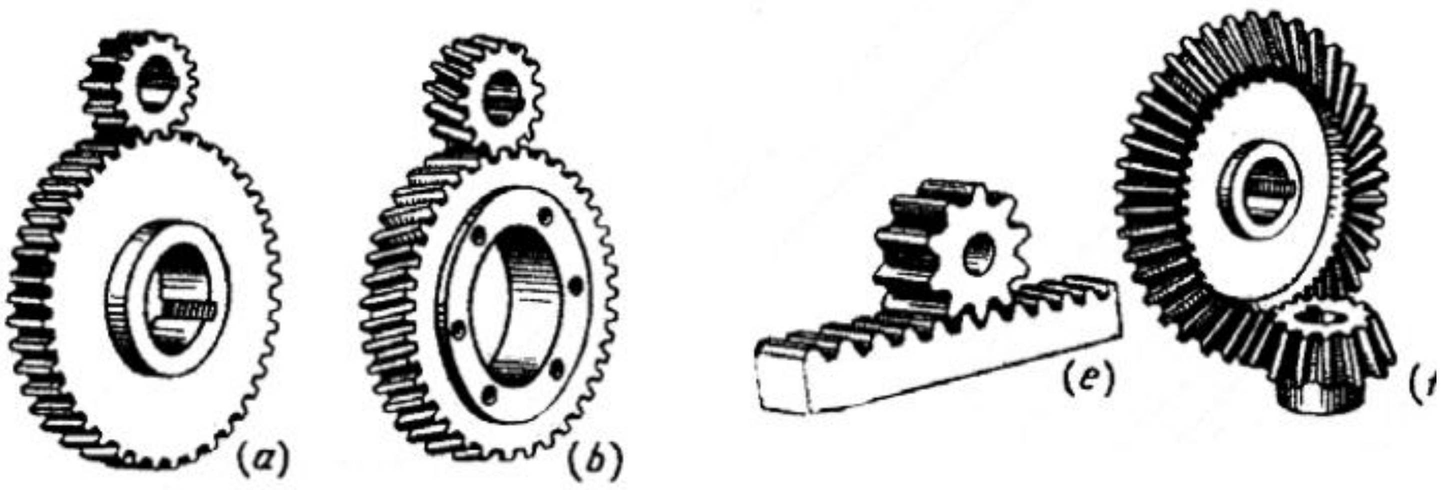
ME 308
MACHINE ELEMENTS II

CHAPTER 5

GEARS_part4



While LEWIS equation is used for static bending stress calculation AGMA equation is used for fatigue condition & gives the bending stress in tooth root under a force of Wt acting tangent to the pitch circle and including effects of stress concentration (J).

For a fatigue-free safe operation the bending stress (σ) obtained from AGMA equation should be compared with the endurance strength (Se) of the gear material with a global safety factor n_G, that is;

$$\sigma = \frac{Wt}{F * m * J * Kv} \leq S_e / n_G$$

$$S_e = k_a . k_b . k_c . k_d . k_e . k_f \, x S_e^{'}$$

For steels:

$$S_e' = 0.5 S_{ut} \quad \text{If } S_{ut} \leq 1400 MP_a$$

$$S_e' = 700 MP_a \quad \text{If } S_{ut} > 1400 MP_a$$

For CI:

$$S_e^{'} = 0.45 S_{ut}$$

$$S_e^{'} = 275 MP_a$$

For factor ka, use machined surface always since the tooth root is always in machined or cast form even if the tooth flank is ground.

<u>If the gear material is CI</u>

The Se' values given in Table A-21 are fully corrected for surface factors (ka), thus use ka=1 but not corrected for other factors.

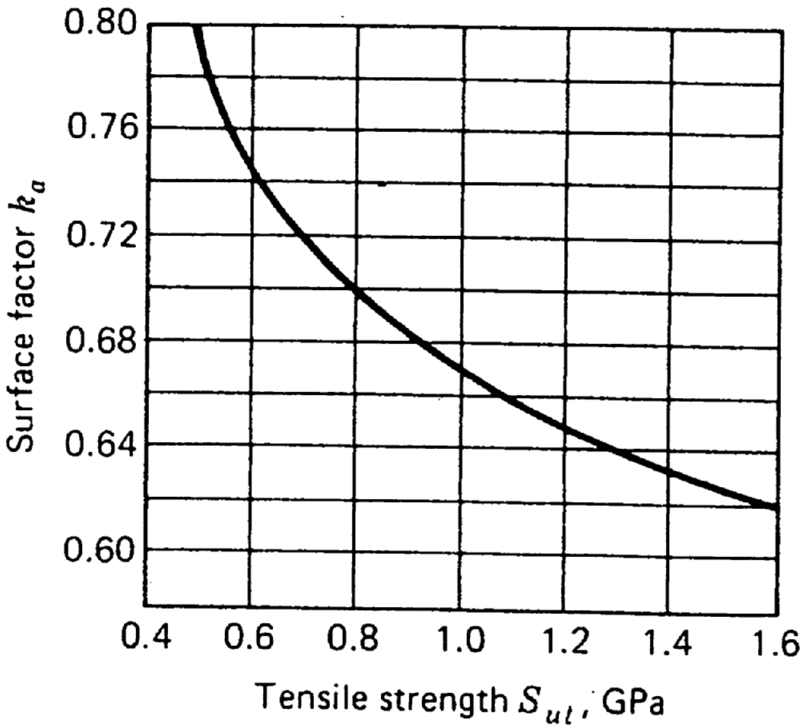


FIGURE 13-25 Surface-finish factors k_a for cut, shaved, and ground gear teeth.

For factor kb, Eq. 7-16 is generally used but: in this equation the dimension d is the diameter of a round specimen.

A spur gear tooth has a rectangular cross section and so the method of Sec. 7-7 must be used to get an equivalent value for d.

For a rectangular cross section the formula for the equivalent diameter is

$$d = 0.808(hb)^{1/2}$$

By using different module values h & b were determined and then kb values were calculated.

The results for kb were then simply tabulated in Table 13-7 for different modules, and kb values are taken from Table 13-7.

Size The size factor, from Eq. (7-16), is

$$k_b = \begin{cases} 1 & d \le 8 \text{ mm} \\ 1.189 d^{-0.097} & 8 \text{ mm} < d \le 250 \text{ mm} \end{cases}$$

Table 13-7 SIZE FACTORS FOR SPUR-GEAR TEETH (Preferred modules in bold face

Module m	Factor k_b	Module m	Factor k_b
1 to 2	1.000	11	0.843
2.25	0.984	**12**	0.836
2.5	0.974	14	0.824
2.75	0.965	**16**	0.813
3	0.956	18	0.804
3.5	0.942	**20**	0.796
4	0.930	22	0.788
4.5	0.920	**25**	0.779
5	0.910	28	0.770
5.5	0.902	**32**	0.760
6	0.894	36	0.752
7	0.881	**40**	0.744
8	0.870	45	0.736
9	0.860	**50**	0.728
10	0.851		

For factor kc, Table13-8 is used.

Table 13-8 RELIABILITY FACTORS

Reliability R	0.50	0.90	0.95	0.99	0.999	0.9999
Factor k_c	1.000	0.897	0.868	0.814	0.753	0.702

kd=1.0 for T<=350°C

ke =1.0 since stress concentration due to fillet geometry is included in the geometry factor J.

Regarding factor kf, since most of the time gears rotate only in one direction the loading type is repeated, therefore tooth is subjected to one-way bending.

