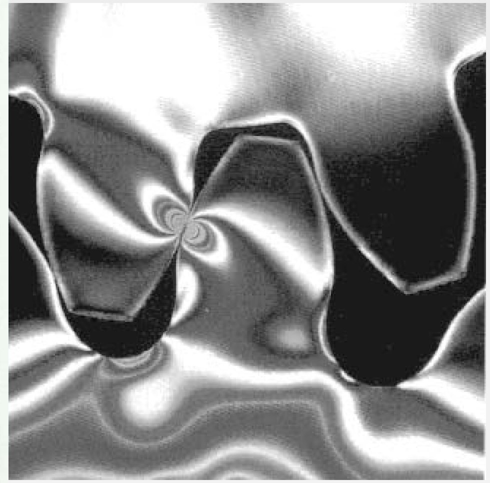
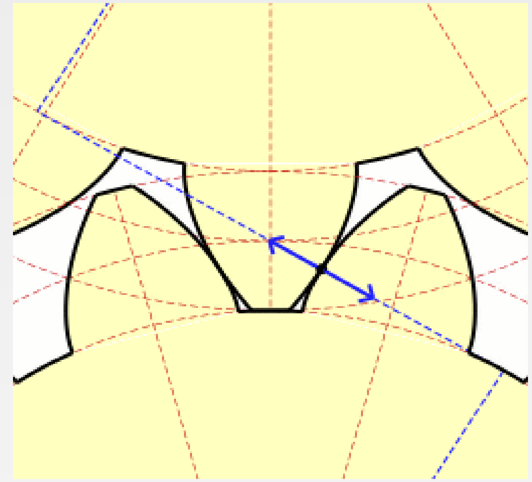
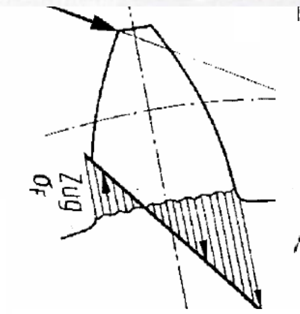
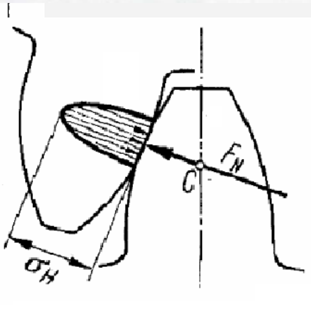
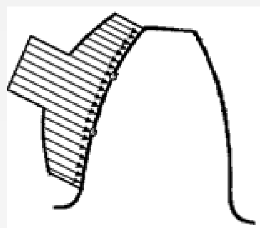




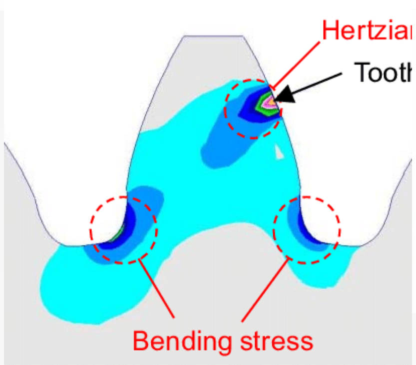


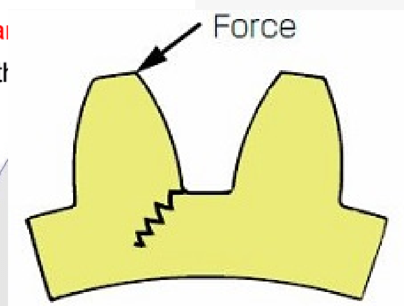




Bölüm 2 (section 2)
Düz dişli çark
Mukavemet Kontrolü
Spur Gears Stress

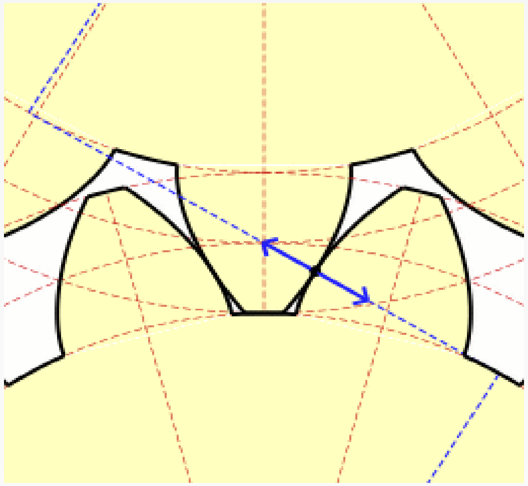
Prof. Dr Hikmet Kocabaş
İstanbul Teknik Üniversitesi İ.T.Ü. Makina Fakültesi
Istanbul Technical University ITU Faculty of Mechanical Engg.





