DESIGN OF BEVEL GEAR PAIR FOR GATE VALVE

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Abstract

Bevel gears are widely used in the application where requirement is to transmit the power in 90 degrees. Where shafts intersect each other, the basic need is fulfilled by the bevel gear pair only. These gears are used in many applications and the most common application is differential use in automotives. This paper mainly focuses on the design and mathematical modeling of the bevel gear pair used in gate valve unit. Gate valve is valve which opens by lifting a round or rectangular gate/wedge out of the path of the fluid. To operate this gate valve the main component used is bevel gear pair. It is needed to check the maximum torque transmission capacity of gear pair so as to know its limits. Involute gear is used in this application. Gate valves are actuated by a threaded stem which connects the actuator (e.g. handwheel or motor) to the gate.

Keywords: *Involute Bevel Gear, Gate Valve.*

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1. INTRODUCTION

A gate valve, which is also known as a sluice valve, is a valve which opens up by lifting a round or rectangular gate or wedge out of the path of the fluid. The distinct feature of a gate valve is the sealing surfaces between the gate and seats are planar, so gate valves are often used when a straight-line flow of fluid and minimum restriction is desired. The gate faces can be parallel, but are most commonly wedge-shaped. Gate valves are primarily used to permit or prevent the flow of liquids, but typical gate valves shouldn't be used for regulating flow, unless they are specifically designed for that purpose. Because of their ability to cut through liquids, gate valves are often used in the petroleum industry. To operate these gate valves the important element is bevel gear pair. Bevel gears are widely used because of their suitability towards transferring power between nonparallel shafts at almost any angle or speed.

For solving some complex problems regarding the generation of the gears with crossed axes, are known more techniques. There are taken into account the undercutting and sharpening of the gear's teeth, as well as correct positioning of the contact area between teeth sides.[1]

To find out the maximum torque transmission capacity of bevel pair is the prime requirement, as these are used in gate valve and need to sustain that load. So as to cope up with this requirement the design is to be made.

1.1 Bevel Gear Pair Requirement

Bevel gears are useful when the direction of a shaft's rotation needs to be changed. They are usually mounted on shafts that are 90 degrees apart, but can be designed to work at other angles as well. The pitch surface of bevel gears is a cone.

The teeth on bevel gears can be straight, spiral or hypoid. Straight bevel gear teeth actually have the same problem as straight spur gear teeth as each tooth engages, it impacts the corresponding tooth all at once. On straight and spiral bevel gears, the shafts must be perpendicular to each other, but they must also be in the same plane. Straight-bevel gears are the simplest form of bevel gears. The teeth are straight and tapered, and if extended inward, they would pass through the point of intersection of the axes.

A basic load and a suitable factor encompassing protection from intermittent overloads, desired life, and safety are determined from
1. The power rating of the prime mover, its overload potential, and the uniformity of its output torque.
2. The normal output loading, peak loads and their duration, and the possibility of stalling or severe loading at infrequent intervals.
3. Inertia loads arising from acceleration or deceleration.

2. DESIGN OF BEVEL GEAR PAIR

A) Input parameter:
Number of teeth on pinion = Z_1 =12
Number of teeth on gear = Z_2=52
Module= m = 4mm
Pressure angle = $\Psi = 20°$
Shaft angle = $\sum = 90°$

B) Detail dimensions of bevel gear pair:
1) Reference diameter(d):
Diameter of pinion=d_1 = $m\times Z_1 = 4\times 12 = 48$mm
Diameter of gear = $d_2 = m\times Z_2 = 4\times 52 = 208$mm

2) Pitch angle(δ):
Pitch angle of pinion=$\delta_1 = \tan^{-1}(\frac{Z_1}{Z_2}) = \tan^{-1}(\frac{12}{52})$
δ_1 =12.994°

Pitch angle of gear=$\delta_2=\tan^{-1}(\frac{Z_2}{Z_1})=\tan^{-1}(\frac{52}{12})= 77.006°$

Cross check,
Shaft angle = $\sum= 90°$
$\sum=\delta_1 + \delta_2= 12.994° + 77.006° = 90°$