Since endurance limits Se’ and Se are calculated for two-way bending specimens the actual endurance limits has to be increased for one-way bending teeth and kf is taken from Fig 13-26.

For two-way bending (both direction rotating or idler) gears, however, k_f is taken as 1.0

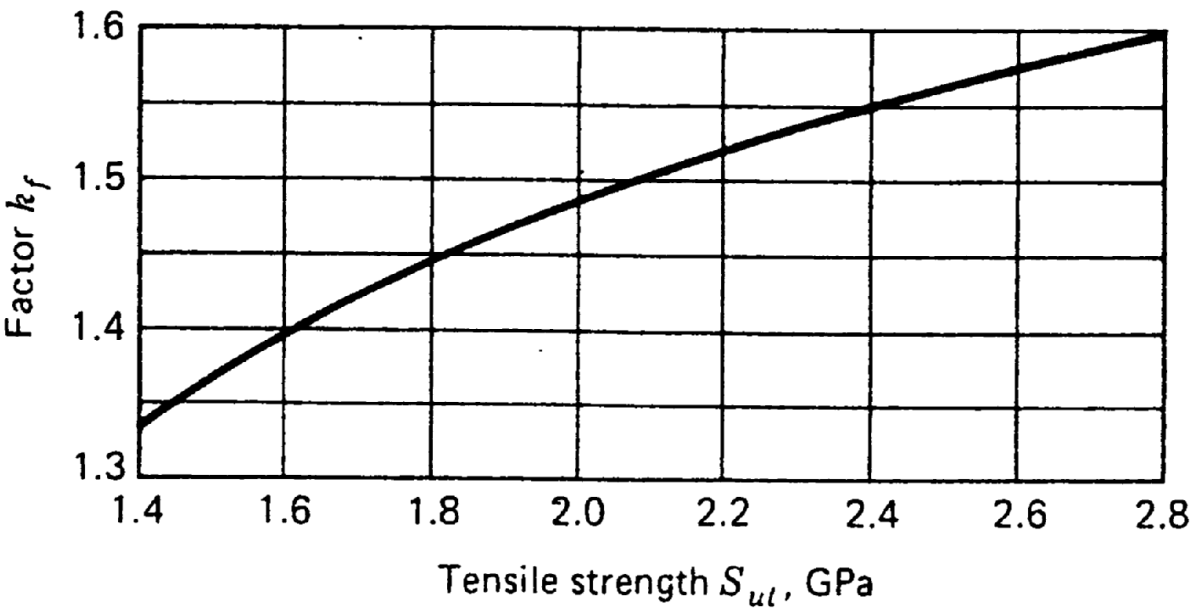


FIGURE 13-26 Miscellaneous-effects factors for one-way bending of gear teeth. Use $k_f = 1.33$ for values of S_{ut} less than 1.4 GPa.

Once the Se value is calculated from

$$S_e = k_a.k_b.k_c.k_d.k_e.k_f\, x S_e'$$

and the actual stress value σ from;

$$\sigma = \frac{Wt}{F * m * J * Kv}$$

Then global safety factor n_G can be calculated as:

$$n_G = \frac{S_e}{\sigma} \qquad n_G = n.K_o.K_m$$

Here

n is the usual safety factor (suggested to be n>1.0 even n>2)

Ko is the overload factor given in table 13-9 for different power sources & driven machinery conditions (Ko >1).

Km is the load-distribution factor given in table 13-10 for different face-width & mounting conditions (Km >1).

Table 13-9 OVERLOAD CORRECTION FACTOR K_o

	Driven machinery		
Source of power	**Uniform**	**Moderate shock**	**Heavy shock**
Uniform	1.00	1.25	1.75
Light shock	1.25	1.50	2.00
Medium shock	1.50	1.75	2.25

Table 13-10 LOAD-DISTRIBUTION FACTOR K_m FOR SPUR GEARS

	Face width, mm			
Characteristics of support	**0 to 50**	**150**	**225**	**400 up**
Accurate mountings, small bearing clearances, minimum deflection, precision gears	1.3	1.4	1.5	1.8
Less rigid mountings, less accurate gears, contact across full face	1.6	1.7	1.8	2.2
Accuracy and mounting such that less than full-face contact exists	Over 2.2			

EXAMPLE: A gear pair with 1.5mm module, 25degrees pressure angle, 15teeth pinion driving a 64teeth gear, 25mm facewidth, hobbed teeth, a=1m, b=1.25m is made of BS080M40 HR steel and experiences light shock loads on both driving & driven machineries.

For n=2.5, R=50%, average mounting conditions, and a pitch line velocity 3.8 m/s find safe power capacity of the gear set based on bending?

P=? kW

$$m = 1.5mm$$

$$T_p = 15$$

$$T_g = 64$$

$$F = 25mm$$

Φ=25degrees
Sut=490 MPa
From Table in App.

$$\mathrm{P = W_t x V} \quad \mathrm{W_t = ?} \quad \mathrm{s} = \frac{\mathrm{W_t}}{\mathrm{FmJK_v}} = \frac{\mathrm{S_e}}{\mathrm{n_G = K_o K_m n}}$$

$$\mathrm{W_t = FmJK_v} \frac{\mathrm{S_e}}{\mathrm{K_o K_m n}} \qquad \mathrm{J} \cong 0.389 \qquad \text{T13-5}$$

$$\mathrm{K_v} = \frac{50}{50 + \sqrt{200\mathrm{V}}} = 0.644$$

$$\mathrm{K_o} = 1.5 \quad \text{Tb 13-9}$$

$$\mathrm{K_m} = 1.6 \quad \text{T13-10}$$

$$k_a = 0.8, k_b = 1.0, k_c = 1.0, k_d = 1.0, k_e = 1.0, k_f = 1.33$$

$S_e' = 0.5x490 = 245MPa$

$S_e = 200MPa$

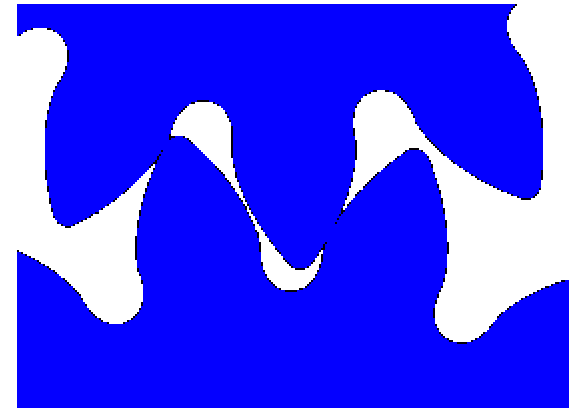
$$W_t = 25x1.5x0.389x0.644x \frac{200}{1.5x1.6x2.}$$