Bölüm 2 Dişli çark mukavemet kontrolü

- Düz Dişli Çark Kinematiği, Profil Kaydırma
- **Dişli Çark Mukavemet Kontrolü**
- Helisel, Konik, Spiral Dişli Çark Mek.
- Sonsuz Vida Mekanizmaları
- Düz-, V-, Dişli-Kayış-Kasnak Mek.
- Zincir-Dişli Mekanizmaları



Kaynaklar

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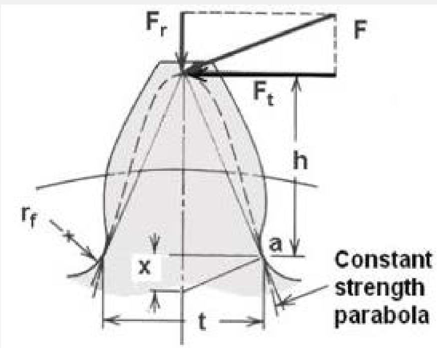
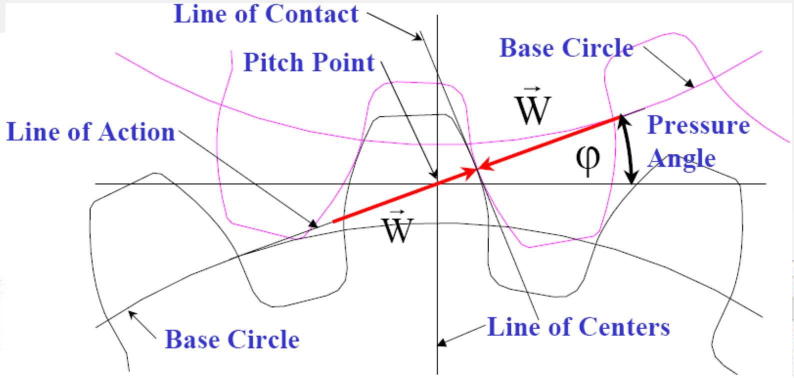
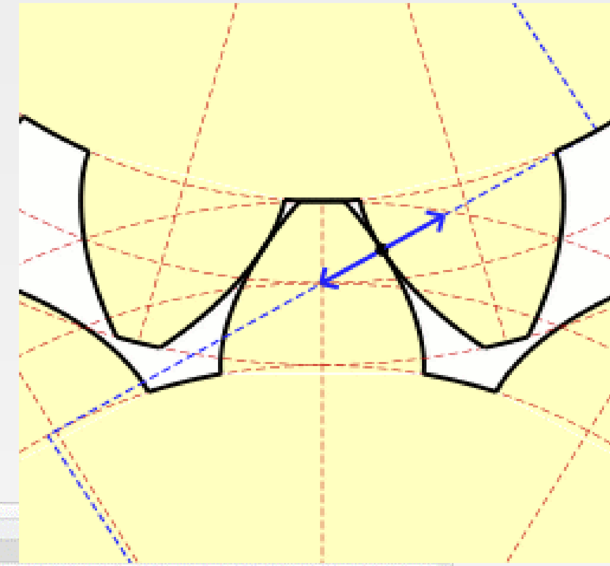
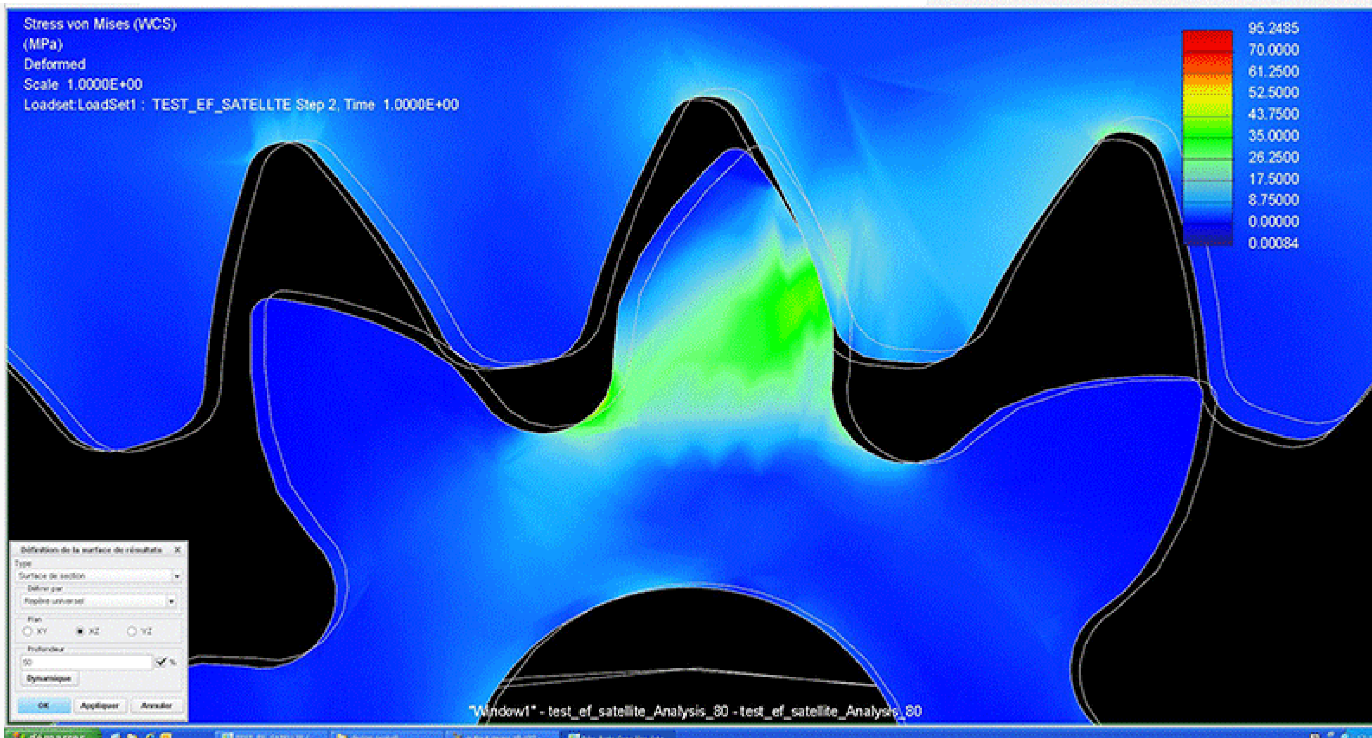
Dişlerde kontrol noktaları









