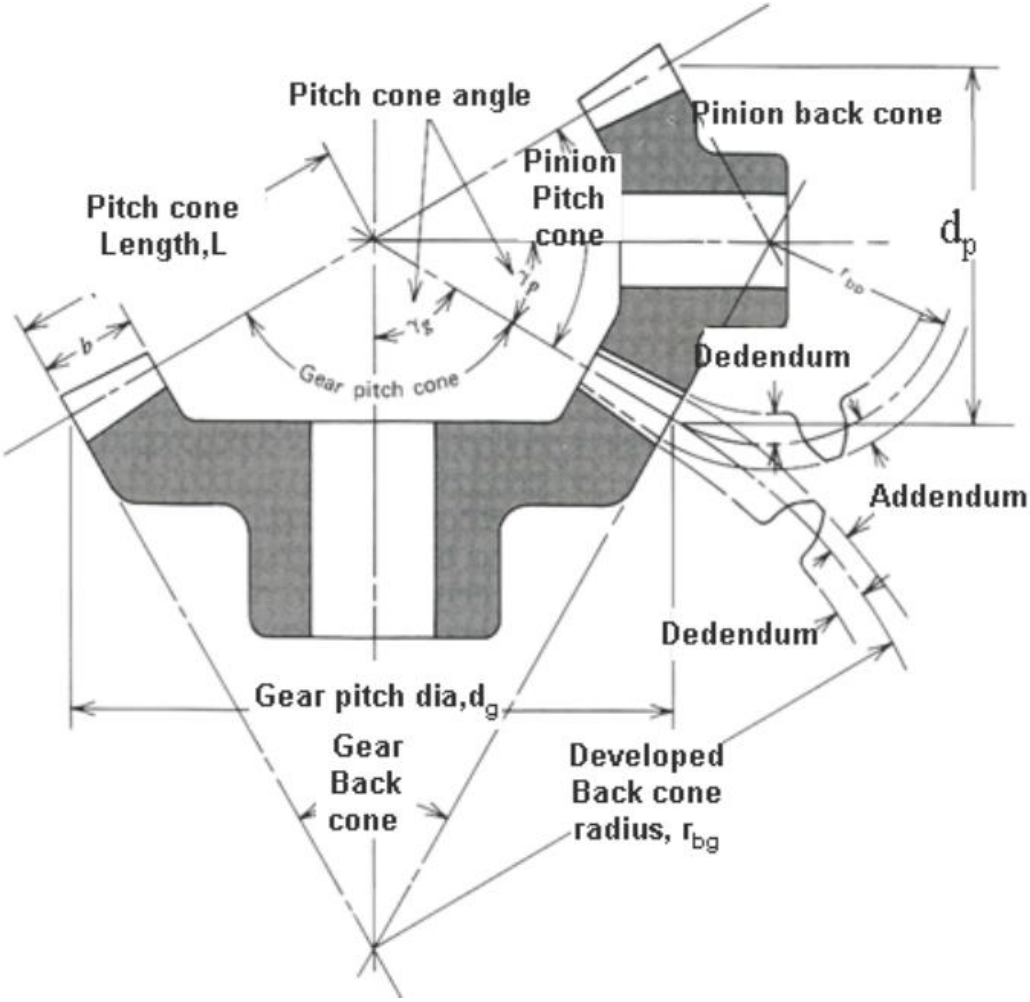
Bevel Gear

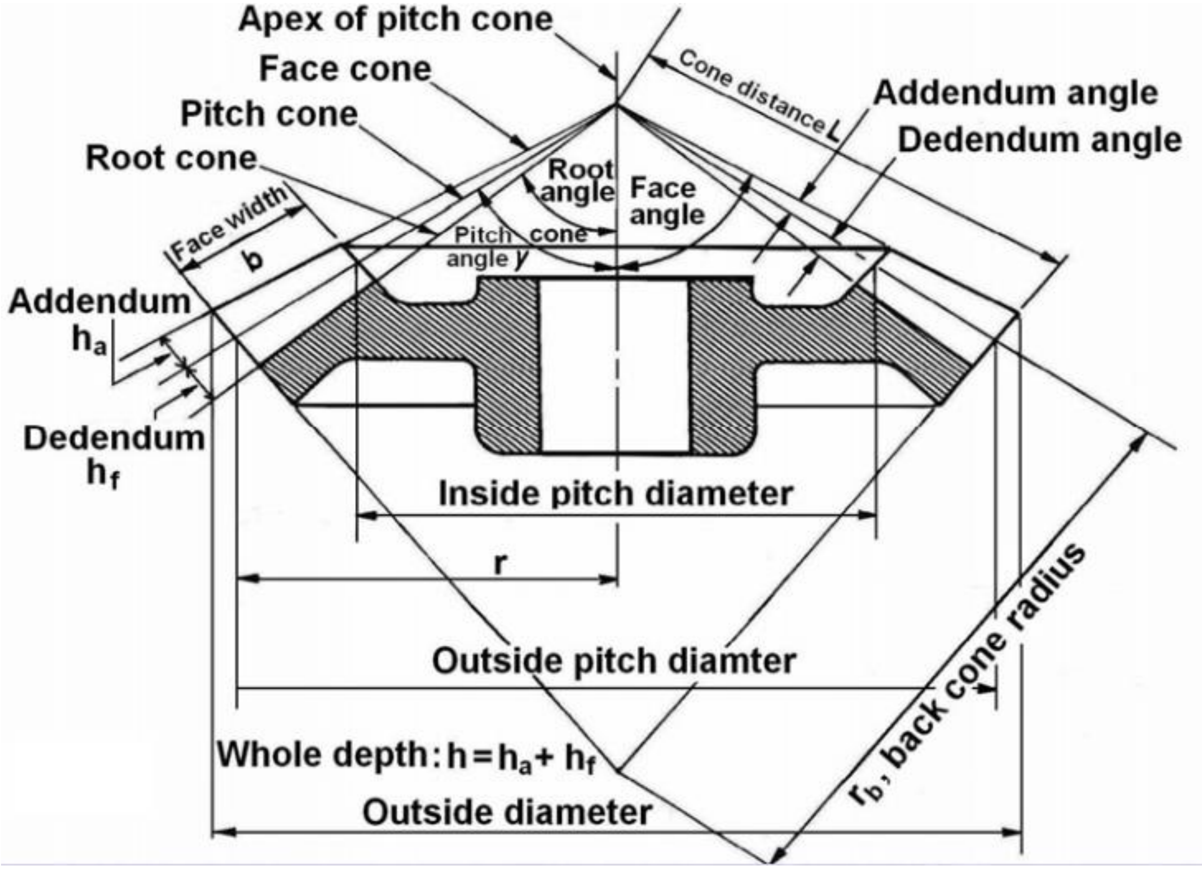
Bevel gears transmit power between two intersecting shafts at any angle or between non-intersecting shafts. They are classified as straight and spiral tooth bevel and hypoid gears.

Geometry And Terminology



When intersecting shafts are connected by gears, the pitch cones (analogous to the pitch cylinders of spur and helical gears) are tangent along an element, with their apexes at the intersection of the shafts as in Fig.13.2 where two bevel gears are in mesh. The size and shape of the teeth are defined at the large end, where they intersect the back cones. Pitch cone and back cone elements are perpendicular to each other. The

tooth profiles resemble those of spur gears having pitch radii equal to the developed back cone radii rbg and rbp and are shown in Fig. which explains the nomenclatures of a bevel gear.



Z= = p cosγ (13.1) (13.2) b2 2 Z = v2 2πr Z = p cosγ where Zv is called the virtual number of teeth, p is the circular pitch of both the imaginary spur gears and the bevel gears. Z1 and Z2 are the number of teeth on the pinion and gear, γ1 and γ2 are the pitch cone angles of pinion and gears. It is a practice to characterize the size and shape of bevel gear teeth as those of an imaginary spur gear appearing on the developed back cone corresponding to Tredgold's approximation.