$W_t = 407N$

P=407x3.8=1546watt or less

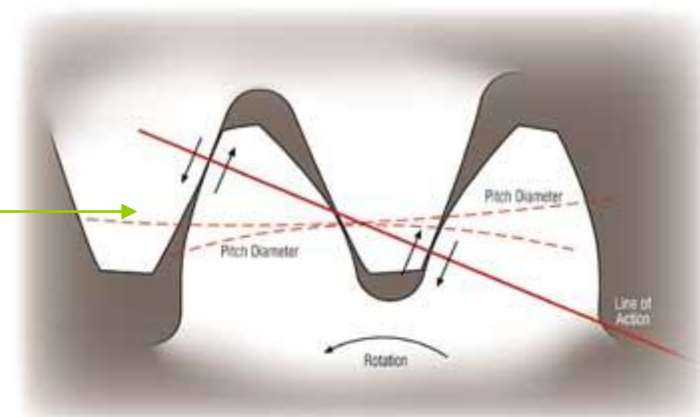
$P \cong 1.5kW$ or less

Surface durability of gear teeth



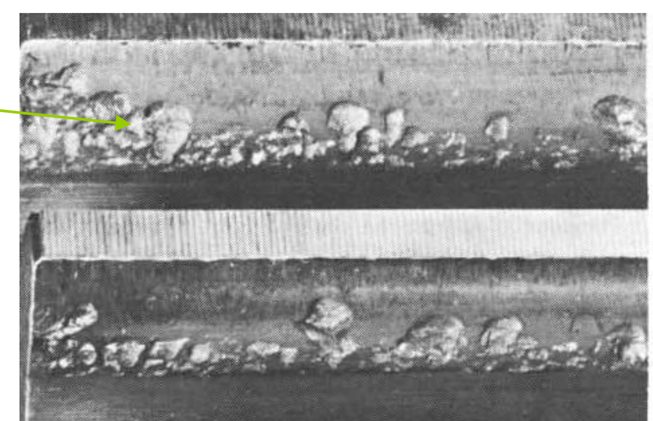
Other than the fatigue failure of tooth due to bending the teeth surfaces may also fail due to high contact stresses on the teeth.

The tooth surface will wear or pit due to high contact stresses along with the sliding action near tip & root of the tooth.



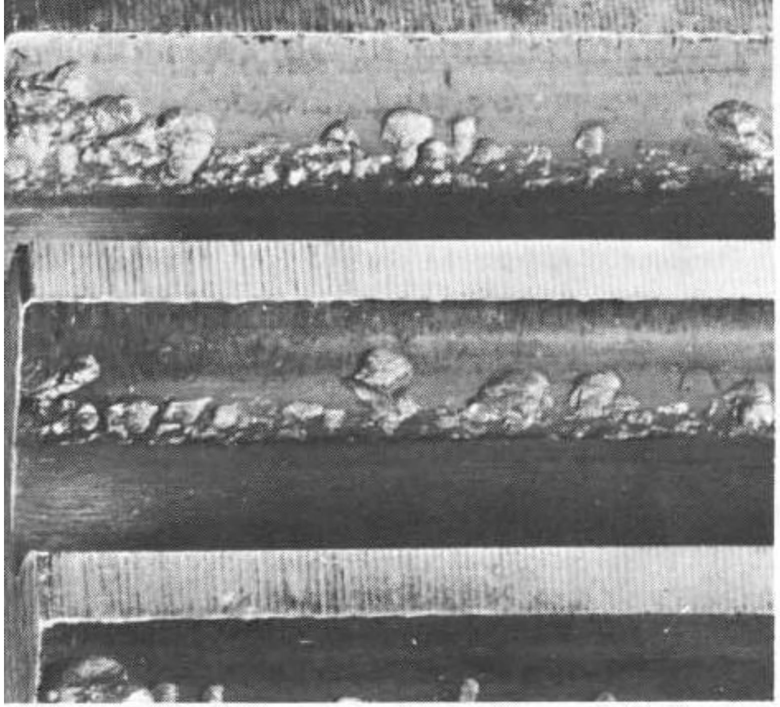
The common failure types are:

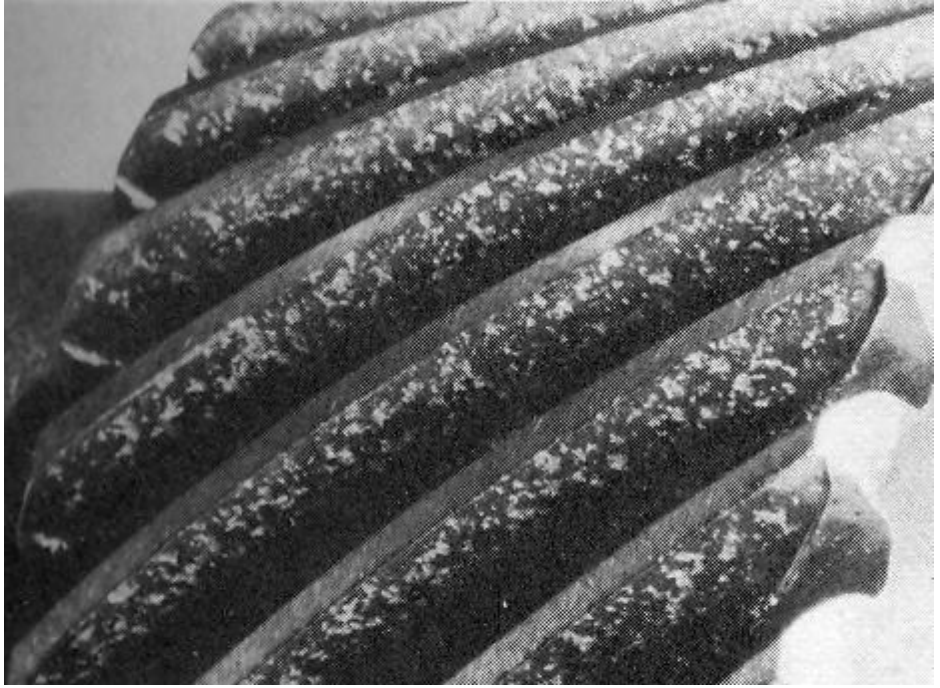
-pitting (due to many repitations of high cont. stresses)



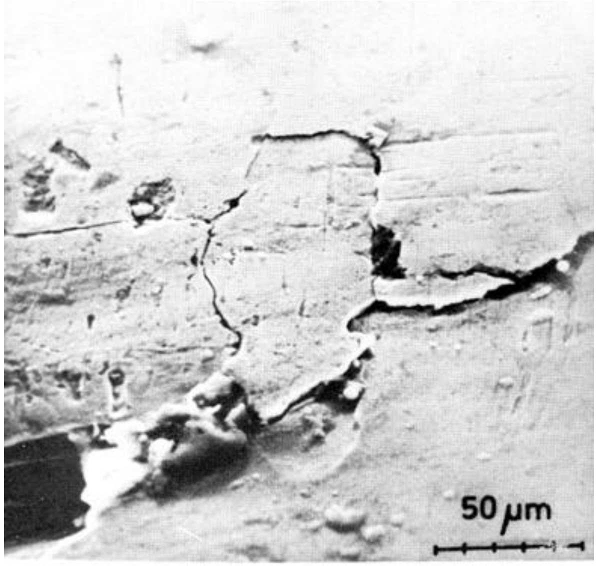
-scoring (due to lubricant failure)

-abrasion (due to presence of foreign material)





Surface failures due to contact stresses



Contours of a particle to be spalled out.