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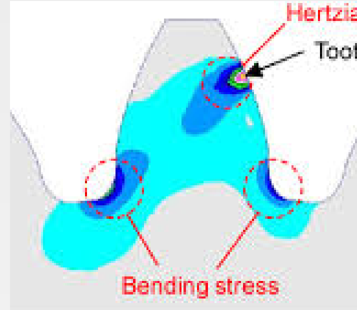
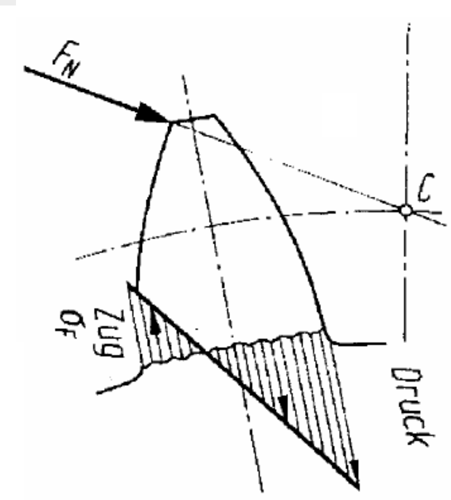
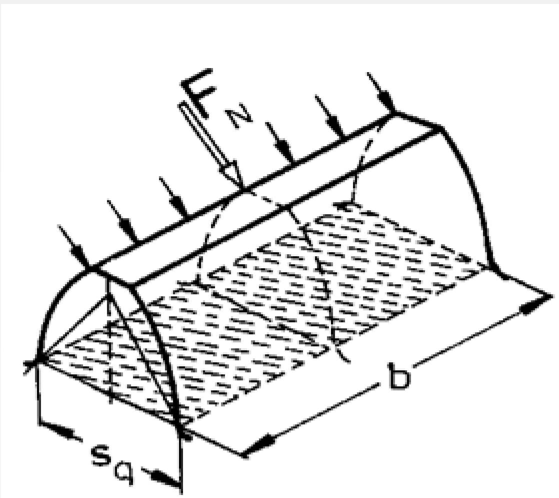
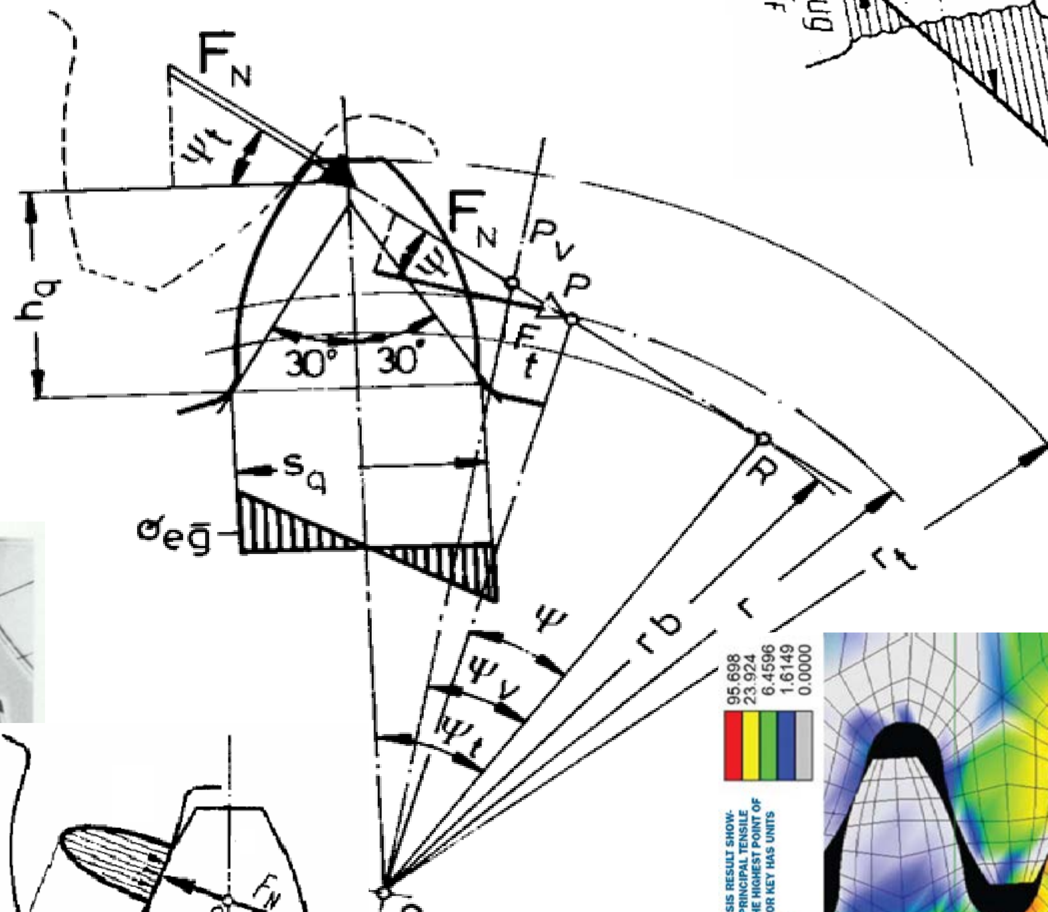
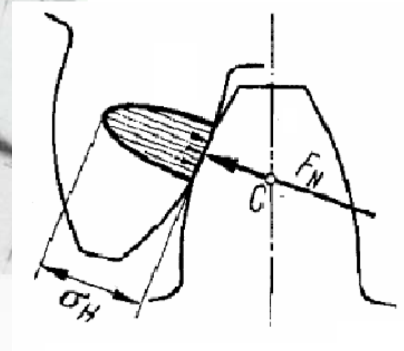










