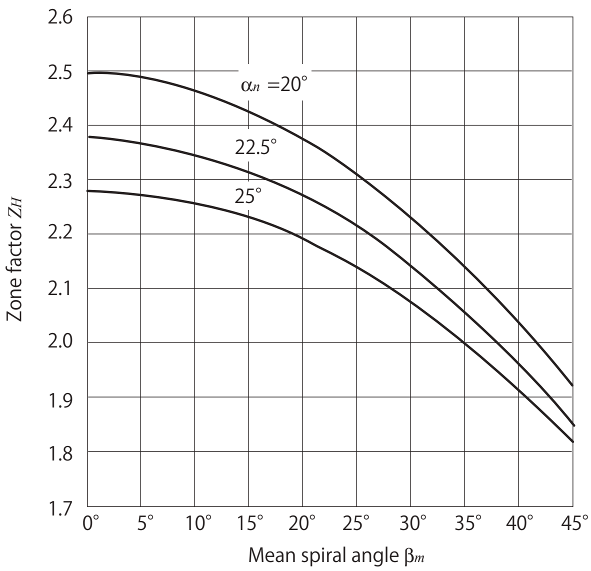
Straight bevel gear : $Z_\varepsilon = 1.0$ ………………………… (29)

Spiral bevel gear :

In case of $\varepsilon_\beta \leqq 1$, $\quad Z_\varepsilon = \sqrt{1 - \varepsilon_\beta + \dfrac{\varepsilon_\beta}{\varepsilon_\alpha}}$ ………………………… (30)

In case of $\varepsilon_\beta > 1$, $\quad Z_\varepsilon = \sqrt{\dfrac{1}{\varepsilon_\alpha}}$ ………………………… (31)

Fig. 7 Zone factor



Hereby

ε_α : Transverse contact ratio

ε_β : Overlap ratio

Calculate Transverse contact ratio from 5.1.3 (a) under Bevel gear.

Overlap ratio is defined below

$$\varepsilon_\beta = \frac{R_e}{R_e - 0.5b} \; \frac{b\tan\beta_m}{\pi m} \quad \cdots\cdots (32)$$

Hereby

R_e : Cone distance (mm)

b : Facewidth (mm)

β_m : Mean spiral angle (°)

m : Outer transverse module (mm)

5.2.5 Spiral angle factor for Surface durability Z_β

Spiral angle factor for Surface durability is difficult to stipulate accurately due to insufficient data. Calculation formula is $Z_\beta = 1.0$ ………………………… (33)

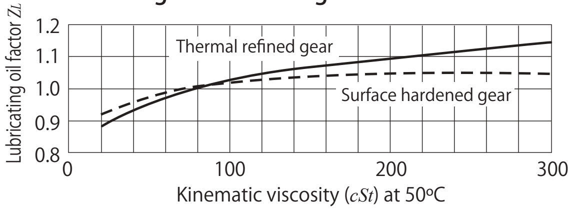
5.2.6 Life factor for Surface durability K_{HL}

Refer to Table 7 of 5.2.6 under Spur gear.

5.2.7 Lubricating oil factor Z_L

For the 2 types of gear stated below, obtain Lubricating oil factor from Fig.8 based on Kinematic viscosity (cSt) at 50°C.

Fig. 8 Lubricating oil factor



(1) Thermal refined gear [1]: Use solid line in Fig. 8.

(2) Surface hardened gear: Use broken line in Fig. 8.

Note (1) Thermal refined gear includes gear with quenching, tempering and normalizing.

Remark: Casting steel gear is equivalent to thermal refined gear.

5.2.8 Roughness factor Z_R

For 2 types of gear stated below, obtain average of maximum height of profile factor from Fig. 9 based on mean roughness of flank $R_{\max m}$(μm). Use the following formula to obtain the average of maximum height of profile roughness of flank $R_{\max m}$ from $R_{\max 1}$, $R_{\max 2}$. (Meaning of $R_{\max 1}$, $R_{\max 2}$ is Maximum height if profile roughness of flank inclusive of the effects of warm up and test run.)

$$R_{\max m} = \frac{R_{\max 1} + R_{\max 2}}{2} \sqrt[3]{\frac{100}{a}} (\mu m) \quad \cdots\cdots (34)$$

Hereby

$a = R_m (\sin\delta_1 + \cos\delta_1)$

R_m : Mean cone distance (mm)

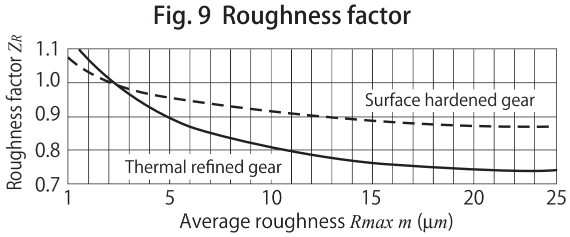
δ_1 : Pitch angle (°) of Pinion

(1) Thermal refined gear [1]: Use solid line in Fig. 9.

(2) Surface hardened gear: Use broken line in Fig. 9.

Refer to 5.2.7 for Note (1) and Remark

Fig. 9 Roughness factor



5.2.9 Lubricating speed factor Z_V

For the 2 types of gear stated below, obtain Lubricating velocity factor from Fig. 10 based on Circumferential velocity υ(m/s) on the Outer pitch circle.

(1) Thermal refined gear (1): Use solid line in Fig. 10.

(2) Surface hardened gear: Use broken line in Fig. 10.

Refer to 5.2.7 for Note (1) and Remark

Fig. 10 Lubricating speed factor

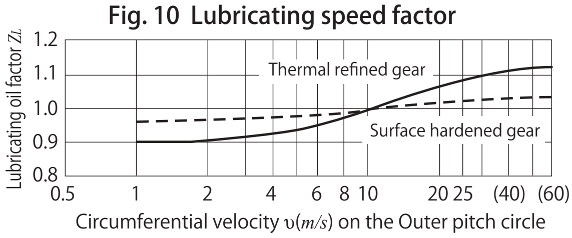


Table 11. Nitriding gear [1]

Material		Flank hardness (reference)	$\sigma_{H\lim}$ kgf/mm²	
Nitriding steel	SACM 645 and others	Above HV 650	Normal	120
			Sustained period of Nitriding treatment	130 - 140

Note (1) Applicable to Gear with proper Nitriding depth and hardened surface to improve Surface durability. When Surface hardness is remarkably lower than above table. Starting point of maximum shear-stress force at inner gear tooth is remarkably deeper than depth of Nitriding, take note of providing a larger safety factor than usual.

Table 12. Nitrocarburizing gear [1]

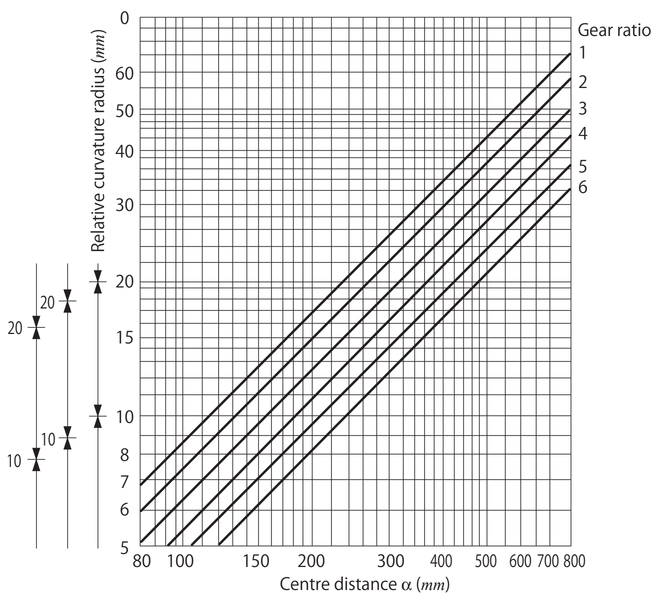
Material	Nitriding period (h)	$\sigma_{H\lim}$ kgf/mm²		
		Relative curvature radius (mm) [2]		
		Below 10	10 - 20	Above 20
Carbon steel and Alloy steel for structural use	2	100	90	80
	4	110	100	90
	6	120	110	100

Note (1) Applicable to Salt bath and Gas Nitro-carburizing gears.

(2) Use Fig. 11 to obtain Relative curvature radius

Remark. Use properly adjusted material for core.

Fig. 11 Relative curvature radius



5.2.10 Hardness ratio factor Z_W

Refer to formula (35) and Table 8 from 5.2.10 under Spur gear.

5.2.11 Diameter factor K_{HX} for Surface durability

If Tooth profile and gear size increases, Surface durability also increases but has a tendency to increase disproportionately. Due to insufficient data at the moment, Dimension factor $K_{HX} = 1.0$ ·························· (35)

5.2.12 Tooth trace load distribution factor $K_{H\beta}$ for Surface durability

Obtain Tooth trace load distribution factor for Surface durability from Tables 9 and 10. If both gears are without surface hardening, use 90% of values from Tables 9 and 10.

Table 9. Tooth trace load distribution factor $K_{H\beta}$ for Spiral Bevel, Zerol Bevel and Straight bevel gears (including Crowning)

Stiffness of axis and gearbox	Condition for gear support		
	Full support to both gears	Support to one side of gear	Support to both gears on one side
Especially strong	1.3	1.5	1.7
Normal	1.6	1.85	2.1
Weak	1.75	2.1	2.5

Table 10. Tooth trace load distribution factor $K_{H\beta}$ for Straight bevel gear without Crowning.

Stiffness of axis and gearbox	Condition for gear support		
	Full support to both gears	Support to one side of gear	Support to both gears on one side
Especially strong	1.3	1.5	1.7
Normal	1.85	2.1	2.6
Weak	2.8	3.3	3.8

5.2.13 Dynamic factor K_V

Refer to Table 8 from 5.1.9 under Bevel gear.

5.2.14 Overload factor K_o

Refer to formula (23) and Table 4 of 5.1.8 under Spur gear.

5.2.15 Reliability factor CR

Reliability factor for Surface durability is above 1.15.

5.2.16 Allowable hertz stress $\sigma_{H\lim}$

Refer to Tables 9 ~ 12 for Allowable hertz stress. For values not listed, use interpolation. Meaning of flank's hardness is hardness near Pitch circle.

10.3 Calculation for Cylindrical worm gear pair strength

Gear strength calculation formula for Cylindrical worm gear pair **JGMA 405-01 (1978)**

1. Applicable range (Common)

1.1 This standard is applied to Worm gear pair with following ranges and shaft angle 90° for power transfer used in general industrial machinery.

Axial module	: 1 ～ 25 mm
Reference diameter of Worm wheel	: Below 900 mm
Sliding velocity	: Below 30 m/s
Revolving velocity of Worm wheel	: Below 600 min^{-1}
Tooth profile	: Stipulated in JIS B1723 (Cylindrical worm gear pair)
Material	: Refer to Table 7

1.2. This standard is used for calculating Allowable load from given dimension of Cylindrical worm gear pair or is used for determining suitable dimensions of Cylindrical worm gear pair from given load.

2. Definition

Gear strength of Cylindrical worm gear pair is Allowable load for Surface durability.

3. Basic conversion formula and numerical value

3.1 Sliding velocity v_s (m/s)

$$v_s = \frac{d_1 n_1}{19100 \cos\gamma} \quad \cdots (1)$$

Hereby

d_1 : Reference pitch diameter of Worm gear (mm)
n_1 : Revolving velocity of Worm gear (min^{-1})
γ : Reference pitch cylindrical lead angle (°)

3.2 Torque, Tangential load and Efficiency

(1) When Worm gear is driver (speed reduction)

$$T_2 = \frac{F_t d_2}{2000} \text{ (kgf・m)} \quad \cdots (2)$$

$$T_1 = \frac{T_2}{u \eta_R} = \frac{F_t d_2}{2000 u \eta_R} \text{ (kgf・m)} \quad \cdots (3)$$

$$\eta_R = \frac{\tan\gamma\left(1 - \tan\gamma \dfrac{\mu}{\cos\alpha_n}\right)}{\tan\gamma + \dfrac{\mu}{\cos\alpha_n}} \quad \cdots (4)$$

Hereby

T_2 : Nominal torque (kgf・m) for Worm wheel
T_1 : Nominal torque (kgf・m) for Worm gear
F_t : Nominal Tangential load (kgf) on the Reference pitch circle for Worm wheel
d_2 : Reference pitch diameter (mm) for Worm wheel
u : Gear ratio
η_R : Transfer efficiency of Worm gear when Worm gear is driver (excludes bearing loss and mixer loss of lubricating oil)
μ : Friction factor {Refer to (3) of 3.2}
α_n : Normal reference pressure angle(°)

(2) When Worm wheel is driver (speed increment)

$$T_2 = \frac{F_t d_2}{2000} \text{ (kgf・m)} \quad \cdots \text{Same as (2)}$$

$$T_1 = \frac{T_2 \eta_1}{u} = \frac{F_t d_2 \eta_1}{2000 u} \text{ (kgf・m)} \quad \cdots (5)$$

$$\eta_1 = \frac{\tan\gamma - \dfrac{\mu}{\cos\alpha_n}}{\tan\gamma\left(1 + \tan\gamma \dfrac{\mu}{\cos\alpha_n}\right)} \quad \cdots (6)$$

Hereby

η_1 : Transfer efficiency of Worm gear pair when Worm wheel is driver (excludes bearing loss and mixer loss of lubricating oil).

(3) Numerical value of friction factor μ

Obtain Friction factor μ from Fig. 1 of sliding velocity when engaged with Worm gear with Case harden and ground or Worm wheel with phosphor bronze.