3) Outer cone distance(R):
$R=R_1=R_2=\frac{d_1}{2} \times \operatorname{cosec}\delta_1$
$R=\frac{48}{2} \times \operatorname{cosec} 12.994°=106.738mm$

4) Addendum modification coefficient(K_a):
For pinion,
$K_{a1}=0.460\times [1-(\frac{Z_1}{Z_2})^2]$
$K_{a1}= 0.460\times [1-(\frac{12}{52})^2]$
$K_{a1} = 0.435$
For gear,
$K_{a2}= -K_{a1}$
$K_{a2}= -0.435$

5) Tooth width modification coefficient(K_w):
The value of $2K_{w1} \tan\Psi$ is 0.110,
$2K_{w1} \tan\Psi=0.110$
$2 K_{w1} \times \tan 20°=0.110$
For pinion,
$K_{w1}= 0.151$
For gear,
$K_{w2}= - K_{w1}$
$K_{w2}= -0.151$

6) Equivalent tooth modification coefficient(K_{eq}):
For pinion,
$K_{eq\,1}=K_{a1}+K_{w1}= 0.435+0.151= 0.586$

For gear,
$K_{eq\,2}= -K_{eq\,1}= -0.586$

7) Addendum(h_a) :
For pinion,
$h_{a1}= m\times(1+K_{a1})= 4\times(1+0.435)= 5.74mm$

For gear,
$h_{a2}= m\times(1+K_{a2})= 4\times(1-0.435)= 2.26mm$

8) Dedendum(h_f):
For pinion,
$h_{f1} = m\times(1.2-K_{a1})= 4(1.2-0.435)= 3.06mm$

For gear,
$h_{f2}= m\times(1.2-K_{a2})= 4(1.2+0.435)= 6.54mm$

9) Face width(b):
$b= b_1=b_2=0.3\times R= 0.3\times106.738= 32.03mm$

10) Dedendum angle(x_f):
For pinion,
$x_{f1}= \tan^{-1}(\frac{h_{f1}}{R})= \tan^{-1}(\frac{3.06}{106.738})= 1.642°$

For gear,
$x_{f2}= \tan^{-1}(\frac{h_{f2}}{R})=\tan^{-1}(\frac{6.54}{106.738})= 3.506°$

11) Root angle(δ_f):
For pinion,
$\delta_{f1} =\delta_1- x_{f1}= 12.994°-1.642°=11.352°$

For gear,
$\delta_{f2} =\delta_2- x_{f2}=77.006° - 3.506° = 73.5°$

12) Face angle of blank(δ_a):
For pinion,
$\delta_{a1} = \delta_1+x_{f2} = 12.994°+3.506° = 16.5°$

For gear,
$\delta_{a2} = \delta_2+x_{f1}= 77.006° + 1.642° = 78.648°$

13) Outside diameter of blank(d_a):
For pinion,
$d_{a1}=d_1+2 h_{a1} \times \cos\delta_1=48+2\times 5.74 \times \cos 12.994°=53.59mm$

For gear,
$d_{a2} = d_2+2 h_{a2} \times \cos\delta_2=208+2\times 2.26 \times \cos 77.006°=209.01mm$

14) Apex to crown distance(h_p):
For pinion,
$h_{p1}= R\times \cos\delta_1- h_{a1} \times \sin\delta_1=106.73\cos 12.994° - 5.67\sin12.994°= 102.706mm$

For gear,
$H_{p2}=R \times \cos\delta_2 - h_{a2} \times\sin\delta_2$
$=106.73\times\cos77.006°- 2.26\times\sin77.006°= 21.79mm$

15) Circular tooth thickness at the reference diameter(S):
For pinion,
$s_1= m\times(\frac{\pi}{2}+2K_{eq\,1} \times \tan\Psi) = 4\times(\frac{\pi}{2} + 2 \times 0.586 \times \tan20°)= 7.99mm$

For gear,
$s_2= m\times(\frac{\pi}{2}+2K_{eq\,2} \times \tan\Psi)= 4\times(\frac{\pi}{2} - 2 \times 0.586 \times \tan20°) = 4.58mm$

16) Virtual spur gear diameter(d_v):
For pinion,
$d_{v1}= d_1 \times\sec\delta_1= 48\times\sec12.994°= 49.27mm$