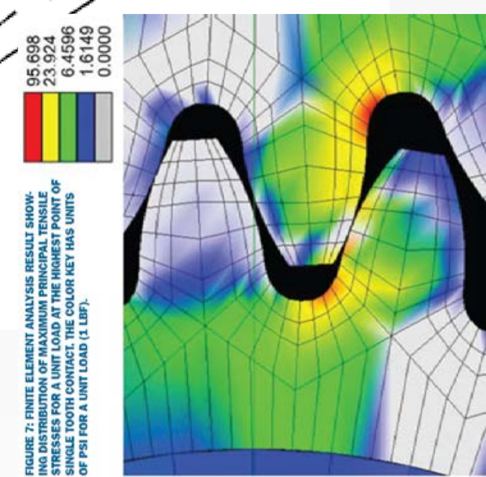


FIGURE 7: FINITE ELEMENT ANALYSIS RESULT SHOWING DISTRIBUTION OF MAXIMUM PRINCIPAL TENSILE STRESSES FOR A UNIT LOAD AT THE HIGHEST POINT OF SINGLE TOOTH CONTACT. THE COLOR KEY HAS UNITS OF PSI FOR A UNIT LOAD (1 LBF).

Diş dibi eğilmesi

DIN 3990'da dişdibindeki eğilme gerilmesi

$$\sigma = \frac{F_t}{bm} Y_F Y_S Y_\beta$$

Burada:

b: dişli genişliği

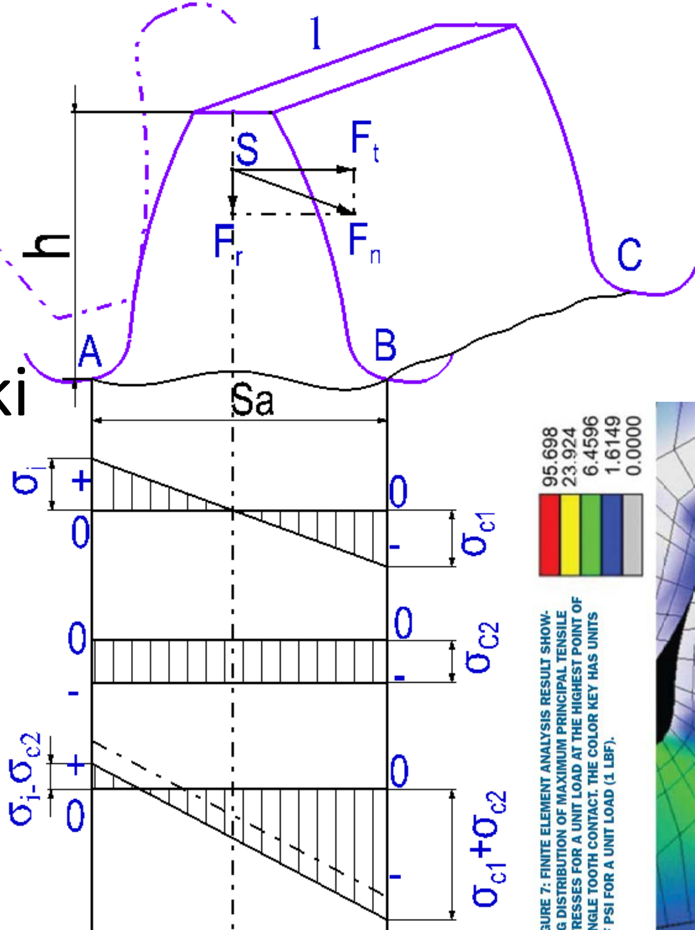
m: modül

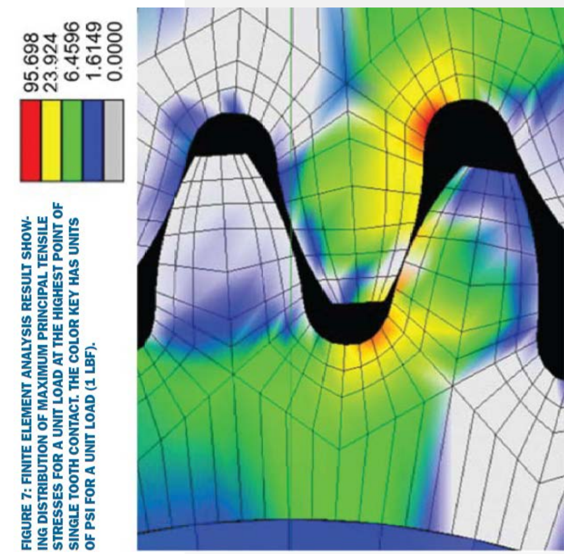
DIN-Faktörleri:

Y_F : form faktörü

Y_S : dişdibi çentik faktörü

Y_β : dişdibi helis faktörü





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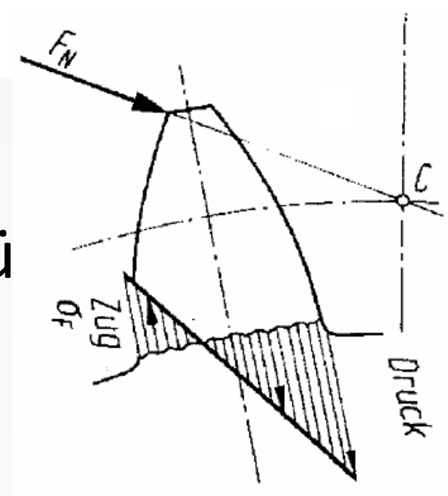
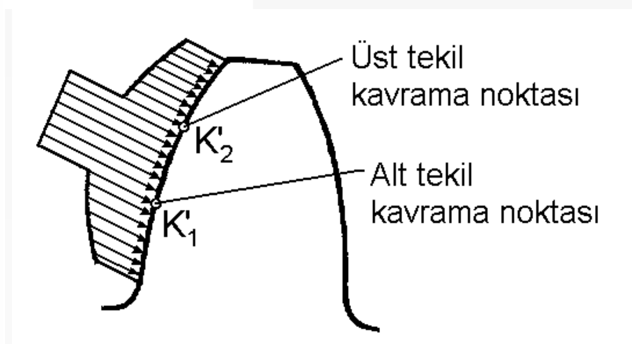
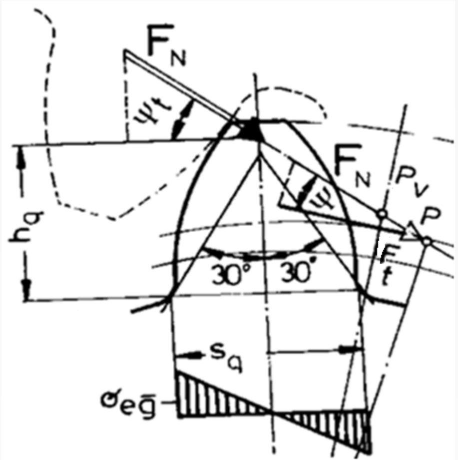
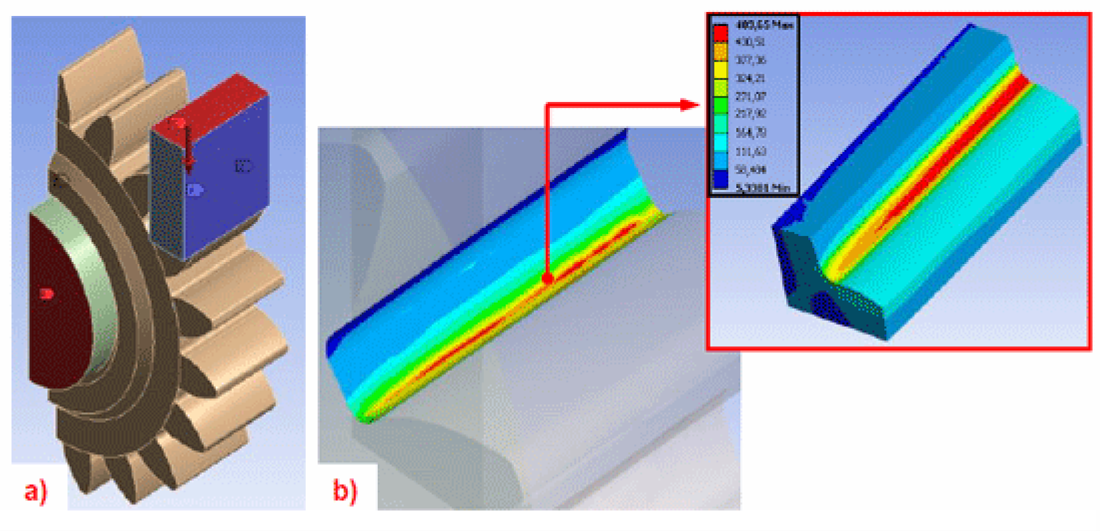




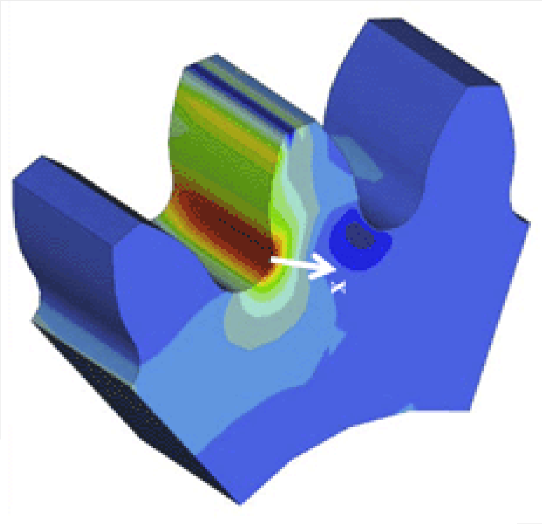
Diş dibi eğilmesi

- Hesaplama kolaylığı için en kötü şartlar gözönüne alınır. Yükün sadece bir diş çifti tarafından taşındığı ve kuvvetin diş başına geldiği düşünülür.
- Optik gerilme analiz sonuçları, en yüksek eğilme gerilmelerinin dişdibinde ve iyi bir yaklaşımla diş dibindeki 30° `lik teğetlerin temas noktalarında, hasıl olduğunu gösterir.