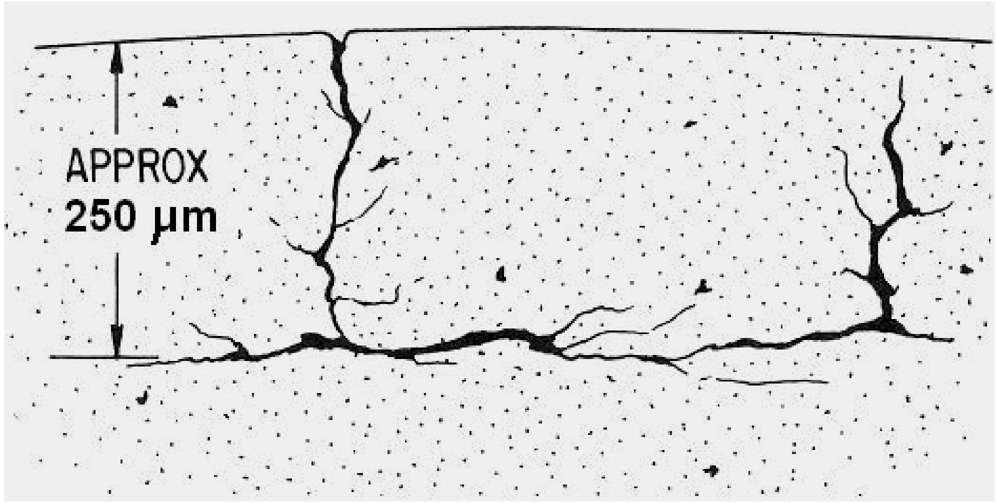
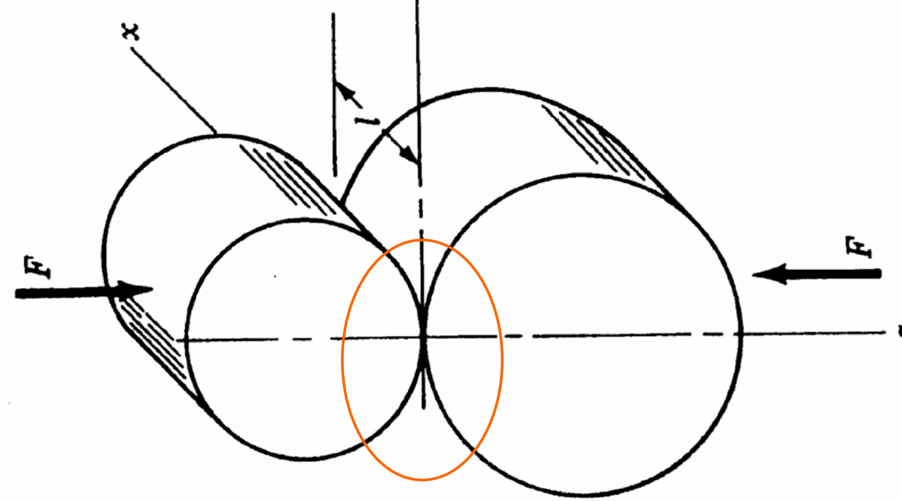
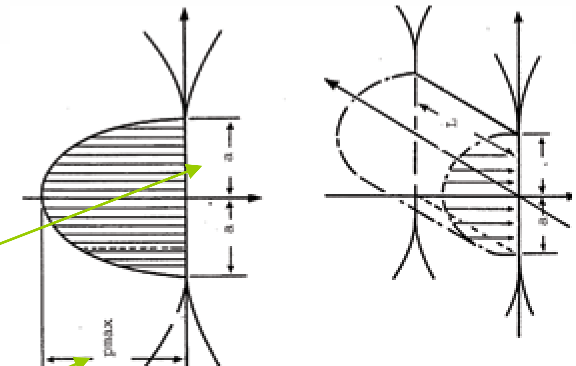


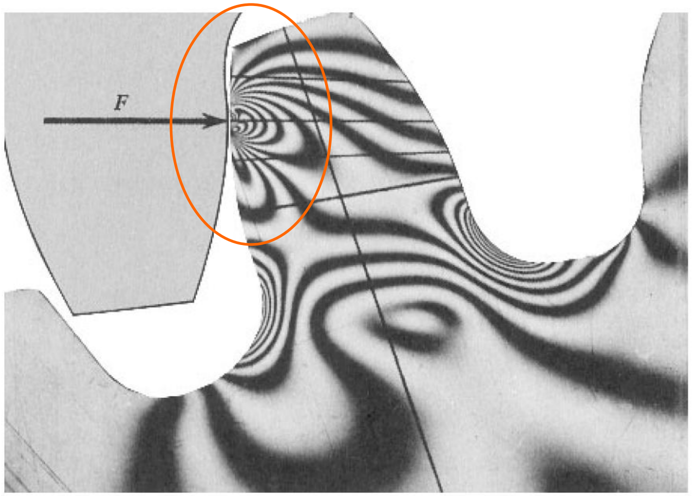
Figure illustrates a case in which the contacting elements are two cylinders of length l and diameters d1 and d2 .



The area of contact is a narrow rectangle of width 2b and length l, and the pressure distribution is elliptical. The half-width b is giyen by the equation



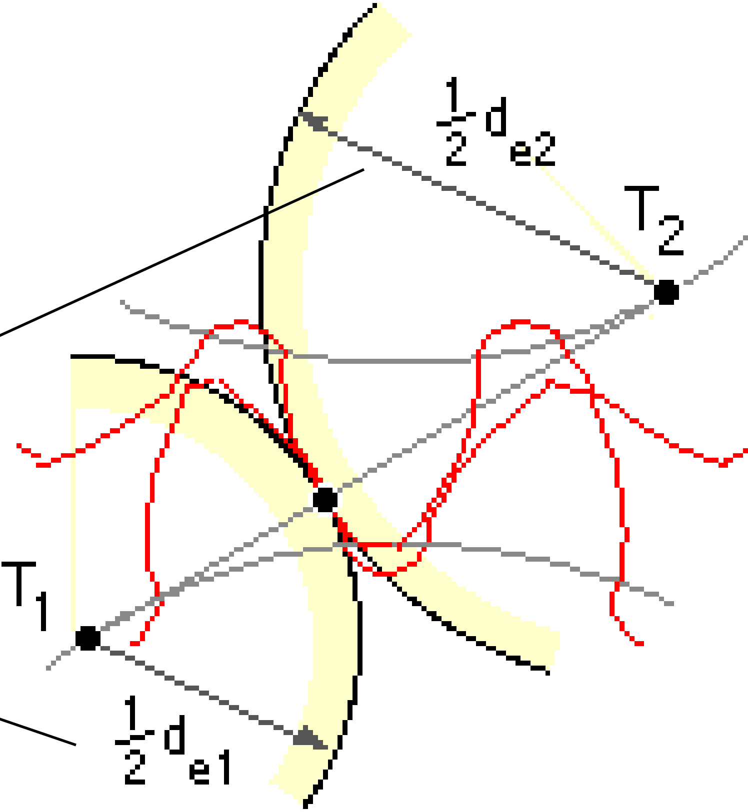
$$b = \sqrt{\frac{2F}{\pi l}\,\frac{[(1-\nu_1^2)/E_1] + [(1-\nu_2^2)/E_2]}{(1/d_1) + (1/d_2)}}$$



The maximum pressure is

$$p_{\max} = \frac{2F}{\pi b l}$$

For contacting gears at pitch point the contact stress is given as:



$$b = \sqrt{\frac{2F}{\pi l} \frac{[(1 - \nu_1^2)/E_1] + [(1 - \nu_2^2)/E_2]}{(1/d_1) + (1/d_2)}}$$

The maximum pressure is

$$p_{\max} = \frac{2F}{\pi b l}$$

$$\sigma_H = -\sqrt{\frac{W_t}{F d_P} \frac{1}{\pi\left(\frac{1 - \nu_P^2}{E_P} + \frac{1 - \nu_G^2}{E_G}\right)} \frac{1}{\frac{\cos\phi \sin\phi}{2} \frac{m_G}{m_G + 1}}}$$

$$\sigma_H = -C_p \sqrt{\frac{W_t}{C_v F d_P I}}$$

Let:

$$C_p = \sqrt{\frac{1}{\pi\left(\frac{1 - \nu_P^2}{E_P} + \frac{1 - \nu_G^2}{E_G}\right)}}$$

and

$$I = \frac{\cos\phi \sin\phi}{2} \frac{m_G}{m_G + 1}$$

Then eqn. simplifies to;

$$\sigma_H = -C_p \sqrt{\frac{W_t}{C_v F d_P I}}$$

$$C_p = \sqrt{\frac{1}{\pi\left(\frac{1-\nu_P^2}{E_P} + \frac{1-\nu_G^2}{E_G}\right)}}$$

This is the Hertzian contact stress on teeth surfaces created by the tangential load Wt.