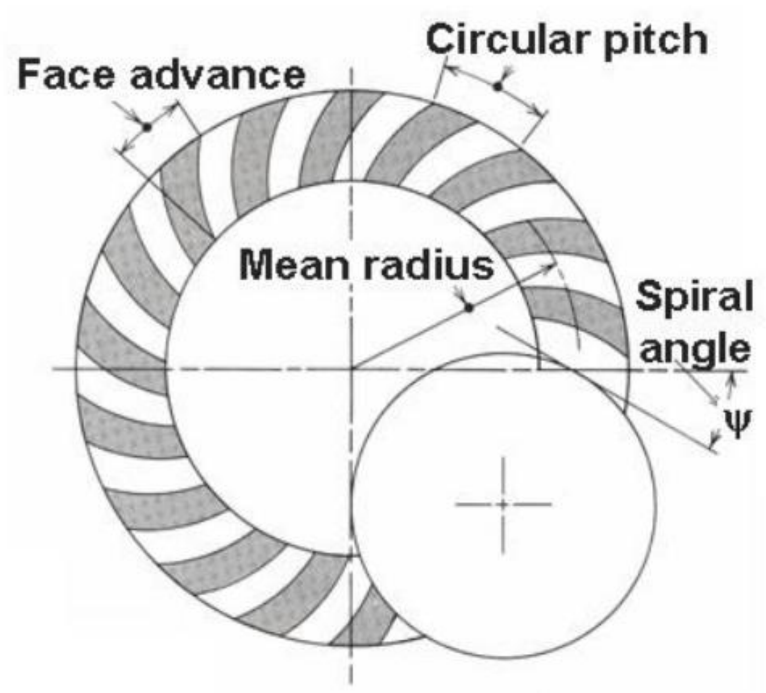
a) Bevel gear teeth are inherently non - interchangeable.

b) The working depth of the teeth is usually 2m, the same as for standard spur and helical gears, but the bevel pinion is designed with the larger addendum (0.7 working depth).

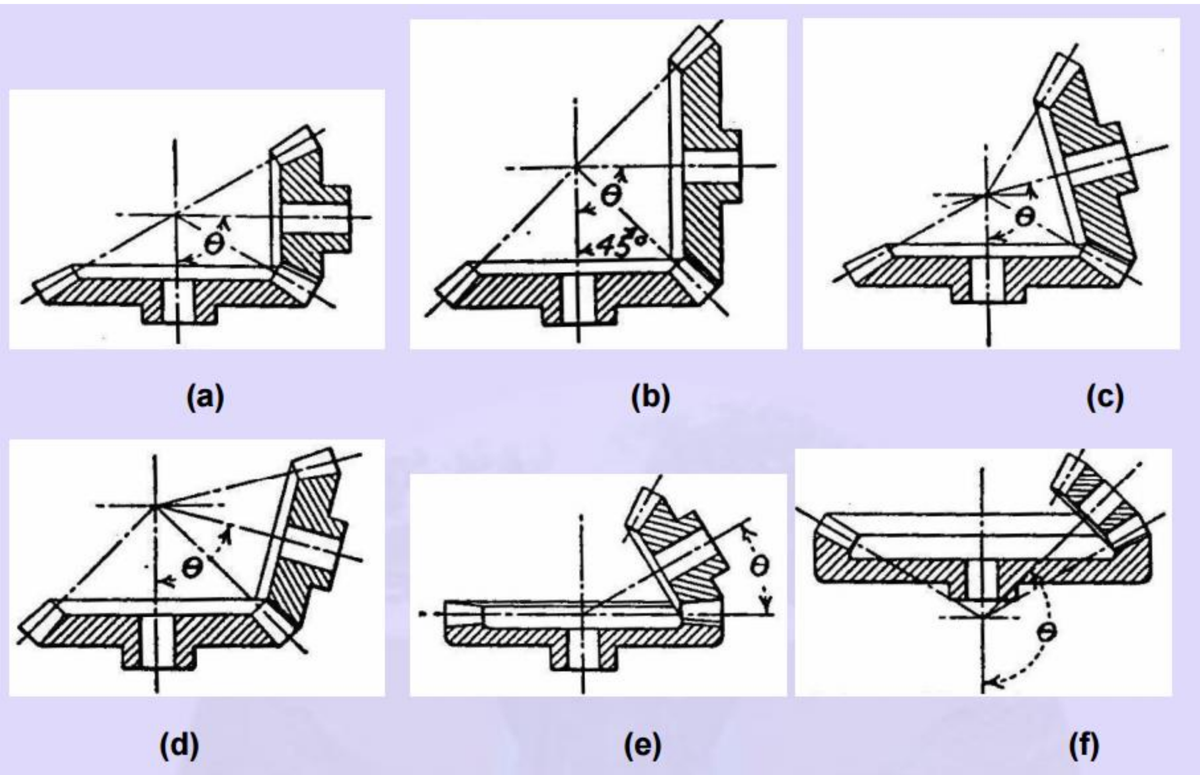
c) This avoids interference and results in stronger pinion teeth. It also increases the contact ratio.

d) The gear addendum varies from 1m for a gear ratio of 1, to 0.54 m for ratios of 6.8 and greater.