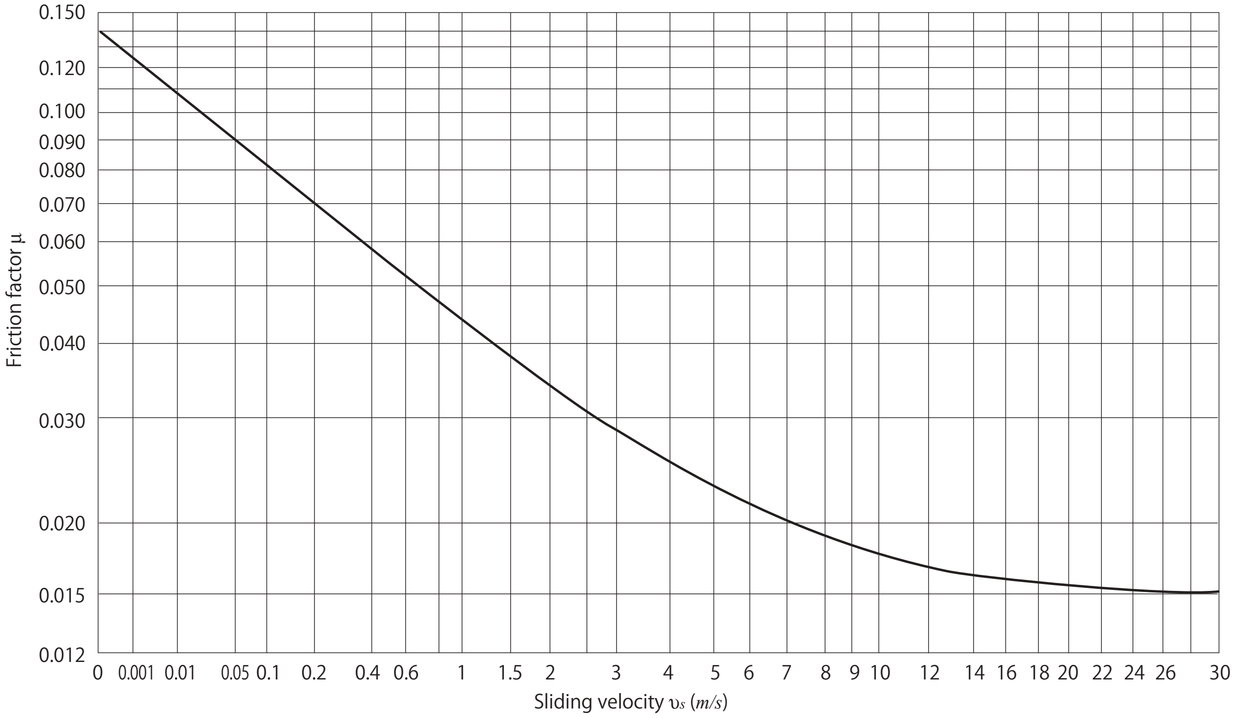
Remark 1. Friction factor for engagement with other materials.

Due to insufficient data, values of Friction factor are difficult to stipulate. Therefore Reference table 1 proposed by H.E Merritt is adopted for reference.

Reference table 1 Friction factor μ for different materials combination

Materials	Value of μ
Casting iron and phosphor bronze	1.15 times value of Fig. 1
Casting iron and Casting iron	1.33 times value of Fig. 1
Hardened steel and Aluminium	1.33 times value of Fig. 1
Steel and Steel	2.0 times value of Fig. 1

Fig. 1 Friction factor



4. Calculation formula of Allowable load for Surface durability

4.1 Basic load capacity calculation

Calculate Basic load capacity for Surface durability from given dimensions and material of Cylindrical worm gear pair using following calculation formula.

Allowable tangential load $F_{t\lim}$ (kg・f)

$$F_{t\lim} = 3.82 K_{\upsilon} K_{n} S_{c\lim} Z d_2^{0.8} m_x \frac{Z_L Z_M Z_R}{K_C} \quad \cdots\cdots(7)$$

Allowable Torque for Worm wheel $T_{2\lim}$ (kgf・m)

$$T_{2\lim} = 0.00191 K_{\upsilon} K_{n} S_{c\lim} Z d_2^{1.8} m_x \frac{Z_L Z_M Z_R}{K_C} \quad \cdots\cdots(8)$$

Hereby

- d_2 : Reference pitch diameter (mm) for Worm wheel
- m_x : Axial module (mm)
- Z : Zone factor
- K_{υ} : Sliding velocity factor
- K_n : Revolving speed factor
- Z_L : lubricating oil factor
- Z_M : Lubrication factor
- Z_R : Roughness factor
- K_c : Tooth contact factor
- $S_{c\lim}$: Allowable stress factor for Surface durability

4.2 Equivalent load calculation

Basic load capacity from formulas (7) and (8) is the limit of Tangential load and torque to withstand 26,000 hours of usage when in a non-impact environment. It is considered non impact if number of starts per hour is under 2 times and starting impact torque is below 200% of rated torque[1]. However, if such condition is not met, calculate Equivalent load and compare with basic load capacity. In other words, when expected life is more or less than 26,000 hours with impact conditions applied. Starting toque is larger than above. Calculation method for Equivalent load is as follow.

Note(1) This is torque for Worm wheel when prime mover (or load) performs rated load operation.

Equivalent tangential load F_{te} (kgf)

$F_{te} = F_t K_h K_s$ ··················(9)

Virtual torque of Worm wheel T_{2e} (kgf・m)

$T_{2e} = T_2 K_h K_s$ ··················(10)

Hereby

F_t : Nominal tangential load on the Pitch circle of Worm wheel (kgf)

T_2 : Nominal torque of Worm wheel (kgf・m)

K_s : Starting factor (Refer to 5.9)

K_h : Time factor (Refer to 5.10)

4.3 Load definition

(1) When non-impact, expected life is 26,000 hours.

It should meet the following conditions.

$F_t \leqq F_{t\lim}$ ··················(11)

$T_2 \leqq T_{2\lim}$ ··················(12)

(2) Other than above cases,

it should meet the following conditions.

$F_{te} \leqq F_{t\lim}$ ··················(13)

$T_{2e} \leqq T_{2\lim}$ ··················(14)

Remark: For fluctuating load, use total torque T_{2c} to define load based on formulas (10) and (12) instead of T_2. Calculation method of T_{2c} is found at「Calculation of Fluctuating load」in Reference table 4 (page 153).

5. How to calculate each factor for Surface durability from calculation formula

Factors used for Surface durability calculation formulas mentioned above are stipulated below.

5.1 Facewidth of Worm wheel b_2 (mm)

Refer to Fig. 2 for Facewidth of Worm wheel.

5.2 Zone factor Z

Calculate Zone factor from (1) and (2) using Table 3.

(1) When , $b_2 < 2.3 m_x \sqrt{Q+1}$ use value in Table 3 multiplied by

$$\frac{b_2}{2 m_x \sqrt{Q+1}}$$ as value for Z.

(2) When $b_2 \geqq 2.3 m_x \sqrt{Q+1}$, use value in Table 3 multiplied by 1.15 as value for Z.

Hereby

Q : Diameter quotient $\left(Q = \frac{d_1}{m_x}\right)$

Z_w : Number of thread for Worm gear

5.3 Sliding velocity K_v

Obtain Sliding velocity factor based on Sliding velocity from Fig. 3.

5.4 Revolving velocity factor K_n

Obtain Revolving velocity factor based on Revolving speed of Worm wheel from Fig. 4.

5.5 Lubricating oil factor Z_L

As long as lubricating oil with proper viscosity containing extreme additives is used, $Z_L = 1.0$.

If bearing is used in Worm gear pair equipment or compelled to use lubricating oil with thin viscosity. Z_L is less than 1.0.

Remark: Viscosity

There are many recommended viscosity values from different sources for proper lubricating oil. However, there is no consensus. Recommended mean values are collected from sources and shown in Reference table 2.

Fig. 2 Facewidth of Worm wheel

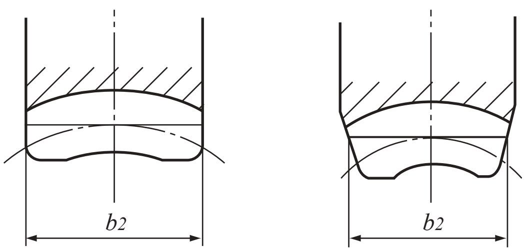


Table 3. Base value of Zone factor

Z_w \ Q	7	7.5	8	8.5	9	9.5	10	11	12	13	14
1	1.052	1.065	1.084	1.107	1.128	1.137	1.143	1.160	1.202	1.260	1.318
2	1.055	1.099	1.144	1.183	1.114	1.223	1.231	1.250	1.280	1.320	1.360
3	0.989	1.109	1.209	1.266	1.305	1.333	1.350	1.365	1.393	1.422	1.442
4	0.981	1.098	1.204	1.301	1.380	1.428	1.460	1.490	1.515	1.545	1.570

Reference table 2 Recommended dynamic viscosity

Unit cSt/37.8°C

Operating oil temperature		Sliding velocity m/s		
Max. oil temperature	Starting oil temperature	Below 2.5	Above 2.5 to below 5	Above 5
0 C° to below 10°C	−10°C to below 0C°	110 - 130	110 - 130	110 - 130
	Above 0°C	110 - 150	110 - 150	110 - 150
10 C° to below 30°C	Above 0°C	200 - 245	150 - 200	150 - 200
30 C° to below 55°C	〃	350 - 510	245 - 350	200 - 245
55 C° to below 80°C	〃	510 - 780	350 - 510	245 - 350
80 C° to below 100°C	〃	900 - 1100	510 - 780	350 - 510

Fig. 3 Sliding velocity factor

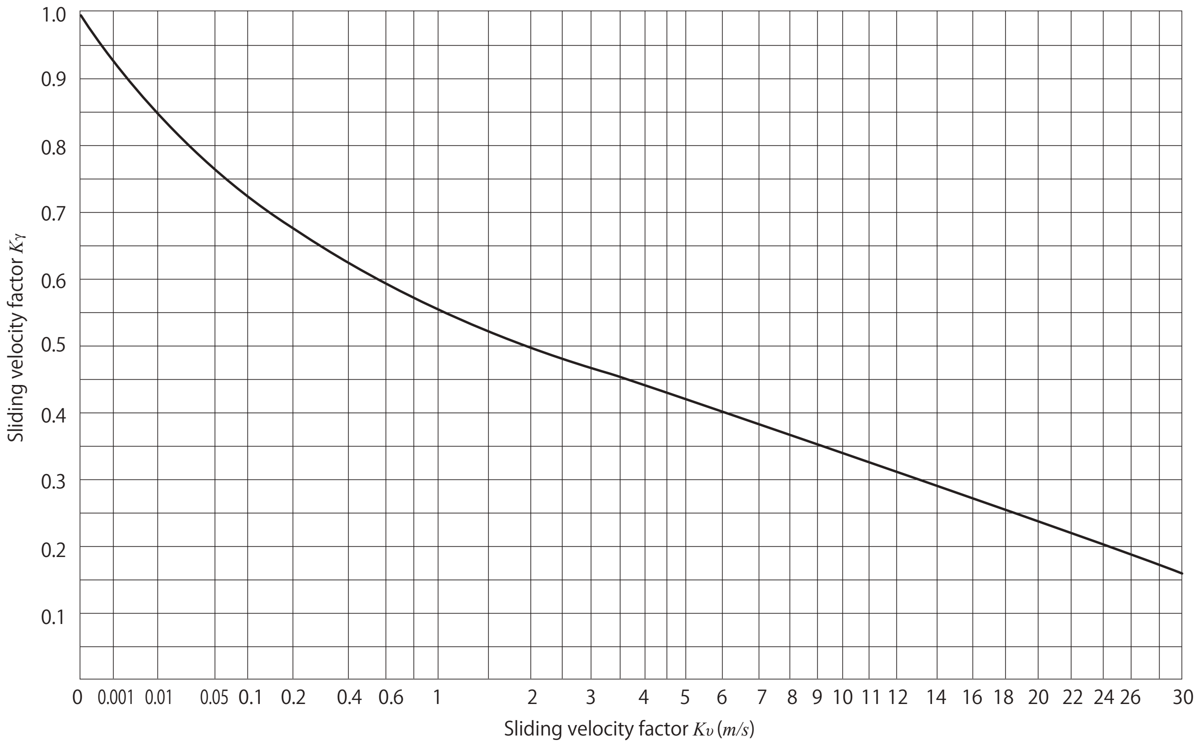
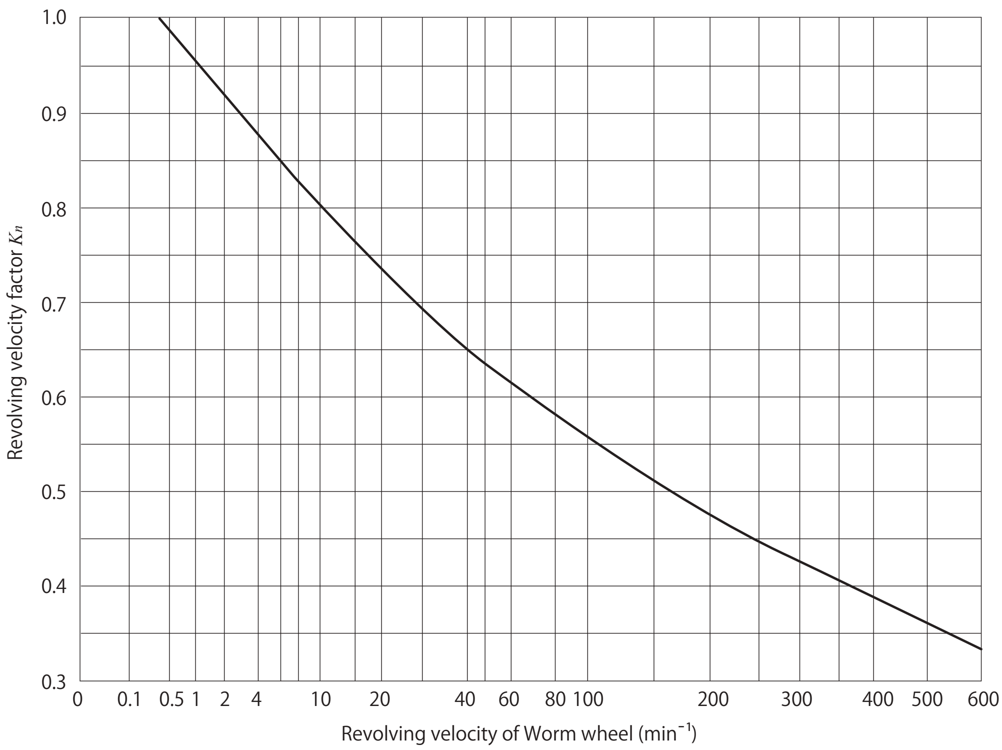


Fig. 4 Revolving velocity factor



5.6 Lubrication factor Z_M

Obtain Lubrication factor from Table 4.

Table 4. Lubrication factor Z_M

Sliding velocity m/s	Below 10	Above 10, below 14	Above 14
Oil bath lubrication	1.0	0.85	-
Forced lubrication	1.0	1.0	1.0

5.7 Roughness factor Z_R

Roughness factor is determined with consideration based on influence on Pitting and Wearing to flank of Worm gear and Worm wheel. Due to insufficient data, $Z_R = 1.0$ is adopted at the moment·································(15)

However, Surface roughness is to be below 3S for Worm gear and below 12S for Worm wheel.

If Surface roughness is rougher than above, Roughness factor Z R should be lower than 1.0.

5.8 Tooth bearing K_c

Quality of Tooth bearing has large influence on load capacity. Due to insufficient data at the moment, Tooth bearing for classification equivalent to A in JIS B 1741 (tooth bearing) will be $K_c = 1.0$··························(16)

Value of K_c for classification B and C is larger than 1.0.

Reference table 3: Shows JIS Tooth bearing ratio and approximate values of K_c.