This contact stress (σ_H) should be smaller than the surface strength (S_H) of the gear material for a safe design ($\sigma_H < S_H$)

$$S_H = \frac{C_L C_H}{C_T C_R} S_C \quad \rightarrow \quad S_C = 2.76\text{HB} - 70 \text{ MPa}$$

where

- SH corrected surface fatigue strength, or Hertzian strength
- CL life factor
- CH hardness-ratio factor; use 1.0 for spur gears
- CT temperature factor; use 1.0 for temperatures less than 120°C
- CR reliability factor

$$W_{t,p} = n_G W_t$$

$$n_G = C_o C_m n$$

In contact stress analysis, safety factor is not used as the ratio of strength to stress but rather used as a load safety factor to increase the applied load Wt

$$\sigma_H = -C_p \sqrt{\frac{W_t}{C_v F d_P I}}$$

Therefore contact stress equation is re-written as:

$W_{tp} = n_G * W_t$ and $\sigma_H = S_H$

$$S_H = C_p \sqrt{\frac{W_{t,p}}{C_v F d_P I}}$$

Table 13-11 VALUES OF THE ELASTIC COEFFICIENT C_p FOR SPUR AND HELICAL GEARS WITH NONLOCALIZED CONTACT AND FOR $\nu = 0.30$
The units of C_p are $(\text{MPa})^{1/2}$.

		Gear					
Pinion	**Modulus of elasticity *E*, GPa**	**Steel**	**Malleable iron**	**Nodular iron**	**Cast iron**	**Aluminum bronze**	**Tin bronze**
Steel	200	191	181	179	174	162	158
Mall. iron	170	181	174	172	168	158	154
Nod. iron	170	179	172	170	166	156	152
Cast iron	150	174	168	166	163	154	149
Al. bronze	120	162	158	156	154	145	141
Tin bronze	110	158	154	152	149	141	137

In case of contact stresses,

$$Cv = Kv = \frac{a}{a+V}$$

$$Cv = Kv = \frac{b}{b+V}$$

$$Cv = Kv = \sqrt{\frac{c}{c+\sqrt{200V}}}$$

Cv = Kv (as in the case of bending stress)

Co=Ko and Cm=Km (Tables 13-9 &10)

I is called the geometry factor and given as:

$$I = \frac{\cos\phi \sin\phi}{2} \frac{m_G}{m_G + 1}$$

where m_G is the gear ratio (Tg/Tp)

Factors C_L and C_R are given in Table 13-12

Table 13-12 LIFE AND RELIABILITY MODIFICATION FACTORS

Cycles of life	Life factor C_L	Reliability R	Reliability factor C_R
10^4	1.5	Up to 0.99	0.80
10^5	1.3	0.99 to 0.999	1.00
10^6	1.1	0.999 up	1.25 up
10^8 up	1.0		

Example:

A 14T precision made pinion is to drive a 21T gear.

A module of 3mm is selected with a 58 mm face width based upon a 25degrees pressure angle and dedendum of 1.25m.

20kW is to be trasmitted at pinion speed of 1150rpm, under steady load conditions.

The material selected is forged BS080M40 steel heat treated to a hardness of 235BHN.

Determine the factor of safety

i) guarding against a bending fatigue failure; for 99% reliability with better-than –average mountings & cutting accuracy.

ii) guarding against contact fatigue failure

Given

$$T_P = 14$$

$$T_g = 21$$

$$m = 3mm$$

$$F = 58mm$$

$$\theta = 25^o$$

$$b = 1.25m$$

$$n_p = 1150rpm$$

$$P = 20kW$$

$$R = 99\%$$

$$n = \frac{n_G}{K_o K_m} \rightarrow n_G = \frac{S_e}{\sigma}$$

$$S_e = k_a.k_b......k_f.S_e^{'}$$

$$k_a = 0.70 \quad \text{fig 13-25}$$

$$\mathrm{k_b} = 0.956 \quad \text{T13-7}$$

$$\mathrm{k_c} = 0.814 \quad \text{T13-8}$$

$$\mathrm{k_d} = 1.0$$

$$k_e = 1.0$$

$$k_f = 1.33$$

$$S_e^{'} = 0.5S_{ut} = 387.5MPa$$

$$S_e = 280MPa$$

- Material- (appendix)
- Steady load condition &
- better than average mounting
- BHN=235

$$S_{ut} = 775MPa$$

$$\rightarrow K_o = 1.0 \quad \text{T13-9}$$

$$\rightarrow K_m = 1.4 \quad \text{T13-10}$$

Required

n=? Se/σ?

$$\sigma = \frac{Wt}{F*m*J*Kv}$$

$$Wt = \frac{P}{V} = \frac{60*P}{\pi dn}$$

$$Wt = \frac{60*20000}{\pi(3*14*10^{-3})*1150}$$

$$Wt = 7908N$$

$J \cong 0.362$(table 13-5 interpolation l

For $T_P = 14; T_g = 21$

	1	17	25	35
13				
14 →		X		
15				
16				

$$V = \frac{\pi dn}{60} = 2.529 m/s$$

$$K_v = \sqrt{\frac{78}{78+\sqrt{200}}}$$ precision made

$$K_v = 0.881$$

$$\sigma = \frac{7908}{58*3*0.362*0.881}$$

$$\sigma = 142.5 MPa$$

$$n_G = \frac{S_e}{\sigma} = \frac{280}{142.5} = 1.96$$

$$n = \frac{n_G}{K_o K_m} = \frac{1.96}{1*1.4} = 1.40 \geq 1.0$$

ii)n=? Based on contact stress

$$Wtp = Wt * n_G = Wt * (nC_oC_m) \rightarrow n = \frac{Wtp}{Wt * C_oC_m} = ?$$

$C_0 = K_0 = 1.0$

$C_m = K_m = 1.4$

$$Wt = \frac{P}{V} = \frac{60 * P}{\pi dn}$$

$$Wt = \frac{60 * 20000}{\pi(3*14*10^{-3})*1150}$$

$$Wt = 7908N$$

$$W_t = ? \rightarrow \frac{60P}{\pi dn} = \frac{60x20}{\pi(3+14)1150.10^3} = 7908kN$$

$$S_H = C_P \frac{W_{tp}}{C_V F d_p I} = ?$$

$$W_{tp} = \left(\frac{S_H}{C_P}\right)^2 C_V F d_p I$$

$$S_H = \frac{C_L C_H}{C_T C_R} S_C = \_\_\_\_\_\_\_\_\_\_ (2.76HB - 70)$$

$C_V = K_V = 0.881$ already calculated

F=58mm

$d_p = 3x14 = 42mm$

$$I = \frac{\cos\phi \sin\phi}{2}\frac{\mu_G}{\mu_G + 1}; \phi = 25^o \quad \mu_G = \frac{T_g}{T_P} = \frac{21}{14} = 1.5$$