The gear ratio can be determined from the number of teeth, the pitch diameters or the pitch cone angles as, (13.3) 11 22 2 1 22 11 ω nZd i = = = = = tanγ =cotγ ω nZd Accepted practice usually imposes two limits on the face width (13.4) Where L is the cone distance. Smaller of the two is chosen for design.



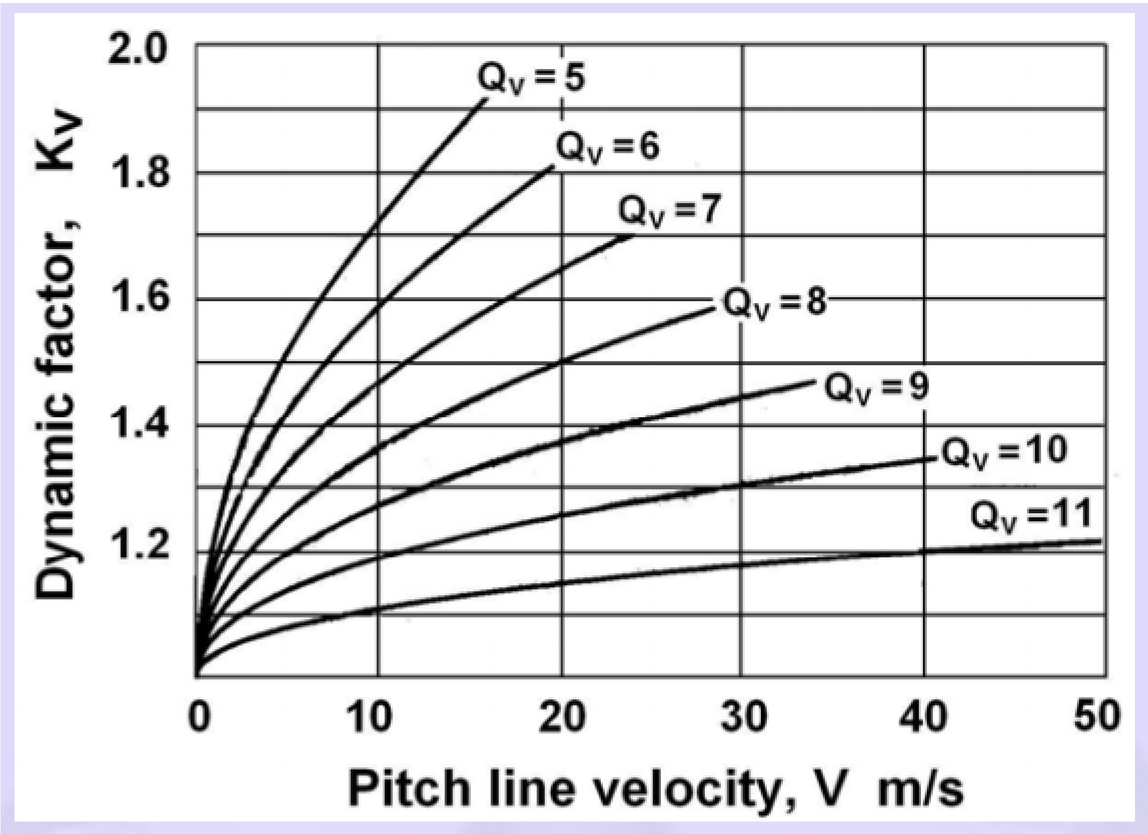
Different types of bevel gears



(a) Usual form, (b) Miter gears, (c), (d), (e) Crown gear, (f) Internal bevel gear

TOOTH BENDING STRESS

The equation for bevel gear bending stress is the same as for spur gears as shown below: t b vom F σ = K K K (13.13) bmJ Where, Ft =Tangential load in N m = module at the large end of the tooth in mm b = Face width in mm J = Geometry form factor based on virtual number of teeth from Fig. 13.12 and 13.13. Kv = Velocity factor, from Fig.13.14. Ko = Overload factor, Table 13.1. Km= Mounting factor, depending on whether gears are straddle mounted (between two bearings) or overhung (outboard of both bearings), and on the degree of mounting rigidity.