5.9 Starting factor K_s

Starting factor is stipulated below

(1) Obtain value from Table 5 if the starting torque is below 200% of rated torque.

(2) If Starting torque exceeds 200% of rated torque, value of $K_s = 1.0$. With starting torque to be as maximum, then calculate fluctuating load (refer to Table 4) to calculate total load.

5.10 Time factor K_h

Obtain Time factor from Table 6 using expected lifespan and extent of impact. Use interpolation when expected lifespan is between the values in the below Table.

Reference table 3 Classification of Tooth bearing and approximate value of K_c

Classification	Ratio of tooth bearing		K_c
	Tooth trace direction	Direction of tooth depth	
A	Above 50% of length of effective trace direction	Above 40% of effective tooth depth	1.0
B	Above 35% of length of effective trace direction	Above 30% of effective tooth depth	1.3 - 1.4
C	Above 20% of length of effective trace direction	Above 20% of effective tooth depth	1.5 - 1.7

Remark: Conditions for tooth bearing from JIS B1741

Table 5. Starting factor K_s

Number of start times per hour	Below 2 times	2 - 4 times	5 - 9 times	Above 10 times
K_s	1.0	1.07	1.13	1.18

Table 6. Time factor K_h

Impact from prime mover side	Expected lifespan	K_h		
		Impact from load		
		Uniform load	Medium impact	Heavy impact
Uniform load (Motor, Turbine, Hydraulic motor and others)	1,500 hours	0.80	0.90	1.0
	5,000 hours	0.90	1.0	1.25
	26,000 hours (1)	1.0	1.25	1.50
	60,000 hours	1.25	1.50	1.75
Light impact (Multiple cylinder engine)	1,500 hours	0.90	1.0	1.25
	5,000 hours	1.0	1.25	1.50
	26,000 hours (1)	1.25	1.50	1.75
	60,000 hours	1.50	1.75	2.0
Medium impact (Cylinder engine)	1,500 hours	1.0	1.25	1.50
	5,000 hours	1.25	1.50	1.75
	26,000 hours (1)	1.50	1.70	2.0
	60,000 hours	1.75	2.0	2.25

Note (1) Operating 10 hours a day for 260 days per a year is equivalent to 10 years and above.

5.11 Allowable stress factor S_{clim}

Table 7 shows Allowable stress factor and limits of sand burning sliding speed for Surface durability.

Table 7. Allowable stress factor S_{clim} for Surface durability

Material of Worm wheel	Material of Worm gear	Sclim	Limits of sand burning sliding velocity (1) m/s
Phosphor bronze centrifugal casting	Alloyed steel with Case hardening	1.55	30
	Alloyed steel HB400	1.34	20
	Alloyed steel HB250	1.12	10
Phosphor bronze chill casting	Alloyed steel with Case hardening	1.27	30
	Alloyed steel HB400	1.05	20
	Alloyed steel HB250	0.88	10
Phosphor bronze sand casting or Forging	Alloyed steel with Case hardening	1.05	30
	Alloyed steel HB400	0.84	20
	Alloyed steel HB250	0.70	10
Aluminum bronze	Alloyed steel with Case hardening	0.84	20
	Alloyed steel HB400	0.67	15
	Alloyed steel HB250	0.56	10
Bronze	Alloyed steel HB400	0.49	8
	Alloyed steel HB250	0.42	5
Graphite flake high strength casting	Same material as Worm wheel but with higher hardness.	0.70	5
Gray iron casting (Pearlite quality)	Phosphor bronze casting and Forging	0.63	2.5
	Same material as Worm wheel but with higher hardness.	0.42	2.5

Note (1): Values of S_{clim} in the table 7 is maximum sliding velocity applicable. Even if used below a calculated load, there is risk of sand burning if the sliding velocity exceeds this limit.

Remark 4 Calculation for Fluctuating load

(1)For combination of uniform torque with different revolving speeds,

When maximum nominal action T_{21}(1) operates Worm wheel at U_1 seconds per 1 cycle, smaller nominal torque T_{22}, $T_{23}\cdots$ at U_2, $U_3\cdots$ seconds and mean revolving speed is n_{21}, n_{22}, n_{23}, $\cdots$. calculate Equivalent time per 1 cycle based on T_{21} and n_{21} using below formula.

$$U_e = U_1 + U_2\frac{n_{22}}{n_{21}}\left(\frac{T_{22}}{T_{21}}\right)^3 + U_3\frac{n_{23}}{n_{21}}\left(\frac{T_{23}}{T_{21}}\right)^3 + \cdots\cdots \quad \text{(R1)}$$

Hereby

U_e : Equivalent time (per 1 cycle) (s) based on T_{21} and n_{21}.

$n_{21} n_{22}\ n_{23}\ \cdots$: mean revolving velocity of Worm wheel (min^{-1})

$T_{21}, T_{22}, T_{23}\ \cdots$: Nominal torque of Worm wheel (kgf・m)

Therefore Total equivalent time within 26,000 hours is as follow,

$$U_{ec} = \frac{U_e}{3600} \times (\text{Total number of cycle within 26,000 hours}) \cdots\cdots \quad \text{(R2)}$$

Hereby, Total equivalent time per 26,000 hours based on U_{ec}・T_{21} and n_{21}.

Calculate Total torque from U_{ec} and Reference table 4 using the following formula.

$$T_{2c} = T_{21}K_h' \cdots\cdots \quad \text{(R3)}$$

Hereby,

T_{2c} : Total sum of torques, T_{21}, T_{22}, $T_{23}\cdots$ (kgf・m)

K_h' : Factor taken from Reference table 4. If U_{ec} is median value, use interpolation.

Reference table 4 K_h'

U_{ee}	K_h	U_{ee}	K_h'
500 hours	0.77	5,000 hours	0.90
1,000 hours	0.79	10,000 hours	0.92
2,000 hours	0.81	25,000 hours	1.0
3,000 hours	0.84	26,000 hours	1.0

Note (1) : This table does not include torque peak with instantaneous change. Please use calculation formula from (2) for such types of torque peak.

Remark: When 1 cycle of the fluctuating load exactly matches one revolution of a Worm wheel, the largest torque always fall on only 1 specific tooth of the Worm wheel. Therefore calculation formula for fluctuating load is not applied. Calculated maximum torque is applied continuously to the whole expected lifespan.

Determine dimensions of Worm gear based on calculated Total torque T_{2c} from formula (R3) from (a) and (b).

(a) Non impact, expected lifespan is 26,000 hours. It is considered non impact if number of starts per hour is under 2 times and starting impact torque is below 200% of rated torque.

Detemine dimensions for worm gear pair in accordance with following relation.

$$T_{2c} \leqq T_{2\lim} \quad \cdots\cdots (R4)$$

Hereby

$T_{2\lim}$: Allowable torque for Worm wheel (kgf・m) to match with revolution velocity n_{21} for Worm wheel.

(b) When life is about 26,000 hours, impact conditions and number of start is above 2 times per hour. Design dimensions for Worm gear pair to form following relation.

$$T_{2c} K_h K_s \leqq T_{2\lim} \quad \cdots\cdots (R5)$$

Hereby $T_{2\lim}$: Allowable torque for Worm wheel (kgf•m) to match with revolution velocity n_{21} for Worm wheel.

(2) For combination of Peak torque and Flat torque when starting {Refer to 5.9 number(2)}.

Value of peak T_{21} during start reaches steady speed of operation after acceleration time of U_a seconds. If constant driving and torque are designated as T_{22}, Equivalent action time U_{1e} (s) is using following calculation.

$$U_{1e} = \frac{U_a}{4}\left(1+\frac{T_{22}}{T_{21}}\right)\left\{1+\left(\frac{T_{22}}{T_{21}}\right)^2\right\} \quad \cdots\cdots (R6)$$

Calculation of U_{1e}'(Torque peak equivalent action time per hour) with N times of start per hour is

$$U_{1e}' = NU_{1e} \quad \cdots\cdots (R7)$$

Actual time is NU_{1e}.

When such peak torque acts NU_a seconds per hour, steady torque T_{22} and Uniform torque T_{23},T_{24}···acts for U_2, U_3, U_4 seconds. When each mean revolution velocity is n_{21}, n_{22}, n_{23}, n_{24}, calculation of Equivalent time U_e(s) per hour is by following formula,

$$U_e = U_{1e}' + U_2\frac{n_{22}}{n_{21}}\left(\frac{T_{22}}{T_{21}}\right)^3 + U_3\frac{n_{23}}{n_{21}}\left(\frac{T_{23}}{T_{21}}\right)^3 + \quad \cdots\cdots (R8)$$

However, standard revolving speed n_{21} is the average value of peak torque between starting and end. Therefore, from standstill to reach n_{21} is calculated by $n_{21} = n'_{21} / 2$. T_{21} is standard torque. (Refer to Reference Fig. 1)

Total Virtual time in 26,000 hours is as follows.

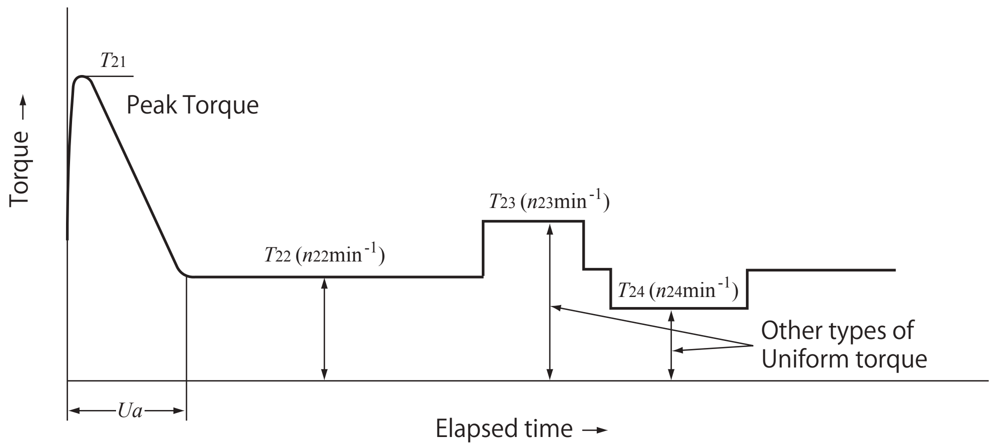
$$U_{ec} = \frac{U_e}{3,600} \times 26000 \quad \cdots\cdots (R9)$$

Hereby

U_{ec} : Total equivalent time (h) per 26,000 hours based on T_{21} and n_{21}.

This U_{ec} is equivalent to U_{ec} of formula (R2) of previous item (1). Dimensions of Worm gear pair can be determined from formula (R3), (R4) or (R5) of (1) but K_s to be 1.0.

Reference Fig. 1 Conditions of Peak and Uniform torque



Reference data

Conversion table for SI units (International System of Units)

Force	N	dyn	kgf
	1	1×10^{5}	$1.019\,72 \times 10^{-1}$
	1×10^{-5}	1	$1.019\,72 \times 10^{-6}$
	9.806 65	$9.806\,65 \times 10^{5}$	1

Pressure	Pa	bar	kgf/cm^2	atm	mmH_2O	mmHg or Torr
	1	1×10^{-5}	$1.019\,72 \times 10^{-5}$	$9.869\,23 \times 10^{-6}$	$1.019\,72 \times 10^{-1}$	$7.500\,62 \times 10^{-3}$
	1×10^{5}	1	1.019 72	$9.869\,23 \times 10^{-1}$	$1.019\,72 \times 10^{4}$	$7.500\,62 \times 10^{2}$
	$9.806\,65 \times 10^{4}$	$9.806\,65 \times 10^{-1}$	1	$9.678\,41 \times 10^{-1}$	1×10^{4}	$7.355\,59 \times 10^{2}$
	$1.013\,25 \times 10^{5}$	1.013 25	1.033 23	1	$1.033\,23 \times 10^{4}$	$7.600\,00 \times 10^{2}$
	9.806 65	$9.806\,65 \times 10^{-5}$	1×10^{-4}	$9.678\,41 \times 10^{-5}$	1	$7.355\,59 \times 10^{-2}$
	$1.333\,22 \times 10^{2}$	$1.333\,22 \times 10^{-3}$	$1.359\,51 \times 10^{-3}$	$1.315\,79 \times 10^{-3}$	$1.359\,51 \times 10$	1

Note IPa=IN/m^2

Stress	Pa	Mpa or N/mm^2	kfg/mm^2	kgf/cm^2
	1	1×10^{-6}	$1.019\,72 \times 10^{-7}$	$1.019\,72 \times 10^{-5}$
	1×10^{6}	1	$1.019\,72 \times 10^{-1}$	$1.019\,72 \times 10$
	$9.806\,65 \times 10^{6}$	9.806 65	1	1×10^{2}
	$9.806\,65 \times 10^{4}$	$9.806\,65 \times 10^{-2}$	1×10^{-2}	1

Coefficient of viscosity	Pa•s	cP	P
	1	1×10^{3}	1×10
	1×10^{-3}	1	1×10^{-2}
	1×10^{-1}	1×10^{2}	1