$$I = \_\_\_$$

$$W_{tp} = \left(\frac{\_\_\_}{191}\right) 0.881x58x42x0.\_\_$$

$$W_{tp} = \_\_\_\_\_ N = \_\_\_\_\_\_\_ kN$$

$$n = \frac{\_\_\_}{7.908x1.0x1.4} =$$

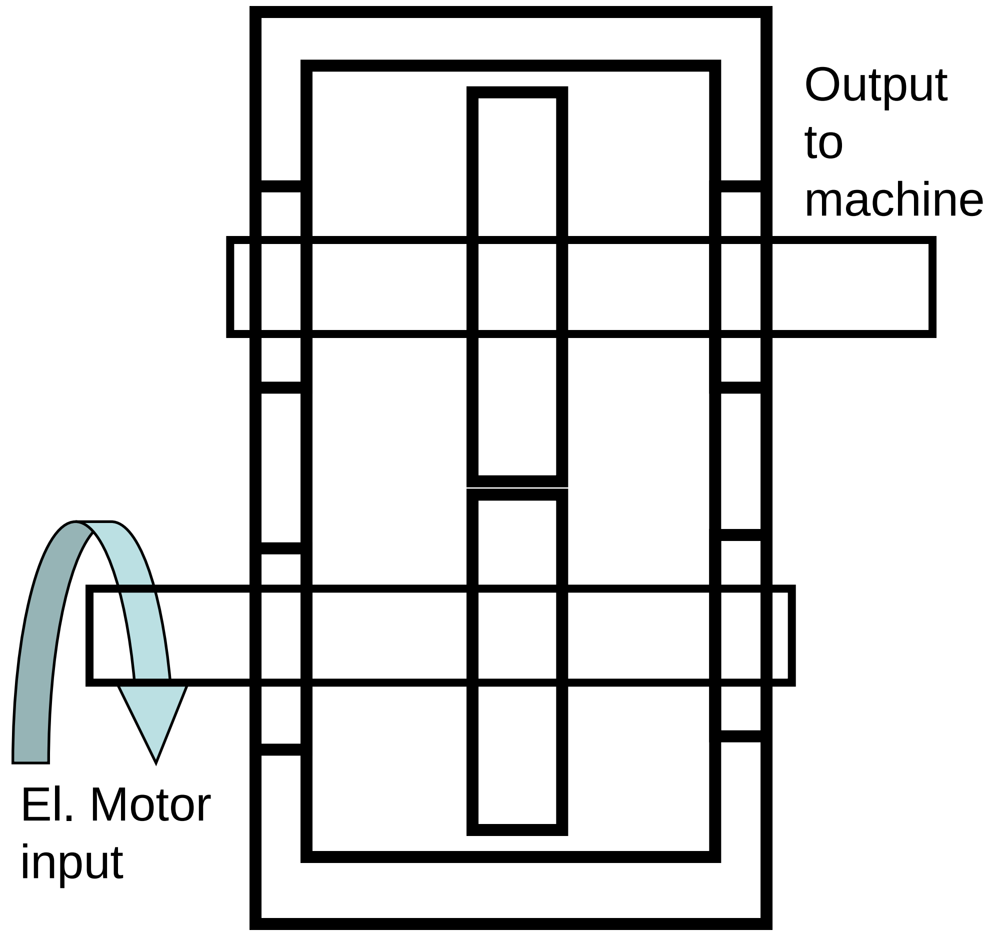
Example:

In the reduction unit shown, the pinion shaft is driven by a 7.5 kW motor by means of a coupling. The motor speed is 1200 rpm & the output shaft is to rotate at 400 rpm.

a) Assuming that the gear & the pinion are made of UN5G10500 cold drawn steel (with Sut=689MPa) design the gear & the pinion based on the bending fatigue failure with 90% relaibility.

b) Assuming that the gear & the pinion are made of UNSG10500 at HB=310, design both gear & the pinion based on the surface durability.

For both cases the load is steady & continious and the factor of safety is 1.5. Also state any assumptions you use.



Solution:

Assume that: 1) $\theta = 20^o$

2) full depth gear a=1.0m

3) $T_{\min} = 18 teeth$

Designing a gear means :

Determine: a) $T_1 = ? \quad T_2 = ? \rightarrow for \ T_1 = T_{\min} = 18 \rightarrow T_2 = \frac{1200}{400} x T_1 \quad T_2 = 54$

b) $m = ? \rightarrow d_1 = ? \quad d_2 = ?$

c) $F = ?$

a)Based on bending fatigue failure:

We have 2 criteria : 1) $n_G = \frac{S_e}{\sigma}$

$n_G = ?$

$S_e = ?$

$\sigma = ?$

2) $3p < F < 5p$

Since most parameters are dependent as more than 1 variable (m,F,T) we have to use trial & error method based on 2 criteria

$$n_G = \frac{S_e}{\sigma}$$

$$3p < F < 5p$$

Since both pinion & gear are made of the same material with same S_{ut} design on be based on pinion since it rotates more.

$d = Txm$, $V = \frac{\pi dn}{60}$, $K_V = \frac{50}{50 + \sqrt{200V}}$ For hobbed teeth

$T = \frac{S_e}{n_G}$, $F = \frac{W_t}{TJmK_v}$

$3p = 3x\pi xm$

$5p = 5x\pi xm$

$W_t = \frac{P}{V}$

$n = 1.5$

$K_o = 1.0$ uniform loading

$K_m = 1.3$ assuming $F \le 50$mm & accurate mounting

$J \cong 0.345$ for $18T \to 54T$ Table 13-4

$S_e = k_a.k_b....k_f\,(0.5x690)$

$S_e = k_b x296.34MPa$ (k_b dependent)

$T = \frac{S_e}{n_G} = \frac{296.34kb}{1.95} = 151.97kb$ MPa

m	$d_p mm$	k_b	$V m/\sec$	K_V	W_t, N	$T\ MPa$	3p	F	5p	notes
3	54	0.956	3.392	0.657	2211	145.28	28.27	22.38	47.12	Not suitable
4	72	0.93	4.523	0.624	1658.2	141.33	37.7	26.2	62.83	Gets worse To the left
2	36	1.0	2.262	0.701	3315	151.96	18.85	45.1	31.4	To the right & not suitable
2.5	45	0.974	2.827	0.677	2653	148.01	23.56	30.7	39.26	SUITABLE

<u>Thus , based on bending stress</u>

$$m = 2.5mm$$

$$T_P = 18 \quad \mathrm{d_p} = 45$$

$$T_g = 54 \quad \mathrm{d_g} = 135$$

$$F = 31 - 35mm$$

Assume F<50 mm

$$S_H = -C_P\sqrt{\frac{W_{tp}}{C_V F d_p I}} \quad \text{or} \quad F = \frac{W_{tp}}{\left(\frac{S_H}{C_P}\right)^2 C_V d_p I}$$

$C_m = 1.3$

$$S_H = \frac{C_L C_H}{C_T C_R} x\left(2.76x310 - 70\right) = 982MPa$$

$C_O = 1.0$

$$C_P = 191 \text{ steel on steel}$$

$n = 1.5$

$$\left(\frac{S_H}{C_P}\right)^2 = 26.433$$

$n_G = 1.95$

$$I = \frac{C_u \phi S_m \phi}{2}\frac{3.0}{3.0+1} = 0.107$$

b) Based on surface durability we have 2 criteria:

1)

$$S_H = -C_P\sqrt{\frac{W_{tp}}{C_V F d_p I}} \quad \text{or} \quad F = \frac{W_{tp}}{\left(\frac{S_H}{C_P}\right)^2 C_V d_p I}$$

2) 3p<F<5p

T_1	T_2	m	d_p	V	C_V	W_t, N	W_{tp}, N	3p	F	5p	notes
18	27	2	36	2.262	0.701	3315	6464.25	18.85	100.5	31.4	NO
		4	72	4.523	0.624	1658	3233	37.69	25.44	62.83	NO
		3	54	3.392	0.657	2211	4311.5	28.27	42.9	47.12	OK
		3.5	63	3.958	0.640	1895	3695	33.0	32.3	55	OK

Results

$$m = 3.0$$

$$T_1 = 18 \rightarrow d_1 = 54mm$$

$$T_2 = 54 \rightarrow d_2 = 162mm$$

$$F = 43 - 47mm$$

based on comtact stress

Example

A set of spur gears (pinion cut from hot rolled BS 08040M steel with HB 180 and gear cut from BS 220 cast-iron) is to be designed to transmit 1.25 kW at a pinion speed of 400 rpm and a speed reduction of 1.5:1.