Dynamic load factor, Kv

Worm Wheels

Worm gears are used for transmitting power between two non-parallel, non-intersecting shafts. High gear ratios of 200:1 can be got.

Geometry

a. The geometry of a worm is similar to that of a power screw. Rotation of the worm simulates a linearly advancing involute rack, Fig.15.3 b. The geometry of a worm gear is similar to that of a helical gear, except that the teeth are curved to envelop the worm. c. Enveloping the gear gives a greater area of contact but requires extremely precise mounting. 1. As with a spur or helical gear, the pitch diameter of a worm gear is related to its circular pitch and number of teeth Z by the formula (15.1) 2 2 Z p d π = 2. When the angle is 90° between the nonintersecting shafts, the worm lead angle λ is equal to the gear helix angleψ. Angles λ and ψ have the same hand. 3. The pitch diameter of a worm is not a function of its number of threads, Z1. 4. This means that the velocity ratio of a worm gear set is determined by the ratio of gear teeth to worm threads; it is not equal to the ratio of gear and worm diameters. (15.2) 1 2 1 ω Z = ω Z 2 5. Worm gears usually have at least 24 teeth, and the number of gear teeth plus worm threads should be more than 40: Z1 + Z2 > 40 (15.3) 6. A worm of any pitch diameter can be made with any number of threads and any axial pitch. 7. For maximum power transmitting capacity, the pitch diameter of the worm should normally be related to the shaft center distance 8. Integral worms cut directly on the shaft can, of course, have a smaller diameter than that of shell worms, which are made separately. 9. Shell worms are bored to slip over the shaft and are driven by splines, key, or pin. 10. Strength considerations seldom permit a shell worm to have a pitch diameter less than d1 = 2.4p + 1.1 (15.5) 11. The face width of the gear should not exceed half the worm outside diameter. b ≤ 0.5 da1 (15.6) 12. Lead angle λ, Lead L, and worm pitch diameter d1 have the following relationship in connection with the screw threads. 1 L tan λ= (πd 15.7) 13. To avoid interference, pressure angles are commonly related to the worm lead angle

KINEMATICS

The relationship between worm tangential velocity, gear tangential velocity, and sliding velocity is, (15.17) 2 1 V = tanλ V

EFFICIENCY

Efficiency η is the ratio of work out to work in. For the usual case of the worm serving as input member, (15.18) The overall efficiency of a worm gear is a little lower because of friction losses in the bearings and shaft seals, and because of “churning” of the lubricating oil.

BENDING AND SURFACE FATIGUE STRENGTHS

Worm gear capacity is often limited not by fatigue strength but by cooling capacity. The total gear tooth load Fd is the product of nominal load Ft and factors accounting for impact from tooth inaccuracies and deflections, misalignment, etc.). Fd must be less than the strength the bending fatigue and surface fatigue strengths Fb and Fw The total tooth load is called the dynamic load Fd, the bending fatigue limiting load is called strength capacity Fb, and the surface fatigue limiting load is called the wear capacity Fw. For satisfactory performance, Fb ≥ Fd and Fw

≥ Fd (15.22) The “dynamic load” is estimated by multiplying the nominal value of gear tangential force by velocity factor “Kv”

F =F K =F 6.1 ⎛ ⎞⎟ ⎝ ⎠⎟ Adapting the Lewis equation to the gear teeth, we have (15 F =[ ] bpy = [ ] bmY b σ σ b b .24) Where, [σb] is the permissible bending stress in bending fatigue, in MPa

THERMAL CAPACITY

The continuous rated capacity of a worm gear set is often limited by the ability of the housing to dissipate friction heat without developing excessive gear and lubricant temperatures. Normally, oil temperature must not exceed about 200ºF (93o C) for satisfactory operation. The fundamental relationship between temperature rise and rate of heat dissipation used for journal bearings does hold good for worm gearbox. (15.26) H = C A T -T H 0a ( ) Where H – Time rate of heat dissipation (Nm/sec) CH – Heat transfer coefficient (Nm/sec/m2 /ºC) A – Housing external surface area (m2) To – Oil temperature (º C) Ta – Ambiant air temperature (º C)