For gear,
$d_{v2}= d_2 \times\sec\delta_2= 208\times\sec77.006°= 925mm$

17) Virtual spur gear teeth(Z_v):
For pinion,
$Z_{v1}= \frac{d_{v1}}{m}= \frac{49.27}{4}= 12.31$

For gear,
$Z_{v2}= \frac{d_{v2}}{m}= \frac{925}{4}= 230.25$

18) Chordal tooth thickness(g_c):
For pinion,
$g_{c1}= d_{v1} \times \sin(\frac{s_1}{d_{v1}} \text{rad}) = 49.26 \times \sin(\frac{7.89}{49.26} \text{rad}) =$ 7.86mm

For gear,
$g_{c2}= d_{v2} \times \sin(\frac{s_2}{d_{v2}} \text{rad}) = 925 \times \sin(\frac{4.57}{925} \text{rad}) =$ 4.56mm

19) Chordal height(h_c):
For pinion,
$h_{c1}= h_{a1}+\frac{g_{c1}^2}{4d_{v1}} = 5.74+\frac{7.856^2}{4\times49.26} =$ 6.05mm

For gear,
$h_{c2}= h_{a2}+\frac{g_{c2}^2}{4d_{v2}} = 2.26+\frac{4.56^2}{4\times925} =$ 2.26mm

20) Constant chord(g_{cc}):
For pinion,
$g_{cc1}= s_1 \times \cos^2\Psi = 7.98\times \cos^2 20° =$ 7.04mm

For gear,
$g_{cc2}= s_2 \times \cos^2\Psi = 4.57\times \cos^2 20° =$ 4.03mm

21) Constant chord thickness(h_{cc}):
For pinion,
$h_{cc1}= h_{a1} - \frac{s_1\times\sin 2\Psi}{4} = 5.74 - \frac{7.98\times\sin 40}{4} =$ 4.45mm

For gear,
$h_{cc2}= h_{a2} - \frac{s_2\times\sin 2\Psi}{4} = 2.26 - \frac{4.57\times\sin 40}{4} =$ 1.52mm

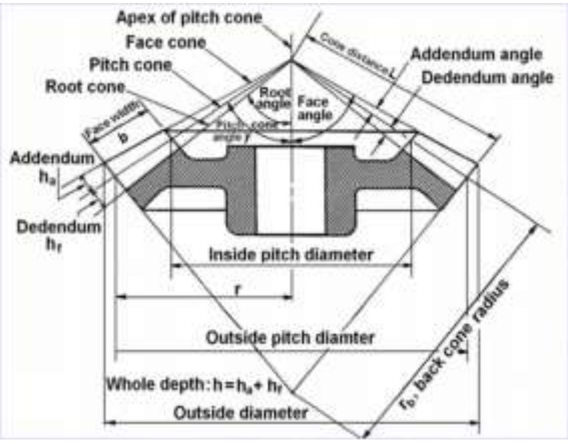

Fig -1: Bevel gear pair

3. FORCE ANALYSIS

Power transmitted= 1.5kW,
Input speed= 190rpm

1) Average pitch radius(d_{av}):
$D_{av} = d_1 - b\times\sin\delta_1$
= 48-32.03×sin12.994°=40.80mm

2) Pitch line velocity(V_{av}):
$V_{av} = \frac{\pi\times d_1\times n_1}{60000} = \frac{\pi\times 40.80\times190}{60000} =$ 0.4059 m/sec

3) Transmitted tangential force(F_t):
$F_t = \frac{1000\times w}{v_{av}} = \frac{1000\times1.5}{0.405} =$ 3703.70 N

4) Torque transmitted(T):
$T= F_t \times \frac{d_{av}}{2}$
$T=F_t \times r_{av}$ = 3703×20.40= 75541 N-mm T=75.54 N-m

5) Tangential force(P_t):
$P_t = \frac{T}{r_{av}} = \frac{75541}{20.40} =$ 3703 N= 3.703kN

6) Radial force(P_r):
$P_r = P_t \times \tan\Psi \times \cos\delta_1 = 3703\times \tan 20° \times \cos 12.994° =$ 1313.26N = 1.313kN