Use a safety factor of 4 and determine suitable values for:

module:

Pitch diameters:

Tooth numbers:

Face width of the pinion & gear.

Use 20degrees full-depth teeth with b=1.25m & make necessary assumptions if required.

Solution: Two creteria:

1) bending stress fatigue(for pinion&gear)

2) Contact stress fatigue (for pinion & gear)

4 total analysis
Or designs

Bending stress:

$$n_G = \frac{S_e}{T} = \frac{k_a.k_b...S_e^{'}}{\frac{W_t}{FK_V Tm}} = k_a.k_b....S_{ut}/2$$

$$S_{ut_{pin}} = 550MPa$$

$$S_{ut_{gear}} = 220MPa$$

Gear with $S_{ut} = 220MPa \left(< S_{ut_{pin}} = 550MPa \right)$ is more ciritical than the pinion. Thus base the bending fatigue design on gear.

Contact stress:

$$n_G = \frac{W_{tp}}{W_t} \text{ or } \frac{1}{\mathrm{n_G}} = \frac{T_H}{S_H} = \frac{C_P\sqrt{\frac{W_t}{C_V F d_p I}}}{\frac{C_L C_H}{C_T C_R} S_e = \frac{C_L C_H}{C_T C_R}(2.76hb - 70)MPa}$$

$$HB_{pin} = 180$$

$$HB_{gear} = 196$$

$$n_G = \frac{\frac{CC}{CC}(2.76HB - 70)}{C_P\sqrt{\frac{W_t}{C_V F d_p I}}}$$

Since HBpin is less than HBgear the contact stress design has to be based on pinion. Also Pinion rotates more.

Bending stress fatigue design of gear

$$T \le \frac{S_e}{n_G} \quad \& \quad 3p<F<5p$$

$$T = \frac{W_t}{K_v F J m} \rightarrow \quad a) W_t = \frac{\text{Power}}{\text{linear velocity}} = \frac{1250 \text{ watt}}{\frac{\pi d_2 n_2}{60}} = \frac{60x1250}{\pi x (T_2 x m) x (400/1.5)}$$

$$S_e = k_a . k_b .... S_e^{'}$$

$$= \frac{60x1250}{\pi (1.5 T_1 x m) x (400/1.5)}$$

$$W_t = \frac{60x1250}{\pi x 400 x T_1 x m}$$

$$\text{let } T_1 = 18, T_2 = 27 \quad W_t = \frac{3.3157}{m} Nt (\text{m in meters})$$

b) $K_v = \dfrac{50}{50+\sqrt{200V}}$ assuming that pinion & gear are hobbed.

(V in meter/sec) $V = \dfrac{\pi d_n}{60} = \dfrac{\pi x T_1 m x 400}{60} = 376.99m$ m/sec m in meters

c) $J = 0.3686$

$\theta = 20^o$ b=1.25

d)m is already unknown

e)F is also unknown but limited to 3p<F<5p.

$$F = \frac{W_t}{TK_v Jm} \quad T = \frac{S_e}{n_G} = \frac{k_a.k_b...S_e^{'}}{n.K_o.K_m}$$

$$S_e' = 0.5 S_{ut} \text{ if } \mathrm{S_{ut}} < 1400 MPa$$

$$S_e' = 0.5x220 = 110 MPa$$

$$k_a = 0.80 \text{ from the figure}$$

$$\mathrm{k_b} = f(m)$$

$$k_c = 0.897 \text{ for 90\% rel. (assumed)}$$

$$\mathrm{k_d} = 1.0 \text{ (for T<350}^{\mathrm{o}} C)$$

$$k_e = 1.0$$

$$k_f = 1.33 \text{ from figure}$$

$$\mathrm{S_e} = 0.80 x k_b x 0.897 x 1.0 x 1.0 x 1.33 x 110$$

$$S_e = 104.985 k_b \text{ MPa}$$

$$\mathrm{n_G} = \mathrm{K_o} K_m n; \ \mathrm{K_o} = 1.5 \text{ light mod. shock}$$

$$\mathrm{K_m} = 1.7 \text{ assumed 50<F<150mm.}$$

$$\mathrm{n_G} = 1.5 x 1.7 x 4$$

$$n_G = 10.2$$

$$T = \frac{104.985 k_b}{10.2} MPa.$$

Since many parameters are dependent on module(m) we use tabulation method of

m mm	k_b	$V m/\sec$	K_V	W_t Nt	S_e MPa	T MPa	F mm	3p	5p	notes
2	1	0.754	0.803	1657.85	104.985	10.292	274	18.85	31.41	NOT GOOD
4	0.93	1.508	0.742	829	97.636	9.572	79.16	37.7	62.83	NOT GOOD
5	0.91	1.885	0.720 0.614	663.14	95.536	9.366	53.35 62.6	47.12	78.53	OKEY
4.5	0.92	1.696	0.73	737	96.586	9.4692	64.27	42.4	70.7	OKEY

Thus

$$T_1 = 18 \qquad d_1 = T_1 m = 90mm$$

$$T_2 = 27 \qquad d_2 = T_2 m = 135mm$$

$$m = 5.0$$

$$F \cong 54mm \rightarrow K_m = 1.7$$ İs correct.

will be satisfactory from the bending fatigue strength point of view considering the gear material.

<u>Checking the pinion</u> $(S_{ut} = 550MPa)$

$$S_e = 0.76x0.91x0.897x1.0x1.0x1.33x\frac{550}{2} = 226.9MPa$$

$$T = \frac{W_t}{K_v FJm} = \frac{663.15Nt}{0.72x54x(0.334)x5.0} = 10.21MPa$$

$$n_G = \frac{226.9}{10.4} = 22.22$$

$$n = \frac{n_G}{K_o.K_m} = \frac{22.22}{1.5x1.7} = 8.70$$

So pinion İs safe!

$$n = 8.70 > 4$$

Contact stress design of pinion

Rather than re-designing the pinion,check the present design based on bending stress for the contact stress

$$n_G = \frac{W_{tp}}{W_t} = n.C_o.C$$

$$S_H = T_H = C_p \sqrt{\frac{W_{tp}}{C_v F d_p I}}$$

S_c =2.76HB-70

$C_p = 174$ steel-CI

S_c =2.76x180-70=426.8MPa

$C_v = K_v = 0.72$

$$S_H = \frac{1.0x1.0}{1.0x0.8}x426.8 = 533.5MPa$$

$F = 54mm$

$d_p = T_1 xm = 90mm$

$$I = \frac{\cos\phi.\sin\phi}{2}.\frac{m_G}{m_G+1} = \frac{\cos 20.\sin 20}{2}\frac{1.5}{1.5+1}$$

