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# HOW TO DESIGN SPUR GEAR EFFECTS OF VARIOUS PARAMETERS ON STRENGTH OF SPUR GEAR

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***Abstract: Gears are the most common means of transmitting power in modern mechanical engineering world. They vary from tiny size used in watches to large gears used in marine speed reducers, bridge lifting mechanisms. The gears are of many types namely spur gear, helical gears, worm gears etc. The design of any gear drive is extremely involved in view of many complex requirements to be met. Until about 1932, gear design was done by the help of Lewis equation based on static strength of tooth bending. Then after 1932, Buckingham proposed a new method of gear design based on dynamic load, endurance limit of of material and wear load. The equation is known as Buckingham equation. This research paper gives simplified format of designing spur gear pair with the help of design data book after research of all related referances.***
***Index Terms: Gear design, spur gear, lewis equation, buckingham equation, design data book.***

## I. INTRODUCTION

Gears are defined as toothed wheels or multilobed cams. Which transmit the power and motion from one shaft to another by means of successive engagement of teeth. Spur gear uses no intermediate link or connector and transmits the motion by direct contact. The two bodies have either a rolling or a sliding motion along the tangent at the point of contact. No motion is possible along the common normal as that will either break the contact or one body will tend to penetrate in to the other. Thus, the load application is gradual which results in low impact stresses and reduction in noise. Therefore, the spur gears are used in transmitting power with very less friction losses.

## II. SPUR GEAR

In spur gears the teeth are straight and parallel to the shaft axis. Spur gears are most common type and cost effective gears. Designed to transmit power and motion between parallel shafts .Plastic, brass, steel, & aluminum is generally used for manufacturing of spur gear. They are usually slow speed gears but if noise is not a serious problem then spur gears can be used at almost any speed which can be handled by other types of gears. The most usual arrangement of spur gear is an external gear and pinion combination, but if center distance is to be reduced, then internal gears and external pinion combination is used.

SPUR GEAR TERMINOLOGY

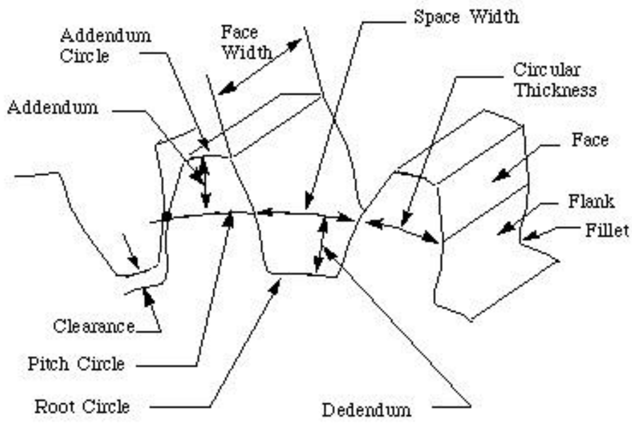


Fig: Spur Gear Terminology

**Speed ratio:** It is the ratio of pinion speed to gear speed. It can also be defined as the ratio of the number of teeth on the gear to the number of teeth on the pinion.

**Module:** It is the ratio of the pitch circle diameter in millimeters to the number of teeth.

**Pitch Circle Diameter:** It is the diameter of an imaginary circle which by pure rolling action would transmit the same motion as actual gear.

**Face Width:** It is the length of gear tooth measured along a line parallel to the gear axis.

**Addendum**: It is the radial distance between the top land of the teeth and the pitch circle.

**Dedendum:** It is the radial distance between the bottom land of the teeth and the pitch circle.

*NOTATION USED*
$\sigma_b$=Permissible bending stress
BHN=Brinell Hardness Number
b=face width
C=deformation factor
C.D.=centre distance
d = pitch circle diameter
$F_b$=beam strength
$F_d$=dynamic load
$F_{eff}$=effective load
$F_t$= theoretical tangential force
$F_{w=}$wear strength
$F_{tmax}$= maximum tangential force
i = Gear Ratio

  

K=load stress factor
$K_a$=service factor
$K_m$=load distribution factor
$K_v$= velocity factor
M =module
N= rotary Speed in RPM
P=power
Q=ratio factor
V=pitch line velocity
Y=lewis form factor
Z=number of teeth

Note: Suffix 'p' and 'g' indicates corresponding values for pinion and gear respectively.

## III. DESIGN PROCEDURE OF SPUR GEAR

I. Selection Of Spur Gear Material
Calculate gear ratio 'i'
From this value of gear ratio select material from design data book.
Desirable properties of gear material are;

1. High endurance strength in bending.
2. High surface endurance strength;
3. Low coefficient of friction
4. Low and consistent thermal distortion.

II. Check for the weaker component in bending
Calculate values of;

1. $б_{bp} * Y_p$
2. $б_{bg} * Y_g$

If $б_{bp} * Y_p > б_{bg} * Y_g$, then gear is weaker than pinion in bending.otherwise vice versa.