7) Axial force(P_a):
$P_a = P_t \times \tan\Psi \times \sin\delta_1 = 3703\times \tan 20° \times \sin 12.994° = 303.04\text{N} = 0.3\text{kN}$

3.1 Bevel Gear Stresses and Strength Analysis:

1) Dynamic factor (K_v):

Table -1: Overload factors

Character of prime mover	Character of load on driven machine			
	Uniform	Light shock	Medium shock	Heavy shock
Uniform	1.00	1.25	1.50	1.75 or higher
Light shock	1.10	1.35	1.60	1.85 or higher
Medium shock	1.25	1.50	1.75	2.00 or higher
Heavy shock	1.50	1.75	2.00	2.25 or higher

Pitch line velocity (V_{av}):
Character of prime mover: Light shock
Character of load on driven machine: Uniform
K_v= 1.10 From table 1
Q_v= 11 From fig.2

Also we have,
$K_v = (\frac{A+\sqrt{200v_a}}{A})^B$
A= 50+56(1-B)
$B= 0.25(12 - Q_v)^{\frac{2}{3}}$
As we have Q_v= 11,

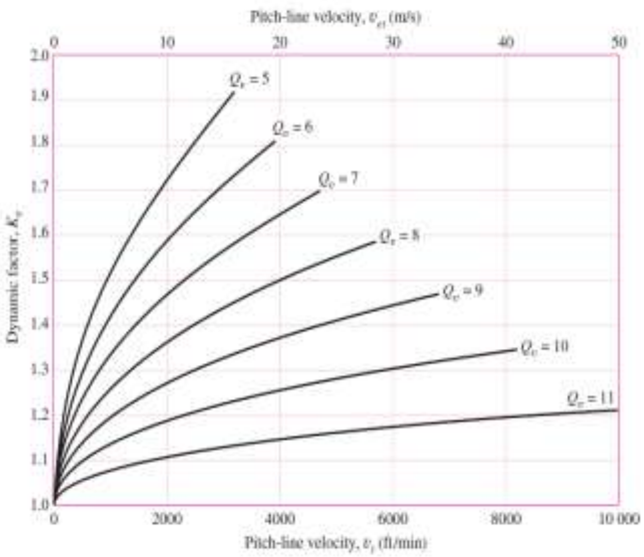

Fig -2: Dynamic factor K_v

$B = 0.25(12 - 11)^{\frac{2}{3}} = 0.25$

A= 50+56(1-0.25) = 92

Putting these values of A & B in above equation,

$K_v = (\frac{A+\sqrt{200v_a}}{A})^B = = (\frac{92+\sqrt{200\times0.405}}{92})^{0.25} = 1.05$

2) Maximum recommended pitch line velocity (V_{amax}):
Maximum recommended pitch-line velocity is associated with the abscissa of the terminal points of the curve in fig. 2:

$$V_{amax} = (\frac{A+(Q_v-3)}{200})^2 = (\frac{92+(11-3)}{200})^2 = 50 \text{ m/sec}$$

3) Size factor for pitting resistance (Z):
$Z_x = 0.00492\times b+0.4375$
for $12.7mm \le b \le 114.3$mm
$Z_x = 0.00492 \times 32.03+0.4375$
$Z_x = 0.7321$

4) Size factor for bending (Y_x):
$Y_x = 0.4867+0.008339\times m$
for $1.6mm \le m \le 50mm$
$Y_x = 0.4867+0.008339\times 4 = 0.520056$
$Y_x = 0.520056$

5) Load-distribution factor ($K_{H\beta}$):
$K_{H\beta} = K_{mb} + 5.6(10^{-6})b^2$
Where,
$K_{mb} = 1.10$ for one member straddle-mounted
$K_{H\beta} = 1.10+5.6(10^{-6}) \times 32.03^2$
$K_{H\beta} = 1.105$

6) Crowing factor for pitting (Z_{xc}):
The teeth of most bevel gears are crowned in the lengthwise direction during manufacture to accommodate to the mountings.

$Z_{xc} = 1.5$

For properly crowned teeth.