Note IP = Idyn•s/cm^2 = Ig/cm•S,
IPa•s = IN•s/m^2, IcP = ImPa•s

Hardness conversion table

Approximate conversion values compared with Vickers hardness of Steel

Vickers hardness	Brinell hardness 10 mm ball 3000kgf			Rockwell hardness [2]				Rockwell superficial hardness diamond cone penetrator			Shore hardness	Tensile strength (Approx. value) MPa (kgf/mm²) [1]	Vickers hardness Load
	Standard ball	Hult-gren ball	Tungsten carbide ball	Scale A Load 60kgf Diamond cone penetrator	Scale B Load 100kgf 1/16 inch Ball	Scale C Load 150kgf Diamond cone penetrator	Scale D Load 100kgf Diamond cone penetrator	15-N Scale Load 15 kgf	30-N Scale Load 30 kgf	45-N Scale Load 45 kgf			
940	-	-	-	85·6	-	68·0	76·9	93·2	84·4	75·4	97	-	940
920	-	-	-	85·3	-	67·5	76·5	93·0	84·0	74·8	96	-	920
900	-	-	-	85·0	-	67·0	76·1	92·9	83·6	74·2	95	-	900
880	-	-	(767)	84·7	-	66·4	75·7	92·7	83·1	73·6	93	-	880
860	-	-	(757)	84·4	-	65·9	75·3	92·5	82·7	73·1	92	-	860
840	-	-	(745)	84·1	-	65·3	74·8	92·3	82·2	72·2	91	-	840
820	-	-	(733)	83·8	-	64·7	74·3	92·1	81·7	71·8	90	-	820
800	-	-	(722)	83·4	-	64·0	73·8	91·8	81·1	71·0	88	-	800
780	-	-	(710)	83·0	-	63·3	73·3	91·5	80·4	70·2	87	-	780
760	-	-	(698)	82·6	-	62·5	72·6	91·2	79·7	69·4	86	-	760
740	-	-	(684)	82·2	-	61·8	72·1	91·0	79·1	68·6	84	-	740
720	-	-	(670)	81·8	-	61·0	71·5	90·7	78·4	67·7	83	-	720
700	-	615	(656)	81·3	-	60·1	70·8	90·3	77·6	66·7	81	-	700
690	-	610	(647)	81·1	-	59·7	70·5	90·1	77·2	66·2	-	-	690
680	-	603	(638)	80·8	-	59·2	70·1	89·8	76·8	65·7	80	-	680
670	-	597	630	80·6	-	58·8	69·8	89·7	76·4	65·3	-	-	670
660	-	590	620	80·3	-	58·3	69·4	89·5	75·9	64·7	79	-	660
650	-	585	611	80·0	-	57·8	69·0	89·2	75·5	64·1	-	-	650
640	-	578	601	79·8	-	57·3	68·7	89·0	75·1	63·5	77	-	640
630	-	571	591	79·5	-	56·8	68·3	88·8	74·6	63·0	-	-	630
620	-	564	582	79·2	-	56·3	67·9	88·5	74·2	62·4	75	-	620
610	-	557	573	78·9	-	55·7	67·5	88·2	73·6	61·7	-	-	610
600	-	550	564	78·6	-	55·2	67·0	88·0	73·2	61·2	74	-	600
590	-	542	554	78·4	-	54·7	66·7	87·8	72·7	60·5	-	2055 (210)	590
580	-	535	545	78·0	-	54·1	66·2	87·5	72·1	59·9	72	2020 (206)	580
570	-	527	535	77·8	-	53·6	65·8	87·2	71·7	59·3	-	1985 (202)	570
560	-	519	525	77·4	-	53·0	65·4	86·9	71·2	58·6	71	1950 (199)	560
550	(505)	512	517	77·0	-	52·3	64·8	86·6	70·5	57·8	-	1905 (194)	550
540	(496)	503	507	76·7	-	51·7	64·4	86·3	70·0	57·0	69	1860 (190)	540
530	(488)	495	497	76·4	-	51·1	63·9	86·0	69·5	56·2	-	1825 (186)	530
520	(480)	487	488	76·1	-	50·5	63·5	85·7	69·0	55·6	67	1795 (183)	520
510	(473)	479	479	75·7	-	49·8	62·9	85·4	68·3	54·7	-	1750 (179)	510
500	(465)	471	471	75·3	-	49·1	62·2	85·0	67·7	53·9	66	1705 (174)	500
490	(456)	460	460	74·9	-	48·4	61·6	84·7	67·1	53·1	-	1660 (169)	490
480	448	452	452	74·5	-	47·7	61·3	84·3	66·4	52·2	64	1620 (165)	480
470	441	442	442	74·1	-	46·9	60·7	83·9	65·7	51·3	-	1570 (160)	470
460	433	433	433	73·6	-	46·1	60·1	83·6	64·9	50·4	62	1530 (156)	460
450	425	425	425	73·3	-	45·3	59·4	83·2	64·3	49·4	-	1495 (153)	450
440	415	415	415	72·8	-	44·5	58·8	82·8	63·5	48·4	59	1460 (149)	440
430	405	405	405	72·3	-	43·6	58·2	82·3	62·7	47·4	-	1410 (144)	430
420	397	397	397	71·8	-	42·7	57·5	81·8	61·9	46·4	57	1370 (140)	420
410	388	388	388	71·4	-	41·8	56·8	81·4	61·1	45·3	-	1330 (136)	410
400	379	379	379	70·8	-	40·8	56·0	81·0	60·2	44·1	55	1290 (131)	400
390	369	369	369	70·3	-	39·8	55·2	80·3	59·3	42·9	-	1240 (127)	390
380	360	360	380	69·8	(110·0)	38·8	54·4	79·8	58·4	41·7	52	1205 (123)	380
370	350	350	350	69·2	-	37·7	53·6	79·2	57·4	40·4	-	1170 (120)	370
360	341	341	341	68·7	(109·0)	36·6	52·8	78·6	56·4	39·1	50	1130 (115)	360
350	331	331	331	68·1	-	35·5	51·9	78·0	55·4	37·8	-	1095 (112)	350
340	322	322	322	67·6	(108·0)	34·4	51·1	77·4	54·4	36·5	47	1070 (109)	340
330	313	313	313	67·0	-	33·3	50·2	76·8	53·6	35·2	-	1035 (105)	330

Approximate conversion values compared with Vickers hardness for Steel

Vickers hardness	Brinell hardness 10 mm ball 3000kgf			Rockwell hardness (2)				Rockwell superficial hardness diamond cone penetrator			Shore hardness	Tensile strength (Approx. value) MPa (kgf/mm²) (1)	Vickers hardness Load
	Standard ball	Hult-gren ball	Tungsten carbide ball	Scale A Load 60kgf Diamond cone penetrator	Scale B Load 100kgf 1/16 inch Ball	Scale C Load 150kgf Diamond cone penetrator	Scale D Load 100kgf Diamond cone penetrator	15-N Scale Load 15 kgf	30-N Scale Load 30 kgf	45-N Scale Load 45 kgf			
320	303	303	303	66·4	(107·0)	33·2	49·4	76·2	52·3	33·9	45	1005 (103)	320
310	294	294	294	65·8	-	31·0	48·4	75·6	51·3	32·5	-	980 (100)	310
300	284	284	284	65·2	(105·5)	29·8	47·5	74·9	50·2	31·1	42	950 (97)	300
295	280	280	280	64·8	-	29·2	47·1	74·6	49·7	30·4	-	935 (96)	295
290	275	275	275	64·5	(104·5)	28·5	46·5	74·2	49·0	29·5	41	915 (94)	290
285	270	270	270	64·2	-	27·8	46·0	73·8	48·4	28·7	-	905 (92)	285
280	265	265	265	63·8	(103·5)	27·1	45·3	73·4	47·8	27·9	40	890 (91)	280
275	261	261	261	63·5	-	26·4	44·9	73·0	47·2	27·1	-	875 (89)	275
270	256	256	256	63·1	(102·0)	25·6	44·3	72·6	46·4	26·2	38	855 (87)	270
265	252	252	252	62·7	-	24·8	43·7	72·1	45·7	25·2	-	840 (86)	265
260	247	247	247	62·4	(101·0)	24·0	43·1	71·6	45·0	24·3	37	825 (84)	260
255	243	243	243	62·0	-	23·1	42·2	71·1	44·2	23·2	-	805 (82)	255
250	238	238	238	61·6	99·5	22·2	41·7	70·6	43·4	22·2	36	795 (81)	250
245	233	233	233	61·2	-	21·3	41·1	70·1	42·5	21·1	-	780 (79)	245
240	228	228	228	60·7	98·1	20·3	40·3	69·6	41·7	19·9	34	765 (78)	240
230	219	219	219	-	96·7	(18·0)	-	-	-	-	33	730 (75)	230
220	209	209	209	-	95·0	(15·7)	-	-	-	-	32	695 (71)	220
210	200	200	200	-	93·4	(13·4)	-	-	-	-	30	670 (68)	210
200	190	190	190	-	91·5	(11·0)	-	-	-	-	29	635 (65)	200
190	181	181	181	-	89·5	(8·5)	-	-	-	-	28	605 (62)	190
180	171	171	171	-	87·1	(6·0)	-	-	-	-	26	580 (59)	180
170	162	162	162	-	85·0	(3·0)	-	-	-	-	25	545 (56)	170
160	152	152	152	-	81·7	(0·0)	-	-	-	-	24	515 (53)	160
150	143	143	143	-	78·7	-	-	-	-	-	22	490 (50)	150
140	133	133	133	-	75·0	-	-	-	-	-	21	455 (46)	140
130	124	124	124	-	71·2	-	-	-	-	-	20	425 (44)	130
120	114	114	114	-	66·7	-	-	-	-	-	-	390 (40)	120
110	105	105	105	-	62·3	-	-	-	-	-	-	-	110
100	95	95	95	-	56·2	-	-	-	-	-	-	-	100
95	90	90	90	-	52·0	-	-	-	-	-	-	-	95
90	86	86	86	-	48·0	-	-	-	-	-	-	-	90
85	81	81	81	-	41·0	-	-	-	-	-	-	-	85

Remark : Bold figure indicates values from Table 1 of ASTM E 140. (SAE-ASM-ASTM combined and adjusted)

Note : (1) Units and Numerical values in brackets ( ) are converted from psi conversion table of JIS Z 8438 with 1MPa = 1N/ mm²

(2) Figures in brackets ( ) from table are seldom used and mainly for reference only.

(3) Iron and Steel quoted from JIS hand book

Approximate converted values compared with Rockwell hardness for Steel (1)

Rockwell Scale C hardness	Vickers hardness	Brinell hardness 10 mm ball 3000kgf			Rockwell hardness (2)			Rockwell superficial hardness diamond cone penetrator			Shore hardness	Tensile strength (Approx. value) MPa (kgf/mm²) (1)	Rockwell Scale C hardness
		Standard ball	Hult-gren ball	Tungsten carbide ball	Scale A Load 60kgf Diamond cone penetrator	Scale B Load 100kgf 1/16 inch Ball	Scale D Load 100kgf Diamond cone penetrator	15-N Scale Load 15 kgf	30-N Scale Load 30 kgf	45-N Scale Load 45 kgf			
68	940	-	-	-	85·6	-	76·9	93·2	84·4	75·4	97	-	68
67	900	-	-	-	85·0	-	76·1	92·9	83·6	74·2	95	-	67
66	865	-	-	-	84·5	-	75·4	92·5	82·8	73·3	92	-	66
65	832	-	-	(739)	83·9	-	74·5	92·2	81·9	72·0	91	-	65
64	800	-	-	(722)	83·4	-	73·8	91·8	81·1	71·0	88	-	64
63	772	-	-	(705)	82·8	-	73·0	91·4	80·1	69·9	87	-	63
62	746	-	-	(688)	82·3	-	72·2	91·1	79·3	68·8	85	-	62
61	720	-	-	(670)	81·8	-	71·5	90·7	78·4	67·7	83	-	61
60	697	-	613	(654)	81·2	-	70·7	90·2	77·5	66·6	81	-	60
59	674	-	599	(634)	80·7	-	69·9	89·8	76·6	65·5	80	-	59
58	653	-	587	615	80·1	-	69·2	89·3	75·7	64·3	78	-	58
57	633	-	575	595	79·6	-	68·5	88·9	74·8	63·2	76	-	57
56	613	-	561	577	79·0	-	67·7	88·3	73·9	62·0	75	-	56
55	595	-	546	560	78·5	-	66·9	87·9	73·0	60·9	74	2075 (212)	55
54	577	-	534	543	78·0	-	66·1	87·4	72·0	59·8	72	2015 (205)	54
53	560	-	519	525	77·4	-	65·4	86·9	71·2	58·6	71	1950 (199)	53
52	544	(500)	508	512	76·8	-	64·6	86·4	70·2	57·4	69	1880 (192)	52
51	528	(487)	494	496	76·3	-	63·8	85·9	69·4	56·1	68	1820 (186)	51
50	513	(475)	481	481	75·9	-	63·1	85·5	68·5	55·0	67	1760 (179)	50
49	498	(464)	469	469	75·2	-	62·1	85·0	67·6	53·8	66	1695 (173)	49
48	484	451	455	455	74·7	-	61·4	84·5	66·7	52·5	64	1635 (167)	48
47	471	442	443	443	74·1	-	60·8	83·9	65·8	51·4	63	1580 (161)	47
46	458	432	432	432	73·6	-	60·0	83·5	64·8	50·3	62	1530 (156)	46
45	446	421	421	421	73·1	-	59·2	83·0	64·0	49·0	60	1480 (151)	45
44	434	409	409	409	72·5	-	58·5	82·5	63·1	47·8	58	1435 (146)	44
43	423	400	400	400	72·0	-	57·7	82·0	62·2	46·7	57	1385 (141)	43
42	412	390	390	390	71·5	-	56·9	81·5	61·3	45·5	56	1340 (136)	42
41	402	381	381	381	70·9	-	56·2	80·9	60·4	44·3	55	1295 (132)	41
40	392	371	371	371	70·4	-	55·4	80·4	59·5	43·1	54	1250 (127)	40
39	382	362	362	362	69·9	-	54·6	79·9	58·6	41·9	52	1215 (124)	39
38	372	353	353	353	69·4	-	53·8	79·4	57·7	40·8	51	1180 (120)	38
37	363	344	344	344	68·9	-	53·1	78·8	56·8	39·6	50	1160 (118)	37
36	354	336	336	336	68·4	(109·0)	52·3	78·3	55·9	38·4	49	1115 (114)	36
35	345	327	327	327	67·9	(108·5)	51·5	77·7	55·0	37·2	48	1080 (110)	35
34	336	319	319	319	67·4	(108·0)	50·8	77·2	54·2	36·1	47	1055 (108)	34
33	327	311	311	311	66·8	(107·5)	50·0	76·6	53·3	34·9	46	1025 (105)	33
32	318	301	301	301	66·3	(107·0)	49·2	76·1	52·1	33·7	44	1000 (102)	32
31	310	294	294	294	65·8	(106·0)	48·4	75·6	51·3	32·5	43	980 (100)	31
30	302	286	286	286	65·3	(105·5)	47·7	75·0	50·4	31·3	42	950 (97)	30
29	294	279	279	279	64·7	(104·5)	47·0	74·5	49·5	30·1	41	930 (95)	29

Approximate converted values compared with Rockwell hardness for Steel (1)