Diş dibi eğilmesi



Taksimat dairesi üzerindeki teğetsel (çevresel) kuvvet,

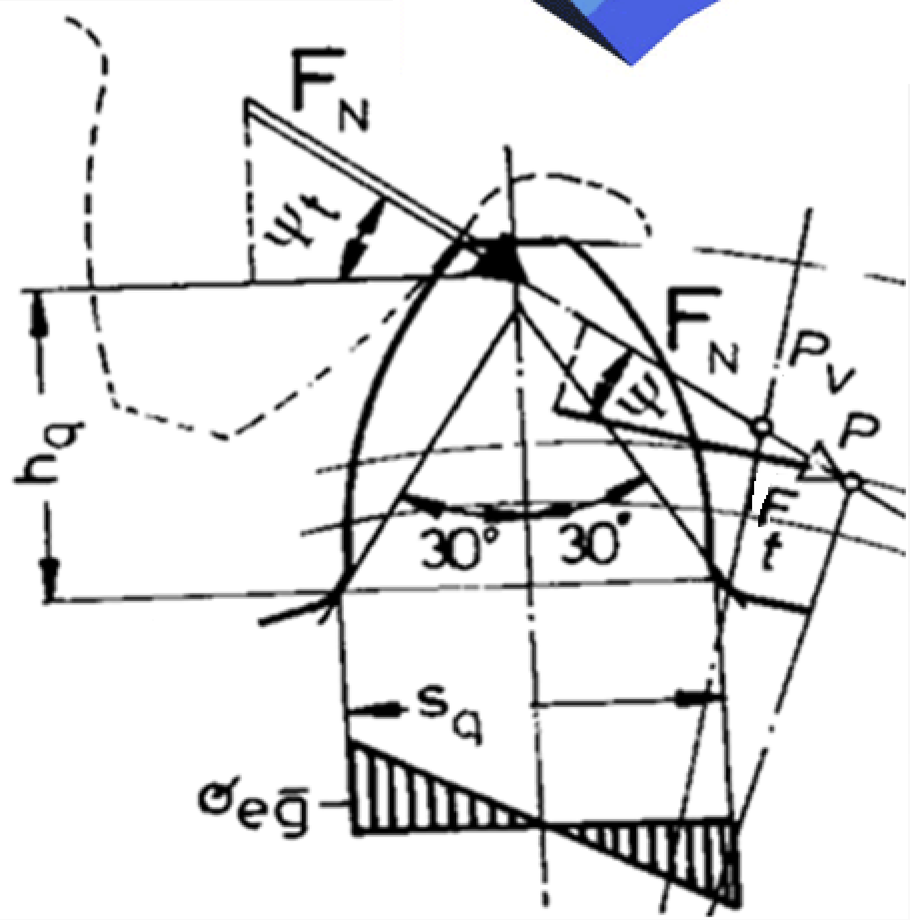
$F_t = M_d / r$

ve normal kuvvet,

$F_N = F_t / \cos\psi$

olarak tanımlanırsa, diş dibindeki eğilme gerilmesi,

$$\sigma_{eğ} = \frac{M_{eğ}}{W_{eğ}} = \frac{F_N . \cos\psi_t h_q}{b . s_q^2 / 6}$$



$$\sigma = \frac{F_t}{bm} Y_F Y_S Y_\beta$$

Diş dibinde eğilme gerilmesi

$$\sigma_{eğ} = \frac{M_{eğ}}{W_{eğ}} = \frac{F_N . \cos\psi_t h_q}{b.s_q^2 / 6}$$

$$\sigma_{eğ} = \frac{F_t . \cos\psi_t \frac{h_q}{m} 6m}{\cos\psi . b \frac{s_q^2}{m^2} m^2}$$