7) Lengthwise curvature factor for bending strength (Y_β):
$Y_\beta = 1$ for straight-bevel gears

8) Pitting resistance geometry factor (Z_I):
Fig. 3 shows the geometry factor Z_I for straight-bevel gears with a 20° pressure angle and 90° shaft angle. Enter the figure ordinate with the number of pinion teeth, move to the number of number of gear-teeth contour, and read from the abscissa.

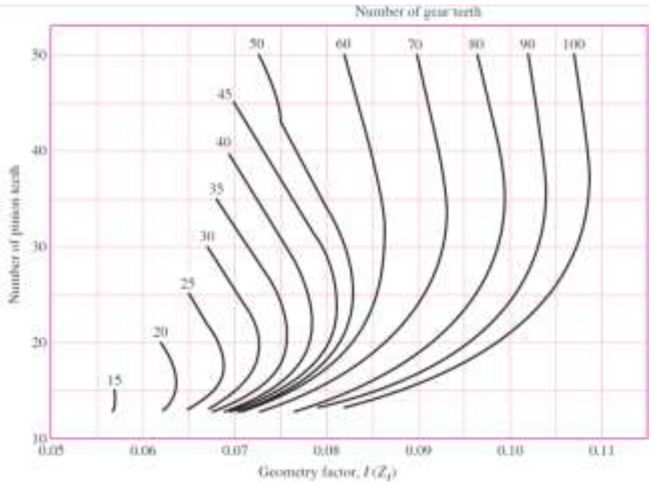

Fig -3: Contact geometry factor (Z_I) for coniflex straight-bevel gears with a 20° normal pressure and a 90° shaft angle

$Z_I = 0.075$ from fig. 3

9) Bending strength geometry factor (Y_J):

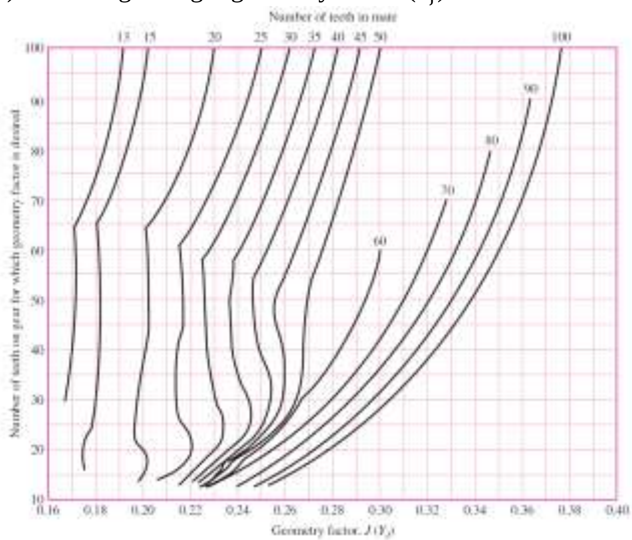

Fig. -4: shows the geometry factor J for straight-bevel gears with a 20° pressure angle and 90° shaft angle .

$Y_J = 0.18$ from fig. 4

10) Stress-cycle factor for pitting resistance (Z_{NT}):
$Z_{NT} = 3.4822\times\ n_L^{-0.0602}$ $10^4 \le n_L \le 10^{10}$
Where n_L = Number of load cycles
In this case, $n_L = 10^6$
$Z_{NT} = 3.4822\times (10^6)^{-0.0602} = 1.515$
$Z_{NT} = 1.515$

11) Stress-cycle factor for bending strength (Y_{NT}):

$Y_{NT} = 6.1514 \times n_L^{-0.1182}$

$10^3 \leq n_L \leq 3(10^6)$

Where n_L =Number of load cycles

In this case, $n_L = 10^6$

$Y_{NT} = 6.1514 \times (10^6)^{-0.1182}$ =1.2013

$Y_{NT} = 1.2013$

12) Material specification:

A) Pinion:

AISI 4140 Low alloy steel

Tensile strength= 655mpa

Yield strength= 517mpa

Brinell Hardness Number(BHN) = 207-235

Quenching= 120°F (water)

Chemical composition elements :

Carbon, Manganese, Phosphorus, Sulphur, Silicon, Nickel, Chromium, Molybdenum, vanadium .