Rockwell Scale C hardness	Vickers hardness	Brinell hardness 10 mm ball 3000kgf			Rockwell hardness (2)			Rockwell superficial hardness diamond cone penetrator			Shore hardness	Tensile strength (Approx. value) MPa (kgf/mm^2) (1)	Rockwell Scale C hardness
		Standard ball	Hult-gren ball	Tungsten carbide ball	Scale A Load 60kgf Diamond cone penetrator	Scale B Load 100kgf 1/16 inch Ball	Scale D Load 100kgf Diamond cone penetrator	15-N Scale Load 15 kgf	30-N Scale Load 30 kgf	45-N Scale Load 45 kgf			
28	286	271	271	271	64·3	(104·0)	46·1	73·9	48·6	28·9	41	910 (93)	28
27	279	264	264	264	63·8	(103·0)	45·2	73·3	47·7	27·8	40	880 (90)	27
26	272	258	258	258	63·3	(102·5)	44·6	72·8	46·8	26·7	38	860 (88)	26
25	266	253	253	253	62·8	(101·5)	43·8	72·2	45·9	25·5	38	840 (86)	25
24	260	247	247	247	62·4	(101·0)	43·1	71·6	45·0	24·3	37	825 (84)	24
23	254	243	243	243	62·0	100·0	42·1	71·0	44·0	23·1	36	805 (82)	23
22	248	237	237	237	61·5	99·0	41·6	70·5	43·2	22·0	35	785 (80)	22
21	243	231	231	231	61·0	98·5	40·9	69·9	42·3	20·7	35	770 (79)	21
20	238	226	226	226	60·5	97·8	40·1	69·4	41·5	19·6	34	760 (77)	20
(18)	230	219	219	219	-	96·7	-	-	-	-	33	730 (75)	(18)
(16)	222	212	212	212	-	95·5	-	-	-	-	32	705 (72)	(16)
(14)	213	203	203	203	-	93·9	-	-	-	-	31	675 (69)	(14)
(12)	204	194	194	194	-	92·3	-	-	-	-	29	650 (66)	(12)
(10)	196	187	187	187	-	90·7	-	-	-	-	28	620 (63)	(10)
(8)	188	179	179	179	-	89·5	-	-	-	-	27	600 (61)	(8)
(6)	180	171	171	161	-	87·1	-	-	-	-	26	580 (59)	(6)
(4)	173	165	165	165	-	85·5	-	-	-	-	25	550 (56)	(4)
(2)	166	158	158	158	-	83·5	-	-	-	-	24	530 (54)	(2)
(0)	160	152	152	152	-	81·7	-	-	-	-	24	515 (53)	(0)

Note : (1) Units and Numerical values in bracket ( ) is converted from psi conversion table of JIS Z 8438 with 1Mpa = 1N/ mm^2
(2) Figures in brackets ( ) from table are seldom used and mainly for reference only.
(3) Iron and Steel quoted from JIS hand book

Commonly used fitting tolerances for bore dimensions

Unit：μm

Dimensions (mm)		B	C		D			E			F			G		H					
Above	Below	B10	C9	C10	D8	D9	D10	E7	E8	E9	F6	F7	F8	G6	G7	H6	H7	H8	H9	H10	H11
-	3	+180	+85	+100	+34	+45	+60	+24	+28	+39	+12	+16	+20	+8	+12	+6	+10	+14	+25	+40	+60
		+140	+60			+20			+14			+6		+2				0			
3	6	+188	+100	+118	+48	+60	+78	+32	+38	+50	+18	+22	+28	+12	+16	+8	+12	+18	+30	+48	+75
		+140	+70			+30			+20			+10		+4				0			
6	10	+208	+116	+138	+62	+76	+98	+40	+47	+61	+22	+28	+35	+14	+20	+9	+15	+22	+36	+58	+90
		+150	+80			+40			+25			+13		+5				0			
10	14	+220	+138	+165	+77	+93	+120	+50	+59	+75	+27	+34	+43	+17	+24	+11	+18	+27	+43	+70	+110
14	18	+150	+95			+50			+32			+16		+6				0			
18	24	+244	+162	+194	+98	+117	+149	+61	+73	+92	+33	+41	+53	+20	+28	+13	+21	+33	+52	+84	+130
24	30	+160	+110			+65			+40			+20		+7				0			
30	40	+270	+182	+220																	
		+170	+120		+119	+142	+180	+75	+89	+112	+41	+50	+64	+25	+34	+16	+25	+39	+62	+100	+160
40	50	+280	+192	+230		+80			+50			+25		+9				0			
		+180	+130																		
50	65	+310	+214	+260																	
		+190	+140		+146	+174	+220	+90	+106	+134	+49	+60	+76	+29	+40	+19	+30	+46	+74	+120	+190
65	80	+320	+224	+270		+100			+60			+30		+10				0			
		+200	+150																		
80	100	+360	+257	+310																	
		+220	+170		+174	+207	+260	+107	+126	+159	+58	+71	+90	+34	+47	+22	+35	+54	+87	+140	+220
100	120	+380	+267	+320		+120			+72			+36		+12							0
		+240	+180																		
120	140	+420	+300	+360																	
		+260	+200																		
140	160	+440	+310	+370	+208	+245	+305	+125	+148	+185	+68	+83	+106	+39	+54	+25	+40	+63	+100	+160	+250
		+280	+210			+145			+85			+43		+14				0			
160	180	+470	+330	+390																	
		+310	+230																		
180	200	+525	+355	+425																	
		+340	+240																		
200	225	+565	+375	+445	+242	+285	+355	+146	+172	+215	+79	+96	+122	+44	+61	+29	+46	+72	+115	+185	+290
		+380	+260			+170			+100			+50		+15				0			
225	250	+605	+395	+465																	
		+420	+280																		
250	280	+690	+430	+510																	
		+480	+300		+271	+320	+400	+162	+191	+240	+88	+108	+137	+49	+69	+32	+52	+81	+130	+210	+320
280	315	+750	+460	+540		+190			+110			+56		+17				0			
		+540	+330																		
315	355	+830	+500	+590																	
		+600	+360		+299	+350	+440	+182	+214	+265	+98	+119	+151	+54	+75	+36	+57	+89	+140	+230	+360
355	400	+910	+540	+630		+210			+125			+62		+18				0			
		+680	+400																		
400	450	+1010	+595	+690																	
		+760	+440		+327	+385	+480	+198	+232	+290	+108	+131	+165	+60	+83	+40	+63	+97	+155	+250	+400
450	500	+1090	+635	+730		+230			+135			+68		+20				0			
		+840	+480																		

Remark : For each cell in the table above, values in the top row shows upper limit of tolerance and values in the bottom row shows lower limit of tolerance.

Commonly used fitting tolerances for bore dimensions

Unit : μm

Dimensions (mm)		Js				K		M		N		P		R	S	T	U	X
Above	Below	Js6	Js7	Js8	Js9	K6	K7	M6	M7	N6	N7	P6	P7	R7	S7	T7	U7	X7
-	3	± 3	± 5	± 7	± 12.5	0 -6	0 -10	-2 -8	-2 -12	-4 -10	-4 -14	-6 -12	-6 -16	-10 -20	-14 -24	-	-18 -28	-20 -30
3	6	± 4	± 6	± 9	± 15	+2 -6	+3 -9	-1 -9	0 -12	-5 -13	-4 -16	-9 -17	-8 -20	-11 -23	-15 -27	-	-19 -31	-24 -36
6	10	± 4.5	± 7.5	± 11	± 18	+2 -7	+5 -10	-3 -12	0 -15	-7 -16	-4 -19	-12 -21	-9 -24	-13 -28	-17 -32	-	-22 -37	-28 -43
10	14	± 5.5	± 9	± 13.5	± 21.5	+2 -9	+6 -12	-4 -15	0 -18	-9 -20	-5 -23	-15 -26	-11 -29	-16 -34	-21 -39	-	-26 -44	-33 -51
14	18																	-38 -56
18	24	± 6.5	± 10.5	± 16.5	± 26	+2 -11	+6 -15	-4 -17	0 -21	-11 -24	-7 -28	-18 -31	-14 -35	-20 -41	-27 -48	-	-33 -54	-46 -67
24	30															-33 -54	-40 -61	-56 -77
30	40	± 8	± 12.5	± 19.5	± 31	+3 -13	+7 -18	-4 -20	0 -25	-12 -28	-8 -33	-21 -37	-17 -42	-25 -50	-31 -59	-39 -64	-51 -76	-
40	50															-45 -70	-61 -86	
50	65	± 9.5	± 15	± 23	± 37	+4 -15	+9 -21	-5 -24	0 -30	-14 -33	-9 -39	-26 -45	-21 -51	-30 -60	-42 -72	-55 -85	-76 -106	-
65	80													-32 -62	-48 -78	-64 -94	-91 -121	
80	100	± 11	± 17.5	± 27	± 43.5	+4 -18	+10 -25	-6 -28	0 -35	-16 -38	-10 -45	-30 -52	-21 -59	-38 -73	-58 -93	-78 -113	-111 -146	-
100	120													-41 -76	-66 -101	-91 -126	-131 -166	
120	140	± 12.5	± 20	± 31.5	± 50	+4 -21	+12 -28	-8 -33	0 -40	-20 -45	-12 -52	-36 -61	-28 -68	-48 -88	-77 -117	-107 -147	-	-
140	160													-50 -90	-85 -125	-119 -159		
160	180													-53 -93	-93 -133	-131 -171		
180	200	± 14.5	± 23	± 36	± 57.5	+5 -24	+13 -33	-8 -37	0 -46	-22 -51	-14 -60	-41 -70	-33 -79	-60 -106	-105 -151	-	-	-
200	225													-63 -109	-113 -159			
225	250													-67 -113	-123 -169			
250	280	± 16	± 26	± 40..5	± 65	+5 -27	+16 -36	-9 -41	0 -52	-25 -57	-14 -66	-47 -79	-36 -88	-74 -126	-	-	-	-
280	315													-78 -130				
315	355	± 18	± 28.5	± 44.5	± 70	+7 -29	+17 -40	-10 -46	0 -57	-26 -62	-16 -73	-51 -87	-41 -93	-87 -144	-	-	-	-
355	400													-93 -150				
400	450	± 20	± 31.5	± 48.5	± 77.5	+8 -32	+18 -45	-10 -50	0 -63	-27 -67	-17 -80	-55 -95	-45 -108	-103 -166	-	-	-	-
450	500													-109 -172				

Remark : For each cell in the table above, values in the top row shows upper limit of tolerance and values in the bottom row shows lower limit of tolerance.

Commonly used fitting tolerances for axis dimensions

Unit：μm

Dimensions (mm)		js				k		m		n	p	r	s	t	u	x
Above	Below	js5	js6	js7	js8	k5	k6	m5	m6	n6	p6	r6	s6	t6	u6	x6
-	3	± 2	± 3	± 5	± 7	+4 0	+6 0	+6 +2	+8 +2	+10 +4	+12 +6	+16 +10	+20 +14	-	+24 +18	+26 +20
3	6	± 2.5	± 4	± 6	± 9	+6 +1	+9 +1	+9 +4	+12 +4	+16 +8	+20 +12	+23 +15	+27 +19	-	+31 +23	+36 +28
6	10	± 3	± 4.5	± 7.5	± 11	+7 +1	+10 +1	+12 +6	+15 +6	+19 +10	+24 +15	+28 +19	+32 +23	-	+37 +28	+43 +34
10	14	± 4	± 5.5	± 9	± 13.5	+9 +1	+12 +1	+15 +7	+18 +7	+23 +12	+29 +18	+34 +23	+39 +28	-	+44 +33	+51 +40
14	18															+56 +45
18	24	± 4.5	± 6.5	± 10.5	± 16.5	+11 +2	+15 +2	+17 +8	+21 +8	+28 +15	+35 +22	+41 +28	+48 +35	-	+54 +41	+67 +54
24	30													+54 +41	+61 +48	+77 +64
30	40	± 5.5	± 8	± 12.5	± 19.5	+13 +2	+18 +2	+20 +9	+25 +9	+33 +17	+42 +26	+50 +34	+59 +43	+64 +48	+76 +60	-
40	50													+70 +54	+86 +70	
50	65	± 6.5	± 9.5	± 15	± 23	+15 +2	+21 +2	+24 +11	+30 +11	+30 +20	+51 +32	+60 +41	+72 +53	+85 +66	+106 +87	-
65	80											+62 +43	+78 +59	+94 +75	+121 +102	
80	100	± 7.5	± 11	± 17.5	± 27	+18 +3	+25 +3	+28 +13	+35 +13	+45 +23	+59 +37	+73 +51	+93 +71	+113 +104	+146 +124	-
100	120											+76 +54	+101 +79	+126 +104	+166 +144	
120	140	± 9	± 12.5	± 20	± 31.5	+21 +3	+28 +3	+33 +15	+40 +15	+52 +27	+68 +43	+88 +63	+117 +92	+147 +122	-	-
140	160											+90 +65	+125 +100	+159 +134		
160	180											+93 +68	+133 +108	+171 +146		
180	200	± 10	± 14.5	± 23	± 36	+24 +4	+33 +4	+37 +17	+46 +17	+60 +31	+79 +50	+106 +77	+151 +122	-	-	-
200	225											+109 +80	+159 +130			
225	250											+113 +84	+169 +140			
250	280	± 11.5	± 16	± 26	± 40.5	+27 +4	+36 +4	+43 +20	+52 +20	+66 +34	+88 +56	+126 +94	-	-	-	-
280	315											+130 +98				
315	355	± 12.5	± 18	± 28.5	± 44.5	+29 +4	+40 +4	+46 +21	+57 +21	+73 +37	+98 +62	+144 +108	-	-	-	-
355	400											+150 +114				
400	450	± 13.5	± 20	± 31.5	± 48.5	+32 +5	+45 +5	+50 +23	+63 +23	+80 +40	+108 +68	+166 +126	-	-	-	-
450	500											+172 +132				

Remark : For each cell in the table above, values in the top row shows upper limit of tolerance and values in the bottom row shows lower limit of tolerance.

Commonly used fitting tolerances for axis dimensions

Unit：μm

Dimensions (mm)		b	c	d		e			f			g		h						
Above	Below	b9	c9	d8	d9	e7	e8	e9	f6	f7	f8	g5	g6	h5	h6	h7	h8	h9	h10	h11
-	3	-140	-60	-20			-14			-6		-2					0			
		-165	-85	-34	-45	-24	-28	-39	-12	-16	-20	-6	-8	-4	-6	-10	-14	-25	-40	-60
3	6	-140	-70	-30			-20			-10		-4					0			
		-170	-100	-48	-60	-32	-38	-50	-18	-22	-28	-9	-12	-5	-8	-12	-18	-30	-48	-75
6	10	-150	-80	-40			-25			-13		-5					0			
		-186	-116	-62	-76	-40	-47	-61	-22	-28	-35	-11	-14	-6	-9	-15	-22	-36	-58	-90
10	14	-150	-95	-50			-32			-16		-6					0			
14	18	-193	-138	-77	-93	-50	-59	-75	-27	-34	-43	-14	-17	-8	-11	-18	-27	-43	-70	-110
18	24	-160	-110	-65			-40			-20		-7					0			
24	30	-212	-162	-98	-117	-61	-73	-92	-33	-41	-53	-16	-20	-9	-13	-21	-33	-52	-84	-130
30	40	-170	-120																	
		-232	-182	-80			-50			-25		-9					0			
40	50	-180	-130	-119	-142	-75	-89	-112	-41	-50	-64	-20	-25	-11	-16	-25	-39	-62	-100	-160
		-242	-192																	
50	65	-190	-140																	
		-264	-214	-100			-60			-30		-10					0			
65	80	-200	-150	-146	-174	-90	-106	-134	-49	-60	-76	-23	-29	-13	-19	-30	-46	-74	-120	-190
		-274	-224																	
80	100	-220	-170																	
		-307	-257	-120			-72			-36		-12					0			
100	120	-240	-180	-174	-207	-107	-126	-159	-58	-71	-90	-27	-34	-15	-22	-35	-54	-87	-140	-220
		-327	-267																	
120	140	-260	-200																	
		-360	-300																	
140	160	-280	-210	-145			-85			-43		-14					0			
		-380	-310	-208	-245	-125	-148	-185	-68	-83	-106	-32	-39	-18	-25	-40	-63	-100	-160	-250
160	180	-310	-230																	
		-410	-330																	
180	200	-340	-240																	
		-455	-355																	
200	225	-380	-260	-170			-100			-50		-15					0			
		-495	-375	-242	-285	-146	-172	-215	-79	-96	-122	-35	-44	-20	-29	-46	-72	-115	-185	-290
225	250	-420	-280																	
		-535	-395																	
250	280	-480	-300																	
		-610	-430	-190			-110			-56		-17					0			
280	315	-540	-330	-271	-320	-162	-191	-240	-88	-108	-137	-40	-49	-23	-32	-52	-81	-130	-210	-320
		-670	-460																	
315	355	-600	-360																	
		-710	-500	-210			-125			-62		-18					0			
355	400	-680	-400	-299	-350	-182	-214	-265	-98	-119	-151	-43	-54	-25	-36	-57	-89	-140	-230	-260
		-820	-540																	
400	450	-760	-440																	
		-915	-595	-230			-135			-68		-20					0			
450	500	-840	-480	-327	-385	-198	-232	-290	-108	-131	-165	-47	-60	-27	-40	-63	-97	-155	-250	-400
		-995	-635																	

Remark : For each cell in the table above, values in the top row shows upper limit of tolerance and values in the bottom row shows lower limit of tolerance.