For values of Y i.e Lewis form factor

$Y= 0.484 - 2.87/Z$ (for $20^0$ full- depth involute)
$Y = 0.55 – 2.64/Z$ (for $20^0$ stub involute)
$Y = 0.39 – 2.15/Z$ (for $14.5^0$ full- depth involute)

III. Find beam strength;
$F_b = б_b * b * m * Y$
Assume b=10*m for spur gear.

IV. Find wear strength;
$F_w = d_p * b * Q * K$
Where,
$d_p = m * z_p$
Assume,
$z_p$=18 to avoid interference.
$Q = (2 Z_g)/ Z_g + Z_P$ Ratio factor for external gear pair
$Q = (2 Z_g)/ Z_g - Z_P$ Ratio factor for internal gear pair
Calculate load stress factor
$K = 0.16 (BHN/100)^2$ For steel pinion and steel gear
$K = 0.21 (BHN/100)^2$ For cast iron pinion and cast iron gear
$K = 0.18 (BHN/100)^2$ For steel pinion and cast iron gear
Calculate effective load
$F_{eff} = (K_a * K_m * F_t)/K_v$
Where,
$F_{eff} = P/V$
$V=(л * d_p * n_p)/(60*1000)$
From design data book one can found values of,
$K_a$ – Application or Service Factor.
$K_m$ – Load distribution Factor

| . | Driven Machinery | | |
|---|---|---|---|
| Source of Power | Uniform | Moderate Shock | Heavy Shock |
| Uniform | 1.00 | 1.25 | 1.75 |
| Moderate Shock | 1.25 | 1.50 | 2.00 |
| Heavy Shock | 1.5 | 1.75 | 2.25 |

Table: Service Factor Selection

$K_v$ is velocity factor. It is specified in terms of quality of gear and pitch velocity. For ordinary cut gears velocity factor can be given as,
$K_v= 3/(3+V)$
If $F_b > F_w$ then gear pair is weaker in pitting.
Find module of gear by equation,
$F_w= N_f * F_{eff}$
If $F_b < F_w$ then gear pair is weaker in bending.
$F_b= N_f * F_{eff}$

V. Selection Of Standard Module
The obtained value of module from above equation is then round off to the next available value of module from recommended series of module.

| Choice-1 Preferred | | Choice-2 | |
|---|---|---|---|
| 1.0 | 5.0 | 1.125 | 5.5 |
| 1.25 | 6.0 | 1.375 | 7 |
| 1.5 | 8.0 | 1.75 | 9 |
| 2.0 | 10 | 2.25 | 11 |
| 2.5 | 12 | 2.75 | 14 |
| 3.0 | 16 | 3.5 | 18 |
| 4.0 | 20 | 4.5 | |

Table: Recommended series of module (mm)

VI. Fix The Dimensions Of The Gear Pair Using Following Empirical Relations.
b= 10*m
$d_p = z_p * m$
$C.D=(d_p+d_g)/2$
$h_a=1*m$
$h_d=1.25*m$
check for design whether it is safe or not

  

VII. Precise Estimation Of Dynamic Load By Buckingham's Equation;

$F_d$= 21V((b*C)+$F_{tmax}$) / 21V+√(b*C)+√ $F_{tmax}$
$F_{tmax}$= $K_a$*$K_m$*$F_t$
Deformation factor C
C= 11500*e N/mm for steel pinion and steel gear
C= 8900*e N/mm for cast iron pinion and cast iron gear
C= 10000*e N/mm for steel pinion and cast iron gear
Pitch error on meshing teeth (e) can be found by equation
$e=e_p+e_g$
Where 'e' can be found by equation as per different grades. The error depends upon gear size, gear grade, and tooth size. The expected pitch error on the gear or pinion teeth is considered to be equal to tolerance.

The method of manufacturing the gear depends on the grade of the gear. Gears of Grade 11 and 12 are manufactured by casting. Gears of Grade 8 and 9 require rough and fine hobbing . Gears of Grade 6 are obtained by hobbing and rough grinding, while Grade 4 requires shaving and finish grinding.
For e.g. for gear of Grade 6,
e= 8.00+0.63(m+√d) (microns)
check for available factor of safety,
$F_{eff}$= $K_a$*$K_m$*$F_t$ + $F_d$
If $F_b$ >$F_w$, then,
$N_f$= $F_w$/$F_{eff}$
If $F_b$ <$F_w$, then,
$N_f$= $F_b$/$F_{eff}$
If the available factor of safety is higher than the required factor of safety, the design is safe.

NOTE: It is important to note that if design is not safe (i.e. available factor of safety is less than the required factor of safety), the design should be modified by increasing the module to the next standard value and checked again for safety by precise estimation of dynamic load.

## IV. CONCLUSION

There is limiting value of minimum number of teeth on pinion. As the number of teeth decreases there is possibility of interference to occur. If the face width is to large, there is possibility of concentration of load at one end of the gear tooth and if face width is too small it will have poor capacity to resist the shock. Hence in practice the optimum range of face width is (8*m)<b<(12*m). Higher value of module will leads to more strong gear pair and hence available factor of safety will be more. Module under choice I is preferred than choice II. If face width, speed and normal module except gear ratio are kept constant and gear ratio is increased the corresponding beam strength remains constant. Above process of designing a spur gear pair is simplest and can be used for almost any type of spur gear pair design for any practical application if input and output speed or input and output torque is known.

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