B) Gear:

AISI 1040 Low alloy steel

Tensile strength= 551mpa

Yield strength= 330mpa

Brinell Hardness Number(BHN) = 150-235

Quenching= 120°F (water)

Chemical composition elements :

Carbon, manganese, Phosphorus, sulphur .

13) Hardness ratio factor (Z_W):

H_{B1}=Minimum Brinell hardness number for pinion material= 207

H_{B2}=Minimum Brinell hardness number for gear material= 150

$Z_W = 1 + B_1 \times (\frac{Z_1}{Z_2} - 1)$

Where $B_1 = 0.00898 \times (H_{B1}/H_{B2}) - 0.00829$

$B_1 = 0.00898 \times (207/150) - 0.00829$

$B_1 = 4.1024 \times 10^{-3}$

$Z_W = 1 + B_1 \times (\frac{Z_1}{Z_2} - 1)$

$Z_W = 1 + 4.1024 \times 10^{-3} \times (\frac{12}{52} - 1)$

$Z_W = 0.9968$

14) Temperature factor (K_θ):

$K_\theta = 1$ for $32°F \leq \theta \leq 250°F$

In this case

$K_\theta = 1$ As $\theta = 120°F$

15) Reliability factors (Z_z & Y_z):

Table: 2 Reliability Factors

Requirements of application	Reliability factors for steel	
	Z_Z	Y_Z
Fewer than one failure in 10000	1.22	1.5
Fewer than one failure in 1000	1.12	1.25
Fewer than one failure in 100	1	1
Fewer than one failure in 10	0.92	0.85
Fewer than one failure in 2	0.84	0.70

Considering the one failure fewer than one in 100 .Thus from table 2 the values of Z_Z and Y_Z is 1.

$Z_Z = 1$

$Y_Z = 1$

16) Allowable contact stress number ($\sigma_{H\ \lim}$):

Table: 3 Allowable contact stress number for steel gears, $\sigma_{H\ \lim}$

Material Designation	Heat Treatment	Minimum Surface Hardness	Allowable contact stress number, $\sigma_{H\ \lim}$ (N/mm^2) Grade 1	Grade 2	Grade 3
Steel	Through - hardened Flame or induction hardened	50 HRC	175000 (1210)	190000 (1310)	200000 (1380)
AISI 4140	Nitrided	84.5HR 15N		145000 (1000)	
Nitralloy 135M	Nitrided	90HR 15N		160000 (1100)	

Z_E = elastic coefficient, 190 N/mm^2 for steel

And the equations are as follow,

$\sigma_{H\ \lim} = 2.35\ H_B + 162.89$ MPa grade 1

$\sigma_{H\ \lim} = 2.51\ H_B + 203.86$ MPa grade 2

In our case material is made up of ANSI 4140 and the corresponding grade is 2.

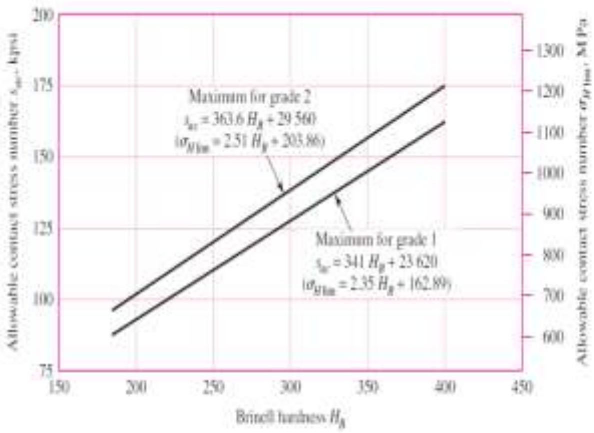

Fig. 5 Allowable contact stress number for through-hardened steel gears, $\sigma_{H\ \lim}$.

To find out the value of H_B there is graph as above.