Involute function ①

α' \ α°	14		15		16		17		18	
	inv α	Differ	inv α	Differ	inv α	Differ	inv α	Differ	inv α	Differ
0	.004 981 9	18 1	.006 149 8	20 9	.007 492 7	23 9	.009 024 7	27 2	.010 760 4	30 8
1	.005 000 0	18 2	.006 170 7	21 0	.007 516 6	24 0	.009 051 9	27 3	.010 791 2	30 8
2	.005 018 2	18 2	.006 191 7	21 0	.007 540 6	24 1	.009 079 2	27 3	.010 822 0	30 8
3	.005 036 4	18 2	.006 212 7	21 0	.007 564 7	24 1	.009 106 5	27 4	.010 852 8	31 0
4	.005 054 6	18 3	.006 233 7	21 1	.007 588 8	24 2	.009 133 9	27 5	.010 883 8	30 9
5	.005 072 9	18 3	.006 254 8	21 2	.007 613 0	24 2	.009 161 4	27 5	.010 914 7	31 1
6	.005 091 2	18 4	.006 276 0	21 2	.007 637 2	24 2	.009 188 9	27 5	.010 945 8	31 1
7	.005 109 6	18 4	.006 297 2	21 2	.007 661 4	24 3	.009 216 4	27 6	.010 976 9	31 2
8	.005 128 0	18 5	.006 318 4	21 3	.007 685 7	24 4	.009 244 0	27 7	.011 008 1	31 2
9	.005 146 5	18 5	.006 339 7	21 4	.007 710 1	24 4	.009 271 7	27 7	.011 039 3	31 3
10	.005 165 0	18 5	.006 361 1	21 4	.007 734 5	24 5	.009 299 4	27 8	.011 070 6	31 3
11	.005 183 5	18 6	.006 382 5	21 4	.007 759 0	24 5	.009 327 2	27 9	.011 101 9	31 4
12	.005 202 1	18 7	.006 403 9	21 5	.007 783 5	24 6	.009 355 1	27 9	.011 133 3	31 5
13	.005 220 8	18 7	.006 425 4	21 6	.007 808 1	24 6	.009 383 0	27 9	.011 164 8	31 6
14	.005 239 5	18 7	.006 447 0	21 6	.007 832 7	24 7	.009 410 9	28 1	.011 196 4	31 6
15	.005 258 2	18 8	.006 468 6	21 6	.007 857 4	24 8	.009 439 0	28 0	.011 228 0	31 6
16	.005 277 0	18 8	.006 490 2	21 7	.007 882 2	24 7	.009 467 0	28 2	.011 259 6	31 7
17	.005 295 8	18 9	.006 511 9	21 8	.007 906 9	24 9	.009 495 2	28 2	.011 291 3	31 8
18	.005 314 7	18 9	.006 533 7	21 8	.007 931 8	24 9	.009 523 4	28 2	.011 323 1	31 9
19	.005 333 6	19 0	.006 555 5	21 8	.007 956 7	25 0	.009 551 6	28 3	.011 355 0	31 9
20	.005 352 6	19 0	.006 577 3	21 9	.007 981 7	25 0	.009 579 9	28 4	.011 386 9	32 0
21	.005 371 6	19 1	.006 599 2	21 9	.008 006 7	25 0	.009 608 3	28 4	.011 418 9	32 0
22	.005 390 7	19 1	.006 621 1	22 0	.008 031 7	25 1	.009 636 7	28 5	.011 450 9	32 1
23	.005 409 8	19 1	.006 643 1	22 1	.008 056 8	25 2	.009 665 2	28 5	.011 483 0	32 1
24	.005 428 9	19 2	.006 665 2	22 1	.008 082 0	25 2	.009 693 7	28 6	.011 515 1	32 3
25	.005 448 1	19 3	.006 687 3	22 1	.008 107 2	25 3	.009 722 3	28 7	.011 547 4	32 2
26	.005 467 4	19 3	.006 709 4	22 2	.008 132 5	25 3	.009 751 0	28 7	.011 579 6	32 4
27	.005 486 7	19 3	.006 731 6	22 3	.008 157 8	25 4	.009 779 7	28 8	.011 612 0	32 4
28	.005 506 0	19 4	.006 753 9	22 3	.008 183 2	25 5	.009 808 5	28 8	.011 644 4	32 5
29	.005 525 4	19 4	.006 776 2	22 3	.008 208 7	25 5	.009 837 3	28 9	.011 676 9	32 5
30	.005 544 8	19 5	.006 798 5	22 4	.008 234 2	25 5	.009 866 2	28 9	.011 709 4	32 6
31	.005 564 3	19 5	.006 820 9	22 5	.008 259 7	25 6	.009 895 1	29 0	.011 742 0	32 7
32	.005 583 8	19 6	.006 843 4	22 5	.008 285 3	25 7	.009 924 1	29 1	.011 774 7	32 7
33	.005 603 4	19 6	.006 865 9	22 5	.008 311 0	25 7	.009 953 2	29 1	.011 807 4	32 8
34	.005 623 0	19 7	.006 888 4	22 6	.008 336 7	25 8	.009 982 3	29 2	.011 840 2	32 8
35	.005 642 7	19 7	.006 911 0	22 7	.008 362 5	25 8	.010 011 5	29 2	.011 873 0	32 9
36	.005 662 4	19 8	.006 933 7	22 7	.008 388 3	25 9	.010 040 7	29 3	.011 905 9	33 0
37	.005 682 2	19 8	.006 956 4	22 7	.008 414 2	25 9	.010 070 0	29 4	.011 938 9	33 1
38	.005 702 0	19 8	.006 979 1	22 8	.008 440 1	26 0	.010 099 4	29 4	.011 902 0	33 1
39	.005 721 8	19 9	.007 001 9	22 9	.008 466 1	26 0	.010 128 8	29 5	.012 005 1	33 1
40	.005 741 7	20 0	.007 024 8	22 9	.008 492 1	26 1	.010 158 3	29 5	.012 038 2	33 3
41	.005 761 7	20 0	.007 047 7	22 9	.008 518 2	26 2	.010 187 8	29 6	.012 071 5	33 3
42	.005 781 7	20 0	.007 070 6	23 0	.008 544 4	26 2	.010 217 4	29 7	.012 104 8	33 3
43	.005 801 7	20 1	.007 093 6	23 1	.008 570 6	26 3	.010 247 1	29 7	.012 138 1	33 4
44	.005 821 8	20 2	.007 116 7	23 1	.008 596 9	26 3	.010 276 8	29 8	.012 171 5	33 5
45	.005 842 0	20 2	.007 139 8	23 2	.008 623 2	26 4	.010 306 6	29 8	.012 205 0	33 6
46	.005 862 2	20 2	.007 163 0	23 2	.008 649 6	26 4	.010 336 4	29 9	.012 238 6	33 6
47	.005 882 4	20 3	.007 186 2	23 3	.008 676 0	26 5	.010 366 3	30 0	.012 272 2	33 7
48	.005 902 7	20 3	.007 209 5	23 3	.008 702 5	26 5	.010 396 3	30 0	.012 305 9	33 7
49	.005 923 0	20 4	.007 232 8	23 3	.008 729 0	26 6	.010 426 3	30 1	.012 339 6	33 8
50	.005 943 4	20 4	.007 256 1	23 5	.008 755 6	26 7	.010 456 4	30 1	.012 373 4	33 9
51	.005 963 8	20 5	.007 279 6	23 4	.008 782 3	26 7	.010 486 5	30 2	.012 407 3	33 9
52	.005 984 3	20 5	.007 303 0	23 6	.008 809 0	26 8	.010 516 7	30 2	.012 441 2	34 0
53	.006 004 8	20 6	.007 326 6	23 5	.008 835 8	26 8	.010 546 9	30 4	.012 475 2	34 1
54	.006 025 4	20 6	.007 350 1	23 7	.008 862 6	26 9	.010 577 3	30 3	.012 509 3	34 1
55	.006 046 0	20 7	.007 373 8	23 7	.008 889 5	26 9	.010 607 6	30 5	.012 543 4	34 2
56	.006 066 7	20 7	.007 397 5	23 7	.008 916 4	27 0	.010 638 1	30 5	.012 577 6	34 3
57	.006 087 4	20 7	.007 421 2	23 8	.008 943 4	27 0	.010 668 6	30 5	.012 611 9	34 3
58	.006 108 1	20 8	.007 445 0	23 8	.008 970 4	27 1	.010 699 1	30 7	.012 646 2	34 4
59	.006 128 9	20 9	.007 468 8	23 9	.008 997 5	27 2	.010 729 8	30 6	.012 680 6	34 5
60	.006 149 8		.007 492 7		.009 024 7		.010 760 4		.012 715 1	

Involute function ②

α′ \ α°	19 inv α	19 Differ	20 inv α	20 Differ	21 inv α	21 Differ	22 inv α	22 Differ	23 inv α	23 Differ
0	.012 715 1	34 5	.014 904 4	38 6	.017 344 9	42 9	.020 053 8	47 5	.023 049 1	52 4
1	.012 749 6	34 6	.014 943 0	38 6	.017 387 8	43 0	.020 101 3	47 6	.023 101 5	52 6
2	.012 784 2	34 6	.014 981 6	38 7	.017 430 8	43 0	.020 148 9	47 7	.023 154 1	52 6
3	.012 818 8	34 7	.015 020 3	38 8	.017 473 8	43 1	.020 196 6	47 8	.023 206 7	52 7
4	.012 853 5	34 8	.015 059 1	38 8	.017 516 9	43 2	.020 244 4	47 8	.023 259 4	52 8
5	.012 888 3	34 9	.015 097 9	39 0	.017 560 1	43 3	.020 292 2	47 9	.023 312 2	52 9
6	.012 923 2	34 9	.015 136 9	38 9	.017 603 4	43 4	.020 340 1	48 0	.023 365 1	53 0
7	.012 958 1	35 0	.015 175 8	39 1	.017 646 8	43 4	.020 388 1	48 1	.023 418 1	53 0
8	.012 993 1	35 0	.015 214 9	39 1	.017 690 2	43 5	.020 436 2	48 2	.023 471 1	53 1
9	.013 028 1	35 1	.015 254 0	39 2	.017 733 7	43 6	.020 484 4	48 2	.023 524 2	53 3
10	.013 063 2	35 2	.015 293 2	39 3	.017 777 3	43 6	.020 532 6	48 3	.023 577 5	53 3
11	.013 098 4	35 2	.015 322 5	39 4	.017 820 9	43 7	.020 580 9	48 4	.023 630 8	53 4
12	.013 133 6	35 3	.015 371 9	39 4	.017 864 6	43 8	.020 629 3	48 5	.023 684 2	53 4
13	.013 168 9	35 4	.015 411 3	39 4	.017 908 4	43 9	.020 677 8	48 6	.023 737 6	53 6
14	.013 204 3	35 5	.015 450 7	39 6	.017 952 3	44 0	.020 726 4	48 6	.023 791 2	53 7
15	.013 239 8	35 5	.015 490 3	39 6	.017 996 3	44 0	.020 775 0	48 8	.023 844 9	53 7
16	.013 275 3	35 5	.015 529 9	39 7	.018 040 3	44 1	.020 823 8	48 8	.023 898 6	53 8
17	.013 310 8	35 7	.015 569 6	39 8	.018 084 4	44 2	.020 872 6	48 9	.023 952 4	53 9
18	.013 346 5	35 7	.015 609 4	39 8	.018 128 6	44 2	.020 921 5	48 9	.024 006 3	54 0
19	.013 382 2	35 8	.015 649 2	39 9	.018 172 8	44 4	.020 970 4	49 1	.024 060 3	54 1
20	.013 418 0	35 8	.015 689 1	40 0	.018 217 2	44 4	.021 019 5	49 1	.024 114 4	54 2
21	.013 453 8	35 9	.015 729 1	40 1	.018 261 6	44 5	.021 068 6	49 2	.024 168 6	54 2
22	.013 489 7	36 0	.015 769 2	40 1	.018 306 1	44 5	.021 117 8	49 3	.024 222 8	54 4
23	.013 525 7	36 0	.015 809 3	40 2	.018 350 6	44 7	.021 167 1	49 4	.024 277 2	54 4
24	.013 561 7	36 1	.015 849 5	40 3	.018 395 3	44 7	.021 216 5	49 5	.024 331 6	54 5
25	.013 597 8	36 2	.015 889 8	40 3	.018 440 0	44 8	.021 266 0	49 5	.024 386 1	54 6
26	.013 634 0	36 2	.015 930 1	40 4	.018 484 8	44 8	.021 315 5	49 6	.024 440 7	54 7
27	.013 670 2	36 3	.015 970 5	40 5	.018 529 6	45 0	.021 365 1	49 7	.024 495 4	54 8
28	.013 706 5	36 4	.016 011 0	40 6	.018 574 6	45 0	.021 414 8	49 8	.024 550 2	54 8
29	.013 742 9	36 5	.016 051 6	40 6	.018 619 6	45 1	.021 464 6	49 9	.024 605 0	55 0
30	.013 779 4	36 5	.016 092 2	40 7	.018 664 7	45 2	.021 514 5	49 9	.024 660 0	55 0
31	.013 815 9	36 6	.016 132 9	40 8	.018 709 9	45 2	.021 564 4	50 1	.024 715 0	55 2
32	.013 852 5	36 6	.016 173 7	40 8	.018 755 1	45 3	.021 614 5	50 1	.024 770 2	55 2
33	.013 889 1	36 7	.016 214 5	40 9	.018 800 4	45 4	.021 664 6	50 2	.024 825 4	55 3
34	.013 925 8	36 8	.016 255 4	41 0	.018 845 8	45 5	.021 714 8	50 3	.024 880 7	55 4
35	.013 962 6	36 8	.016 296 4	41 1	.018 891 3	45 6	.021 765 1	50 3	.024 936 1	55 5
36	.013 999 4	37 0	.016 337 5	41 1	.018 936 9	45 6	.021 815 4	50 5	.024 991 6	55 5
37	.014 036 4	37 0	.016 378 6	41 2	.018 982 5	45 7	.021 865 9	50 5	.025 047 1	55 7
38	.014 073 4	37 0	.016 419 8	41 3	.019 028 2	45 8	.021 916 4	50 6	.025 102 8	55 7
39	.014 110 4	37 1	.016 461 1	41 3	.019 074 0	45 9	.021 967 0	50 7	.025 158 5	55 8
40	.014 147 5	37 2	.016 502 4	41 5	.019 119 9	46 0	.022 017 7	50 8	.025 214 3	56 0
41	.014 184 7	37 3	.016 543 9	41 5	.019 165 9	46 0	.022 068 5	50 8	.025 270 3	56 0
42	.014 222 0	37 3	.016 585 4	41 5	.019 211 9	46 1	.022 119 3	51 0	.025 326 3	56 1
43	.014 259 3	37 4	.016 626 9	41 7	.019 258 0	46 2	.022 170 3	51 0	.025 382 4	56 2
44	.014 296 7	37 5	.016 668 6	41 7	.019 304 2	46 2	.022 221 3	51 1	.025 438 6	56 2
45	.014 334 2	37 5	.016 710 3	41 8	.019 350 4	46 4	.022 272 4	51 2	.025 494 8	56 4
46	.014 371 7	37 6	.016 752 1	41 8	.019 396 8	46 4	.022 323 6	51 3	.025 551 2	56 4
47	.014 409 3	37 7	.016 793 9	42 0	.019 443 2	46 5	.022 374 9	51 3	.025 607 6	56 6
48	.014 447 0	37 7	.016 835 9	42 0	.019 489 7	46 6	.022 426 2	51 5	.025 664 2	56 6
49	.014 484 7	37 8	.016 877 9	42 1	.019 536 3	46 6	.022 477 7	51 5	.025 720 8	56 7
50	.014 522 5	37 9	.016 920 0	42 1	.019 582 9	46 7	.022 529 2	51 6	.025 777 5	56 8
51	.014 560 4	37 9	.016 962 1	42 3	.019 629 6	46 9	.022 580 8	51 7	.025 834 3	56 9
52	.014 598 3	38 0	.017 004 4	42 3	.019 676 5	46 8	.022 632 5	51 8	.025 891 2	57 0
53	.014 636 3	38 1	.017 046 7	42 4	.019 723 3	47 0	.022 684 3	51 8	.025 948 2	57 1
54	.014 674 4	38 2	.017 089 1	42 4	.019 770 3	47 1	.022 736 1	52 0	.026 005 3	57 2
55	.014 712 6	38 2	.017 131 5	42 5	.019 817 4	47 1	.022 788 1	52 0	.026 062 5	57 2
56	.014 750 8	38 3	.017 174 0	42 6	.019 864 5	47 2	.022 840 1	52 1	.026 119 7	57 4
57	.014 789 1	38 4	.017 216 6	42 7	.019 911 7	47 3	.022 892 2	52 2	.026 177 1	57 4
58	.014 827 5	38 4	.017 259 3	42 8	.019 959 0	47 3	.022 944 4	52 3	.026 234 5	57 5
59	.014 865 9	38 5	.017 302 1	42 8	.020 006 3	47 5	.022 996 7	52 4	.026 292 0	57 7
60	.014 904 4		.017 344 9		.020 053 8		.022 049 1		.026 349 7	