$I = 0.0964$

$$W_{tp} = \left(\frac{S_H}{C_p}\right)^2 x C_v F d_p I = \left(\frac{533.5}{174}\right)^2 x0.72x54x90x0.0964$$

$$W_{tp} = 3171 Nt$$

$$n_G = \frac{3171 Nt}{663.15 Nt} = 4.7804 = n.C_o C_m \qquad \text{n} = \frac{4.7804}{1.5\text{x}1.7} = 1.875 < 4$$

So pinion is not safe in terms of contact stress for the data of

$m = 5$, F=54mm & $T_1 = 18, T_2 = 27$

Now re-design the pinion based on contact stress

$$S_H = C_P \sqrt{\frac{W_{tp}}{C_v F d_p I}} \quad \& \quad 3p \leq F \leq 5p$$

$$S_H = 533.5 MPa$$

$$C_p = 174$$

$$I = 0.0964$$

$$d_p = m x T_1 = 18m$$

$$W_{tp} = n.C_o x C_m x W_t$$

$$C_o = 1.5$$

$$C_m = 1.7 \quad 50 < F < 150$$

$$W_t = \frac{60xp}{\pi d n} = \frac{60xp}{\pi n T m}$$

$$W_t = \frac{3315.7}{m} Nt \quad W_{tp} = \frac{33820.14}{m}$$

$$V = 0.377m$$

$$V = \frac{\pi dn}{60} = \frac{\pi xnxTxm}{60}$$

$$C_V = \frac{50}{50 + \sqrt{200V}} \rightarrow C_V = \frac{50}{50 + \sqrt{200x0.377m}}$$

$$F = \frac{W_{tp}}{\left(\frac{S_H}{C_p}\right)^2 C_V d_p I}$$

Use tabulation method

M	Vm/sec	C_v	$W_{tp} Nt$	$d_p mm$	$F mm$	3p	5p	notes
5	1.885	0.72	6764	90	115	47	78	Not Good
8	3.016	0.67	4227.5	144	48.35	75.4	125.6	Not Good
6	2.262	0.7015	5637	108	82	56.5	94.25	OKEY
7	2.359	0.697	4831.5	126	60.7	66	110	Not Good
3	1.131	0.768	11273.4	54	300	28.2	47.12	Not Good

$$m = 6mm$$

$$T_1 = 18 \quad d_1 = 108mm$$

$$T_2 = 27 \quad d_2 = 162mm$$

$$F = 83mm$$

Satisfies both
bending stress fatigue
and contact stress cond's.

Now check <u>gear</u> for bending srength fatigue

$$n_G = \frac{S_e}{T} = n.K_o.K_m$$

$$S_e = 0.80x0.894x0.897x1.33x\frac{220}{2} = 93.85MPa$$

$$T = \frac{W_t}{FK_V Jm} = \frac{552.6}{82x0.7015x0.3686x6}$$

$$T = 4.3437MPa$$

$$n_G = \frac{93.85}{4.3437} = 21.6 = n.1.5x1.7$$

$$n = 8.47 > 4 \quad \text{OKEY!}$$

Also check surface durability

$$W_{tp} = \left(\frac{S_H}{C_p}\right)^2 C_V F d_p I$$

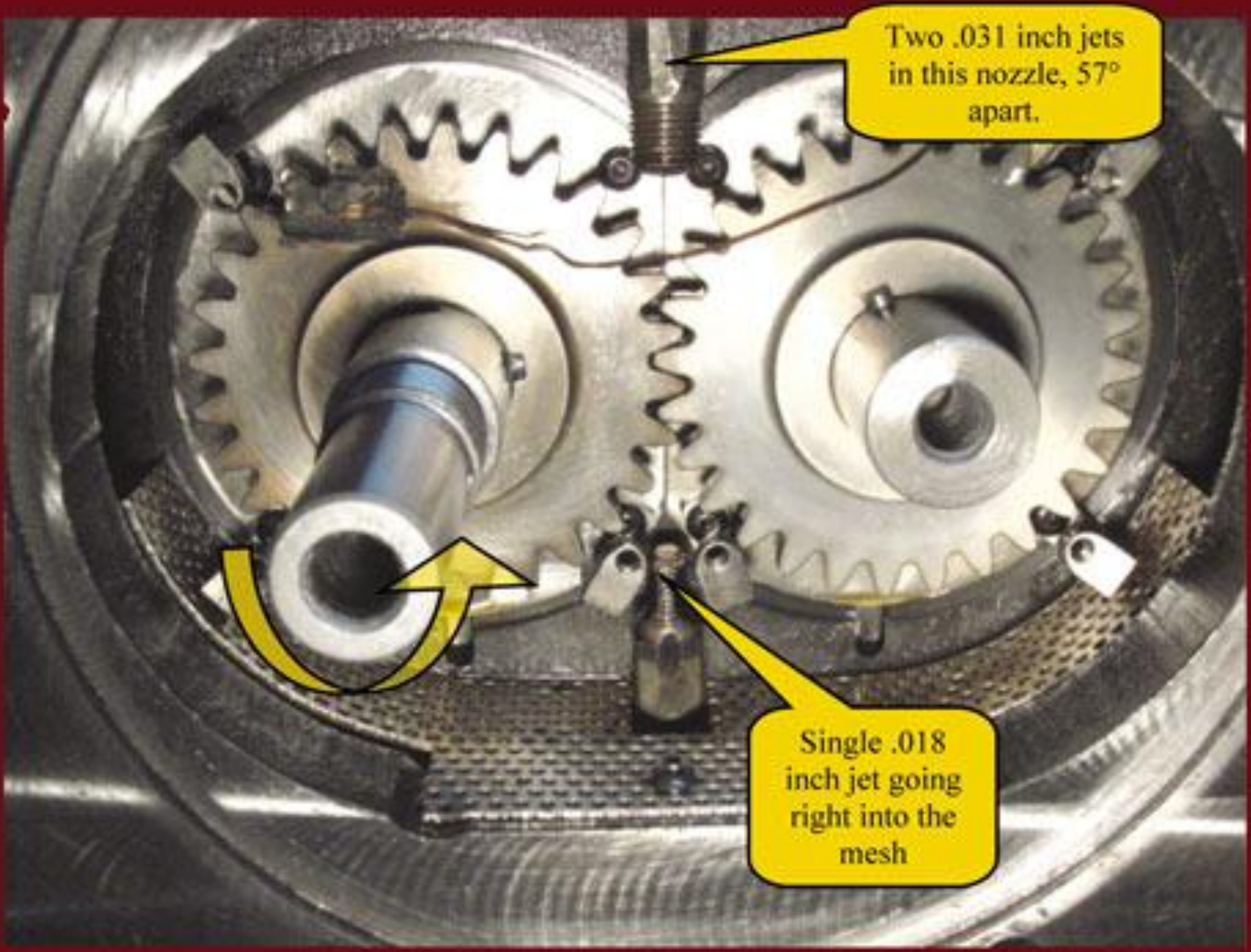
$$S_{H_g} = \frac{1.0x1.0}{1.0x0.8}\left(2.76x196 - 70\right) = 588.7MPa$$

$$W_{tp} = \left(\frac{588.7}{174}\right)^2 x0.7015x82x108x0.0964$$

$$W_{tp} = 6855.4N$$

$$n_G = \frac{W_{tp}}{W_t} = \frac{6855.4}{552.6} = 12.4$$

$$n = \frac{n_G}{C_o C_m} = \frac{12.4}{1.5x1.7} = 4.86 > 4 \quad \text{OKEY!}$$


Two .031 inch jets in this nozzle, 57° apart.
Single .018 inch jet going right into the mesh