From fig. 5, $H_B = 325$

$\sigma_{H\ \lim} = 2.51\ H_B + 203.86$ MPa for grade 2

$\sigma_{H\ \lim} = 2.51 \times 325 + 203.86$ MPa = 1016.36 MPa

17) Allowable bending stress number ($\sigma_{F\ \lim}$):

Table: 4 Allowable bending stress number for steel gears, $\sigma_{\mathrm{F\ lim}}$

Material Designation	Heat Treatment	Minimum Surface Hardness	Allowable bending stress number, $\sigma_{\mathrm{F\ lim}}$ ($\mathrm{N/mm^2}$) Grade 1	Grade 2	Grade3
Steel	Through-hardened Flame or induction hardened	50 HRC	15000 (85)	13500 (95)	12000 (105)
AISI 4140	Nitrided	84.5HR 15N		22000 (150)	
Nitralloy 135M	Nitrided	90HR 15N		24000 (165)	

Z_E = elastic coefficient, 190 $\mathrm{N/mm^2}$ for steel
And the equations are as follow,
$\sigma_{\mathrm{F\ lim}} = 0.30\ H_B + 14.48$ MPa grade 1
$\sigma_{\mathrm{F\ lim}} = 0.33\ H_B + 41.24$ MPa grade 2
In our case material is made up of ANSI 4140 and the corresponding grade is 2.
To find out the value of H_B there is graph as follow.
From fig. 6 , $H_B = 320$
$\sigma_{\mathrm{F\ lim}} = 0.33\ H_B + 41.24$ MPa grade 2
$\sigma_{\mathrm{F\ lim}} = 0.33 \times 320 + 41.24$ MPa = 146.84 MPa

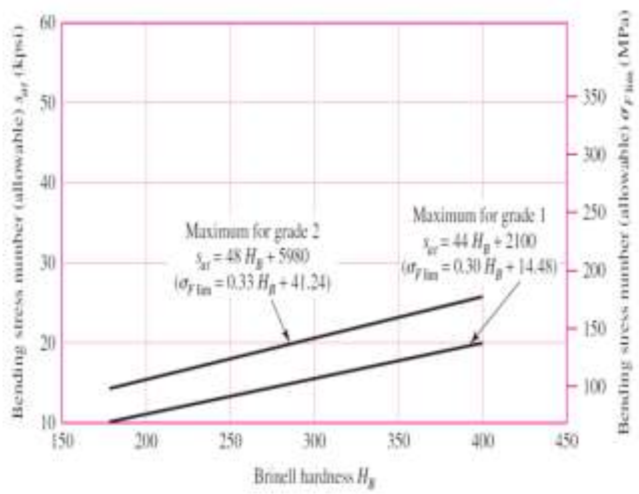

Fig.6 Allowable bending stress number for through-hardened steel gears, $\sigma_{F\ \lim}$.

18) Fundamental contact stress (σ_{H}):
Fundamental contact stress equation is as follow,

$$\sigma_{\mathrm{H}} = Z_E \times \left(\frac{1000 \times W \times K_A \times K_v \times K_{H\beta} \times Z_X \times Z_{XC}}{b \times d \times Z_I}\right)^{1/2}$$

$$\sigma_{\mathrm{H}} = 190 \times \left(\frac{1000 \times 1.5 \times 1 \times 1.1 \times 1.105 \times 0.732 \times 1.5}{32.03 \times 48 \times 0.075}\right)^{1/2}$$

σ_{H} = 791.80 $\mathrm{N/mm^2}$

19) Permissible contact stress (σ_{HP}):
Permissible contact stress number equation is as follow,

$$\sigma_{\mathrm{HP}} = \frac{\sigma_{H\ \lim} \times Z_{NT} \times Z_W}{S_H \times K_\theta \times Z_Z}$$

$$\sigma_{\mathrm{HP}} = \frac{1016.36 \times 1.515 \times 0.9968}{1 \times 1 \times 1} = 1534.85\ \mathrm{N/mm^2}$$

$$\sigma_{\mathrm{HP}} = 1534.85\ \mathrm{N/mm^2}$$

Here, $\sigma_{\mathrm{HP}} > \sigma_{\mathrm{H}}$ clearly shows that the contact stresses are under permissible limit.