Involute function ③

α′ \ α°	24 inv α	24 Differ	25 inv α	25 Differ	26 inv α	26 Differ	27 inv α	27 Differ	28 inv α	28 Differ
0	.026 349 7	57 7	.029 975 3	63 3	.033 947 0	69 2	.038 286 6	75 5	.043 017 2	82 3
1	.026 407 4	57 8	.030 038 6	63 4	.034 016 2	69 4	.038 362 1	75 7	.043 099 5	82 4
2	.026 465 2	57 9	.030 102 0	63 5	.034 085 6	69 4	.038 437 8	75 8	.043 181 9	82 6
3	.026 523 1	57 9	.030 165 5	63 6	.034 155 0	69 6	.038 513 6	75 9	.043 264 5	82 6
4	.026 581 0	58 1	.030 229 1	63 7	.034 224 6	69 6	.038 589 5	76 0	.043 347 1	82 8
5	.026 639 1	58 2	.030 292 8	63 8	.034 294 2	69 8	.038 665 5	76 1	.043 429 9	82 9
6	.026 697 3	58 2	.030 356 6	63 9	.034 364 0	69 9	.038 741 6	76 3	.043 512 8	82 9
7	.026 755 5	58 4	.030 420 5	63 9	.034 433 9	69 9	.038 817 9	76 3	.043 595 7	83 2
8	.026 813 9	58 4	.030 484 4	64 1	.034 503 8	70 1	.038 894 2	76 4	.043 678 9	83 2
9	.026 872 3	58 5	.030 548 5	64 2	.034 573 9	70 2	.038 970 6	76 6	.043 762 1	83 3
10	.026 930 8	58 6	.030 612 7	64 2	.034 644 1	70 3	.039 047 2	76 7	.043 845 4	83 5
11	.026 989 4	58 7	.030 676 9	64 4	.034 714 4	70 3	.039 123 9	76 7	.043 928 9	83 5
12	.027 048 1	58 8	.030 741 3	64 5	.034 784 7	70 5	.039 200 6	76 9	.044 012 4	83 7
13	.027 106 9	58 9	.030 805 8	64 5	.034 855 2	70 6	.039 277 5	77 0	.044 096 1	83 8
14	.027 165 8	59 0	.030 870 3	64 7	.035 925 8	70 7	.039 354 5	77 1	.044 179 9	84 0
15	.027 224 8	59 1	.030 935 0	64 7	.035 996 5	70 8	.039 431 6	77 2	.044 263 9	84 0
16	.027 283 9	59 1	.030 999 7	64 9	.035 067 3	70 9	.039 508 8	77 4	.044 347 9	84 2
17	.027 343 0	59 3	.031 064 6	64 9	.035 138 2	71 0	.039 586 2	77 4	.044 432 1	84 2
18	.027 402 3	59 4	.031 129 5	65 1	.035 209 2	71 1	.039 663 6	77 5	.044 516 3	84 4
19	.027 461 7	59 4	.031 194 6	65 1	.035 280 3	71 2	.039 741 1	77 7	.044 600 7	84 6
20	.027 521 1	59 5	.031 259 7	65 3	.035 351 5	71 3	.039 818 8	77 8	.044 685 3	84 6
21	.027 580 6	59 7	.031 325 0	65 3	.035 422 8	71 4	.039 896 6	77 9	.044 769 9	84 7
22	.027 640 3	59 7	.031 390 3	65 4	.035 494 2	71 6	.039 974 5	77 9	.044 854 6	84 9
23	.027 700 0	59 8	.031 455 7	65 6	.035 565 8	71 6	.040 052 4	78 2	.044 939 5	85 0
24	.027 759 8	59 9	.031 521 3	65 6	.035 637 4	71 7	.040 130 6	78 2	.045 024 5	85 1
25	.027 819 7	60 0	.031 586 9	65 8	.035 709 1	71 9	.040 208 8	78 3	.045 109 6	85 2
26	.027 879 7	60 1	.031 652 7	65 8	.035 781 0	71 9	.040 287 1	78 4	.045 194 8	85 3
27	.027 939 8	60 1	.031 718 5	65 9	.035 852 9	72 0	.040 365 5	78 6	.045 280 1	85 5
28	.027 999 9	60 3	.031 784 4	66 0	.035 924 9	72 2	.040 444 1	78 6	.045 365 6	85 6
29	.028 060 2	60 4	.031 850 4	66 2	.035 997 1	72 3	.040 522 7	78 8	.045 451 2	85 7
30	.028 120 6	60 4	.031 916 6	66 2	.036 069 4	72 3	.040 601 5	78 9	.045 536 9	85 8
31	.028 181 0	60 6	.031 982 8	66 3	.036 141 7	72 5	.040 680 4	79 0	.045 622 7	85 9
32	.028 241 6	60 6	.032 049 1	66 5	.036 214 2	72 6	.040 759 4	79 1	.045 708 6	86 1
33	.028 302 2	60 8	.032 115 6	66 5	.036 286 8	72 6	.040 838 5	79 2	.045 794 7	86 1
34	.028 363 0	60 9	.032 182 1	66 6	.036 359 4	72 8	.040 917 7	79 3	.045 880 8	86 3
35	.028 423 8	61 9	.032 248 7	66 7	.036 432 2	72 9	.040 997 0	79 5	.045 967 1	86 4
36	.028 484 7	61 1	.032 315 4	66 9	.036 505 1	73 0	.041 076 5	79 6	.046 053 5	86 6
37	.028 545 8	61 1	.032 382 3	66 9	.036 578 1	73 1	.041 156 1	79 6	.046 140 1	86 6
38	.028 606 9	61 2	.032 449 2	67 0	.036 651 2	73 2	.041 235 7	79 8	.046 226 7	86 8
39	.028 668 1	61 3	.032 516 2	67 1	.036 724 4	73 3	.041 315 5	79 9	.046 313 5	86 9
40	.028 729 4	61 4	.032 583 3	67 3	.036 797 7	73 5	.041 395 4	80 0	.046 400 4	87 0
41	.028 790 8	61 5	.032 650 6	67 3	.036 871 2	73 5	.041 475 4	80 1	.046 487 4	87 1
42	.028 852 3	61 6	.032 717 9	67 4	.036 944 7	73 6	.041 555 5	80 3	.046 574 5	87 3
43	.028 913 9	61 6	.032 785 3	67 5	.037 018 3	73 8	.041 635 8	80 3	.046 661 8	87 3
44	.028 975 5	61 8	.032 852 8	67 7	.037 092 1	73 8	.041 716 1	80 5	.046 749 1	87 5
45	.029 037 3	61 9	.032 920 5	67 7	.037 165 9	74 0	.041 796 6	80 6	.046 836 6	87 6
46	.029 099 2	62 0	.032 988 2	67 8	.037 239 9	74 0	.041 877 2	80 7	.046 924 2	87 8
47	.029 161 2	62 0	.033 056 0	67 9	.037 313 9	74 2	.041 957 9	80 8	.047 012 0	87 8
48	.029 223 2	62 2	.033 123 9	68 1	.037 388 1	74 3	.042 038 7	80 9	.047 099 8	88 0
49	.029 285 4	62 2	.033 192 0	68 1	.037 462 4	74 4	.042 119 6	81 0	.047 187 8	88 1
50	.029 347 6	62 4	.033 260 1	68 2	.037 536 8	74 5	.042 200 6	81 2	.047 275 9	88 2
51	.029 410 0	62 4	.033 328 3	68 4	.037 611 3	74 6	.042 281 8	81 2	.047 364 1	88 4
52	.029 472 4	62 5	.033 396 7	68 4	.037 685 9	74 7	.042 363 0	81 4	.047 452 5	88 4
53	.029 534 9	62 7	.033 465 1	68 5	.037 760 6	74 8	.042 444 4	81 5	.047 540 9	88 6
54	.029 597 6	62 7	.033 533 6	68 7	.037 835 4	74 9	.042 525 9	81 6	.047 629 5	88 7
55	.029 660 3	62 8	.033 602 3	68 7	.037 910 3	75 0	.042 607 5	81 7	.047 718 2	88 8
56	.029 723 1	62 9	.033 671 0	68 8	.037 985 3	75 2	.042 689 2	81 8	.047 807 0	89 0
57	.029 786 0	63 0	.033 739 8	69 0	.038 060 5	75 2	.042 771 0	82 0	.047 896 0	89 1
58	.029 849 0	63 1	.033 808 8	69 0	.038 135 7	75 4	.042 853 0	82 1	.047 985 1	89 2
59	.029 912 1	63 2	.033 877 8	69 2	.038 211 1	75 5	.042 935 1	82 1	.048 074 3	89 3
60	.029 975 3		.033 947 0		.038 286 6		.043 017 2		.048 163 6	

Involute function ④

α' \ $\alpha°$	29		30		31		32		33	
	inv α	Differ	inv α	Differ	inv α	Differ	inv α	Differ	inv α	Differ
0	.048 163 6	89 4	.053 751 5	97 0	.059 808 6	105 0	.066 364 0	113 6	.073 448 9	122 8
1	.048 253 0	89 6	.053 848 5	97 2	.059 913 6	105 3	.066 477 6	113 9	.073 571 7	122 9
2	.048 342 6	89 7	.053 945 7	97 3	.060 018 9	105 3	.066 591 5	113 9	.073 694 6	123 1
3	.048 432 3	89 8	.054 043 0	97 4	.060 124 2	105 5	.066 705 4	114 1	.073 817 7	123 2
4	.048 522 1	89 9	.054 140 4	97 5	.060 229 7	105 7	.066 819 5	114 2	.073 940 9	123 4
5	.048 612 0	90 0	.054 237 9	97 7	.060 335 4	105 8	.066 933 7	114 4	.074 064 3	123 5
6	.048 702 0	90 2	.054 335 6	97 8	.060 441 2	105 9	.067 048 1	114 6	.074 187 8	123 7
7	.048 792 2	90 3	.054 433 4	98 0	.060 547 1	106 1	.067 162 7	114 7	.074 311 5	123 9
8	.048 882 5	90 5	.054 531 4	98 1	.060 653 2	106 2	.067 277 4	114 8	.074 435 4	124 0
9	.048 973 0	90 5	.054 629 5	98 2	.060 759 4	106 3	.067 392 2	115 0	.074 559 4	124 1
10	.049 063 5	90 7	.054 727 7	98 3	.060 865 7	106 5	.067 507 2	115 1	.074 683 5	124 4
11	.049 154 2	90 8	.054 826 0	98 5	.060 972 2	106 6	.067 622 3	115 3	.074 807 9	124 5
12	.049 245 0	90 9	.054 924 5	98 6	.061 078 8	106 8	.067 737 6	115 4	.074 932 4	124 6
13	.049 335 9	91 0	.055 023 1	98 7	.061 185 6	106 9	.067 853 0	115 6	.075 057 0	124 8
14	.049 426 9	91 2	.055 121 8	98 9	.061 292 5	107 0	.067 968 6	115 7	.075 181 8	125 0
15	.049 518 1	91 3	.055 220 7	99 0	.061 399 5	107 2	.068 084 3	115 9	.075 306 8	125 1
16	.049 609 4	91 4	.055 319 7	99 1	.061 506 7	107 3	.068 200 2	116 0	.075 431 9	125 2
17	.049 700 8	91 6	.055 418 8	99 3	.061 614 0	107 5	.068 316 2	116 2	.075 557 1	125 5
18	.049 792 4	91 6	.055 518 1	99 4	.061 721 5	107 6	.068 432 4	116 3	.075 682 6	125 6
19	.049 884 0	91 8	.055 617 5	99 5	.061 829 1	107 7	.068 548 7	116 5	.075 808 2	125 7
20	.049 975 8	91 9	.055 717 0	99 6	.061 936 8	107 9	.068 665 2	116 6	.075 933 9	125 9
21	.050 067 7	92 1	.055 816 6	99 8	.062 044 7	108 0	.068 781 8	116 8	.076 059 8	126 1
22	.050 159 8	92 1	.055 916 4	100 0	.062 152 7	108 2	.068 898 6	116 9	.076 185 9	126 2
23	.050 251 9	92 3	.056 016 4	100 0	.062 260 9	108 3	.069 015 5	117 1	.076 312 1	126 4
24	.050 344 2	92 5	.056 116 4	100 2	.062 369 2	108 5	.069 132 6	117 3	.076 438 5	126 6
25	.050 436 7	92 5	.056 216 6	100 3	.062 477 7	108 6	.069 249 9	117 3	.076 565 1	126 7
26	.050 529 2	92 7	.056 316 9	100 5	.062 586 3	108 7	.069 367 2	117 6	.076 691 8	126 9
27	.050 621 9	92 8	.056 417 4	100 6	.062 695 0	108 9	.069 484 8	117 6	.076 818 7	127 0
28	.050 714 7	92 9	.056 518 0	100 7	.062 803 9	109 0	.069 602 4	117 9	.076 945 7	127 2
29	.050 807 6	93 0	.056 618 7	100 9	.062 912 9	109 2	.069 720 3	118 0	.077 072 9	127 4
30	.050 900 6	93 2	.056 719 6	101 0	.063 022 1	109 3	.069 838 3	118 1	.077 200 3	127 5
31	.050 993 8	93 3	.056 820 6	101 1	.063 131 4	109 4	.069 956 4	118 3	.077 327 8	127 7
32	.051 087 1	93 5	.056 921 7	101 3	.063 240 8	109 6	.070 074 7	118 4	.077 455 5	127 8
33	.051 180 6	93 5	.057 023 0	101 4	.063 350 4	109 8	.070 193 1	118 6	.077 583 3	128 0
34	.051 274 1	93 7	.057 124 4	101 5	.063 460 2	109 8	.070 311 7	118 7	.077 711 3	128 2
35	.051 367 8	93 8	.057 225 9	101 7	.063 570 0	110 1	.070 430 4	118 9	.077 839 5	128 3
36	.051 461 6	93 9	.057 327 6	101 8	.063 680 1	110 1	.070 549 3	119 1	.077 967 8	128 5
37	.051 555 5	94 1	.057 429 4	101 9	.063 790 2	110 3	.070 668 4	119 2	.078 096 3	128 6
38	.051 649 6	94 2	.057 531 3	102 1	.063 900 5	110 5	.070 787 6	119 3	.078 224 9	128 8
39	.051 743 8	94 3	.057 633 4	102 2	.064 011 0	110 6	.070 906 9	119 6	.078 353 7	129 0
40	.051 838 1	94 5	.057 735 6	102 4	.064 121 6	110 7	.071 026 5	119 6	.078 482 7	129 1
41	.051 932 6	94 5	.057 838 0	102 5	.064 232 3	110 9	.071 146 1	119 8	.078 611 8	129 3
42	.052 027 1	94 7	.057 940 5	102 6	.064 343 2	111 0	.071 265 9	120 0	.078 741 1	129 5
43	.052 121 8	94 9	.058 043 1	102 7	.064 454 2	111 2	.071 385 9	120 1	.078 870 6	129 6
44	.052 216 7	94 9	.058 145 8	102 9	.064 565 4	111 3	.071 506 0	120 3	.079 000 2	129 8
45	.052 311 6	95 1	.058 248 7	103 1	.064 676 7	111 5	.071 626 3	120 4	.079 130 0	130 0
46	.052 406 7	95 2	.058 351 8	103 1	.064 788 2	111 6	.071 746 7	120 6	.079 260 0	130 1
47	.052 501 9	95 4	.058 454 9	103 3	.064 899 8	111 8	.071 867 3	120 7	.079 390 1	130 3
48	.052 597 3	95 5	.058 558 2	103 5	.065 011 6	111 9	.071 988 0	120 9	.079 520 4	130 4
49	.052 692 8	95 6	.058 661 7	103 5	.065 123 5	112 0	.072 108 9	121 1	.079 650 8	130 6
50	.052 788 4	95 7	.058 765 2	103 8	.065 235 5	112 2	.072 230 0	121 2	.079 781 4	130 8
51	.052 884 1	95 9	.058 869 0	103 8	.065 347 7	112 3	.072 351 2	121 3	.079 912 2	130 9
52	.052 980 0	95 9	.058 972 8	104 0	.065 460 0	112 5	.072 472 5	121 5	.080 043 1	131 1
53	.053 075 9	96 2	.059 076 8	104 1	.065 572 5	112 6	.072 594 0	121 7	.080 174 2	131 3
54	.053 172 1	96 2	.059 180 9	104 3	.065 685 1	112 8	.072 715 7	121 8	.080 305 5	131 4
55	.053 268 3	96 4	.059 285 2	104 4	.065 797 9	112 9	.072 837 5	122 0	.080 436 9	131 6
56	.053 364 7	96 5	.059 389 6	104 5	.065 910 8	113 1	.072 959 5	122 1	.080 568 5	131 8
57	.053 461 2	96 6	.059 494 1	104 7	.066 023 9	113 2	.073 081 6	122 3	.080 700 3	131 9
58	.053 557 8	96 8	.059 598 8	104 8	.066 137 1	113 4	.073 203 9	122 4	.080 832 2	132 1
59	.053 654 6	96 9	.059 703 6	105 0	.066 250 5	113 5	.073 326 3	122 6	.080 964 3	132 3
60	.053 751 5		.059 808 6		.066 364 0		.073 448 9		.081 096 6	

Metric coarse and Fine screw threads

Extracted from JIS B0205, 0207

Nominal threads	Pitch P			
	Coarse screw	Fine screw		
M1	0.25	0.2		
M1.1	0.25	0.2		
M1.2	0.25	0.2		
M1.4	0.3	0.2		
M1.6	0.35	0.2		
M1.8	0.35	0.2		
M2	0.4	0.25		
M2.2	0.45	0.25		
M2.5	0.45	0.35		
M3	0.5	0.35		
M3.5	0.6	0.35		
M4	0.7	0.5		
M4.5	0.75	0.5		
M5	0.8	0.5		
M6	1	0.75		
M8	1.25	0.75	1	
M10	1.5	0.75	1	1.25
M12	1.75	1	1.25	1.5
M14	2	1	1.25	1.5
M16	2	1	1.5	1.5
M18	2.5	1	1.5	2
M20	2.5	1	1.5	2
M22	2.5	1	1.5	2
M24	3	1	1.5	2
M27	3	1	1.5	2
M30	3.5	1	1.5	2

Following details are for reference only and not part of JIS standard.

Fig. A

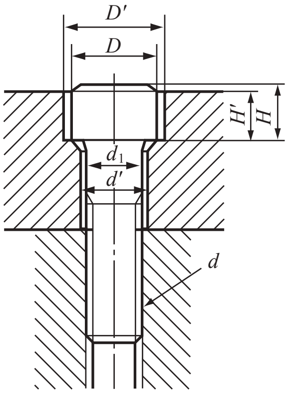
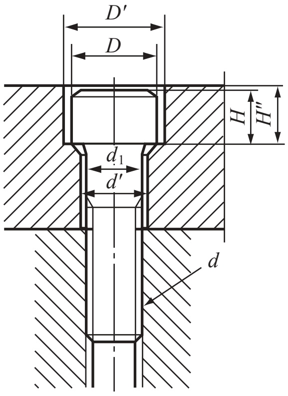


Fig. B



Spot facing and Thread hole for Hexagon socket head cap screws

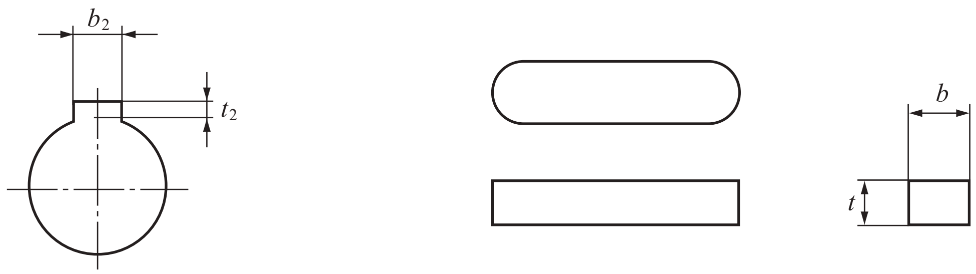
Unit: mm

Nominal thread (d)	M3	M4	M5	M6	M8	M10	M12	M14	M16	M18	M20	M22	M24	M27	M30	M33	M36	M39	M42	M45	M48	M52
d_1	3	4	5	6	8	10	12	14	16	18	20	22	24	27	30	33	36	39	42	45	48	52
d'	3.4	4.5	5.5	6.6	9	11	14	16	18	20	22	24	26	30	33	36	39	42	45	48	52	56
D	5.5	7	8.5	10	13	16	18	21	24	27	30	33	36	40	45	50	54	58	63	68	72	78
D'	6.5	8	9.5	11	14	17.5	20	23	26	29	32	35	39	43	48	54	58	62	67	72	76	82
H	3	4	5	6	8	10	12	14	16	18	20	22	24	27	30	33	36	39	42	45	48	52
H'	2.7	3.6	4.6	5.5	7.4	9.2	11	12.8	14.5	16.5	18.5	20.5	22.5	25	28	31	34	37	39	42	45	49
H''	3.3	4.4	5.4	6.5	8.6	10.8	13	15.2	17.5	19.5	21.5	23.5	25.5	29	32	35	38	41	44	47	50	54

Remark: Thread holes (d’) provide Class 2 from JIS B 1001 (Thread holes and Spot facing holes)

Parallel key and Key Way

Dimensions and tolerances for KG-gear with Key way are equivalent to JIS B1301.



Tolerances for Key

$b \times t$	3 × 3	4 × 4	5 × 5	6 × 6	8 × 7	10 × 8	12 × 8	14 × 9
b Tolerance (h)	h9	h9	h9	h9	h9	h9	h9	h9
t Tolerance (h)	h9	h9	h9	h9	h11	h11	h11	h11

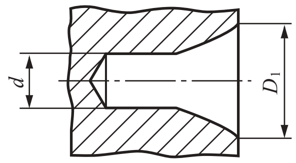
Key way for KG-STOCK GEARS

Unit : mm

Dimensions	Bore dimensions	Key way $b_2 \times t_2$	Width		Depth	
			b_2	Tolerance Js 9	t_2	Tolerance
ϕ 8 ～ ϕ10	ϕ 8	3 × 1.4	3	± 0.0125	1.4	$^{+0.1}_{0}$
	ϕ10					
ϕ10 ～ ϕ12	ϕ12	4 × 1.8	4	± 0.015	1.8	
ϕ12 ～ ϕ17	ϕ14	5 × 2.3	5		2.3	
	ϕ15					
	ϕ16					
ϕ17 ～ ϕ22	ϕ18	6 × 2.8	6		2.8	
	ϕ20					
	ϕ22					
ϕ22 ～ ϕ30	ϕ25	8 × 3.3	8	± 0.018	3.3	$^{+0.2}_{0}$
	ϕ28					
	ϕ30					
ϕ30 ～ ϕ38	ϕ32	10 × 3.3	10		3.3	
	ϕ35					
ϕ38 ～ ϕ44	ϕ40	12 × 3.3	12	± 0.0215	3.3	
ϕ44 ～ ϕ50	ϕ45	14 × 3.8	14		3.8	
	ϕ50					

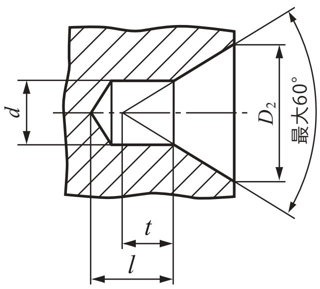
Centre bore JIS B1011

Type R



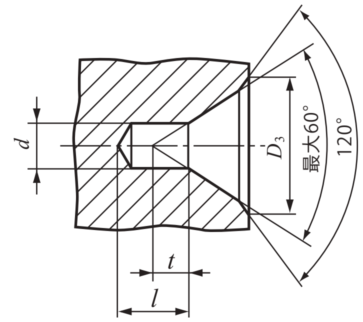
Form with circular arc
(Drilling centre bore from JIS B4304)

Type A



Form without chamfering
(Drilling centre bore from JIS B4304)

Type B



Form with chamfering
(Drilling centre bore from JIS B4304)

Note∗ : Length ‘l’ is based on centre drill but length must be longer than dimension ‘t’.

Centre bore (recommended)

Unit : mm

Nominal d	Type				
	Type R JIS B4304	Type A JIS B4304		Type B JIS B4304	
	D_1 Nominal	D_2 Nominal	t Reference	D_3 Nominal	t Reference
(0.5)		1.06	0.5		
(0.63)		1.32	0.6		
(0.8)		1.70	0.7		
1.0	2.12	2.12	0.9	3.15	0.9
(1.25)	2.65	2.65	1.1	4	1.1
1.6	3.35	3.35	1.4	5	1.4
2.0	4.25	4.25	1.8	6.3	1.8
2.5	5.3	5.30	2.2	8	2.2
3.15	6.7	6.70	2.8	10	2.8
4.0	8.5	8.50	3.5	12.5	3.5
(5.0)	10.6	10.60	4.4	16	4.4
6.3	13.2	13.20	5.5	18	5.5
(8.0)	17.0	17.00	7.0	22.4	7.0
10.0	21.2	21.20	8.7	28	8.7

Using figures in bracket ( ) is not advisable.