20) Bending stress (σ_{F}):
Bending stress equation is as follow,

$$\sigma_{\mathrm{F}} = \frac{1000 \times \mathrm{w} \times \mathrm{K_A} \times \mathrm{K_v} \times \mathrm{Y_X} \times \mathrm{K_{H\beta}}}{\mathrm{b} \times \mathrm{m} \times \mathrm{Y_\beta} \times \mathrm{Y_J}}$$

$$\sigma_{\mathrm{F}} = \frac{1000 \times 1.5 \times 1 \times 1.1 \times 0.52005 \times 1.105}{32.03 \times 4 \times 1 \times 0.18}$$

$$\sigma_{\mathrm{F}} = 41.11\ \mathrm{N/mm^2}$$

22) Permissible bending stress (σ_{FP}):

$$\sigma_{\mathrm{FP}} = \frac{\sigma_{\mathrm{F\ lim}} \times \mathrm{Y_{NT}}}{\mathrm{S_F} \times \mathrm{K_\theta} \times \mathrm{Y_Z}}$$

$$\sigma_{\mathrm{FP}} = \frac{146.84 \times 1.2013}{1 \times 1 \times 1} = 176.39\ \mathrm{N/mm^2}$$

$$\sigma_{\mathrm{FP}} = 176.39\ \mathrm{N/mm^2}$$

Here, $\sigma_{\mathrm{FP}} > \sigma_{\mathrm{F}}$ clearly shows that the bending stresses are under permissible limit.

As, $\sigma_{\mathrm{FP}} > \sigma_{\mathrm{F}}$ and $\sigma_{\mathrm{HP}} > \sigma_{\mathrm{H}}$, it shows that design is safe.

4. RESULTS AND DISCUSSION

Result shows that the design is safe under varying load conditions. The maximum contact stress is found out to be 791.80 $\mathrm{N/mm^2}$ and permissible contact stress is 1534.85 $\mathrm{N/mm^2}$. It clearly shows that the design is safe. Bending stress also found out to be 41.11 $\mathrm{N/mm^2}$ and permissible bending stress is 176.39 $\mathrm{N/mm^2}$ which means the design can withstand against such loads and design is safe.

5. CONCLUSION

In this paper the bevel gear pair is designed for gate valve application. Results shows that the design is safe and can withstand against certain loads under certain conditions. We have find out the maximum torque transmission capacity and it can transmit 75.54 N-m of torque under maximum load conditions.

REFERENCES

[1]. N.Mohan Raj, et al. "Design of Contact Stress Analysis in Straight Bevel Gear" PG Scholar, Dept.of Mech.Engg,, Sri Krishna college of Technology, Kovaipudur, Tamilnadu, India-641042 2Assistant Professor, Dept.of Mech.Engg, Sri Krishna college of Technology, Kovaipudur, Tamilnadu, India-42

[2]. M. Li, et al. "Dynamic analysis of spiral bevel geared rotor bearing system" Institute of Vibration Engineering Research, Nanjing University of Aeronautics and Astronautics.

[3]. Faydor L. Litvin, et al."Computerized design, simulation of meshing, and contact and stress analysis of face-milled formate generated spiral bevel gears", Department of Mechanical Engineering, Gear Research Center, University of Illinois at Chicago, Chicago, IL 60607-7022, USA.

[4]. John Argyris, et al. "Computerized integrated approach for design and stress analysis of spiral bevel gears", The university of Illinois at Chicago.

[5]. Jixin Wang,et al."The Mathematical Model of Spiral Bevel Gears - A Review", Jilin University, School of Mechanical Science and Engineering, China .

[6]. M. Kolivand, et al. "Actual Tooth Contact Analysis of Straight Bevel Gears" American Axle and Manufacturing",

[7]. Ning Ma and Wenji Xu "Mathematical modeling for finishing tooth surfaces of spiral bevel gears using pulse electrochemical dissolution".

[8]. Wern-Kueir Jehng " Computer solid modeling technologies applied to develop and form mathematical parametric tooth profiles of bevel gear and skew gear sets" Department of Industrial Engineering and Management, National Kaohsiung University of Applied Sciences, Kaohsiung 807, Taiwan, ROC.

[9]. Vladimir I. Medvedev et al. ,"Mathematical model and algorithm for contact stress analysis of gears with multi-pair contact" Chair of Theoretical Mechanics, Moscow State University of Technology, Moscow Russia.

[10]. Isamu Tsuji et al. "Tooth Contact Analysis and Manufacture on Multitasking Machine of Large-Sized Straight Bevel Gears With Equi-Depth Teeth".