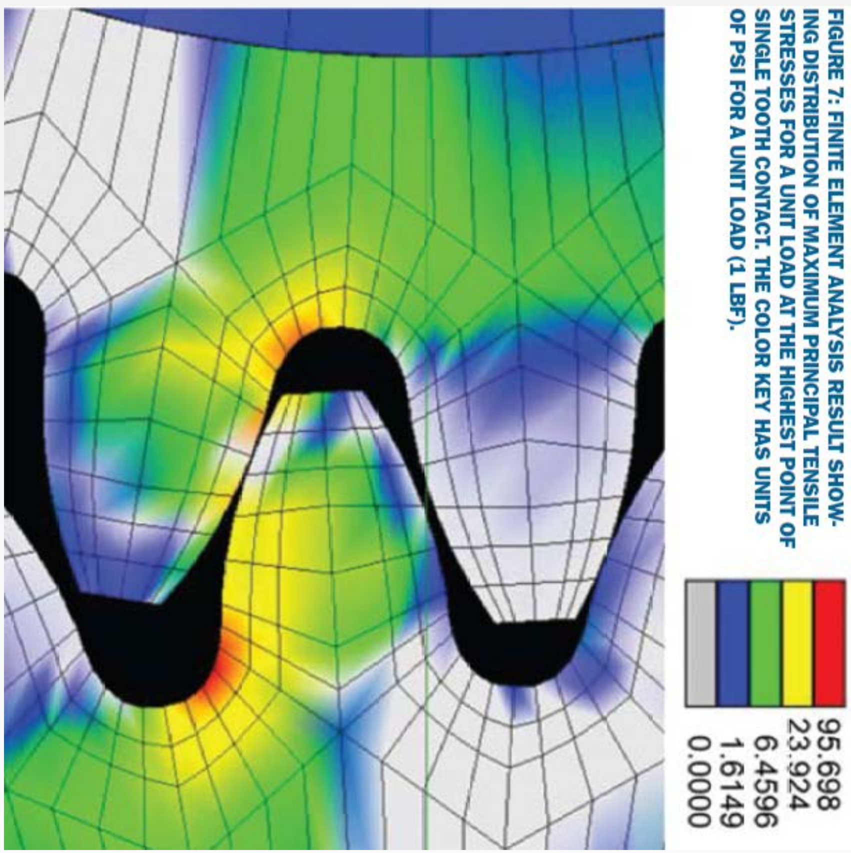


FIGURE 7: FINITE ELEMENT ANALYSIS RESULT SHOWING DISTRIBUTION OF MAXIMUM PRINCIPAL TENSILE STRESSES FOR A UNIT LOAD AT THE HIGHEST POINT OF SINGLE TOOTH CONTACT. THE COLOR KEY HAS UNITS OF PSI FOR A UNIT LOAD (1 LBF).

$$\sigma_{eğ} = \frac{F_t}{bm} \frac{6 \frac{h_q}{m} \cos\psi_t}{(s_q / m)^2 \cos\psi} = \frac{F_t}{bm} q_k$$

$$\sigma = \frac{F_t}{bm} Y_F Y_S Y_\beta$$

Diş dibi eğilmesi

Pinyon ve çark diş dibindeki eğilme gerilmeleri,

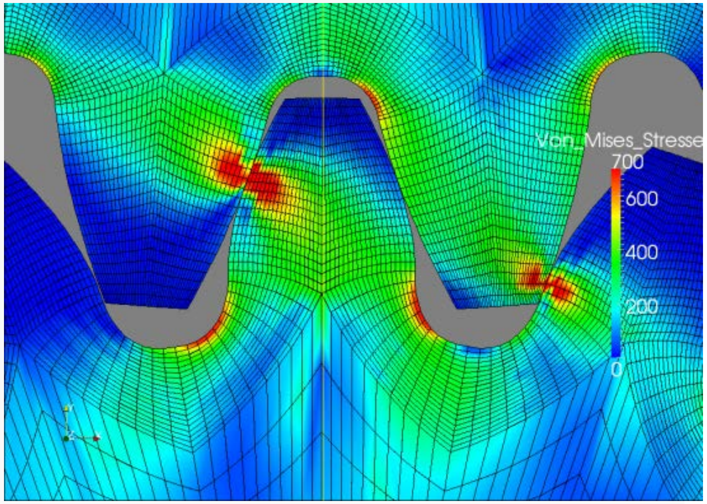
$$\sigma_{e\breve{g}1} = \frac{F_t}{b_1 m} q_{k1} \leqq \sigma_{egem1} \qquad \sigma_{e\breve{g}2} = \frac{F_t}{b_2 m} q_{k2} \leqq \sigma_{egem2}.$$

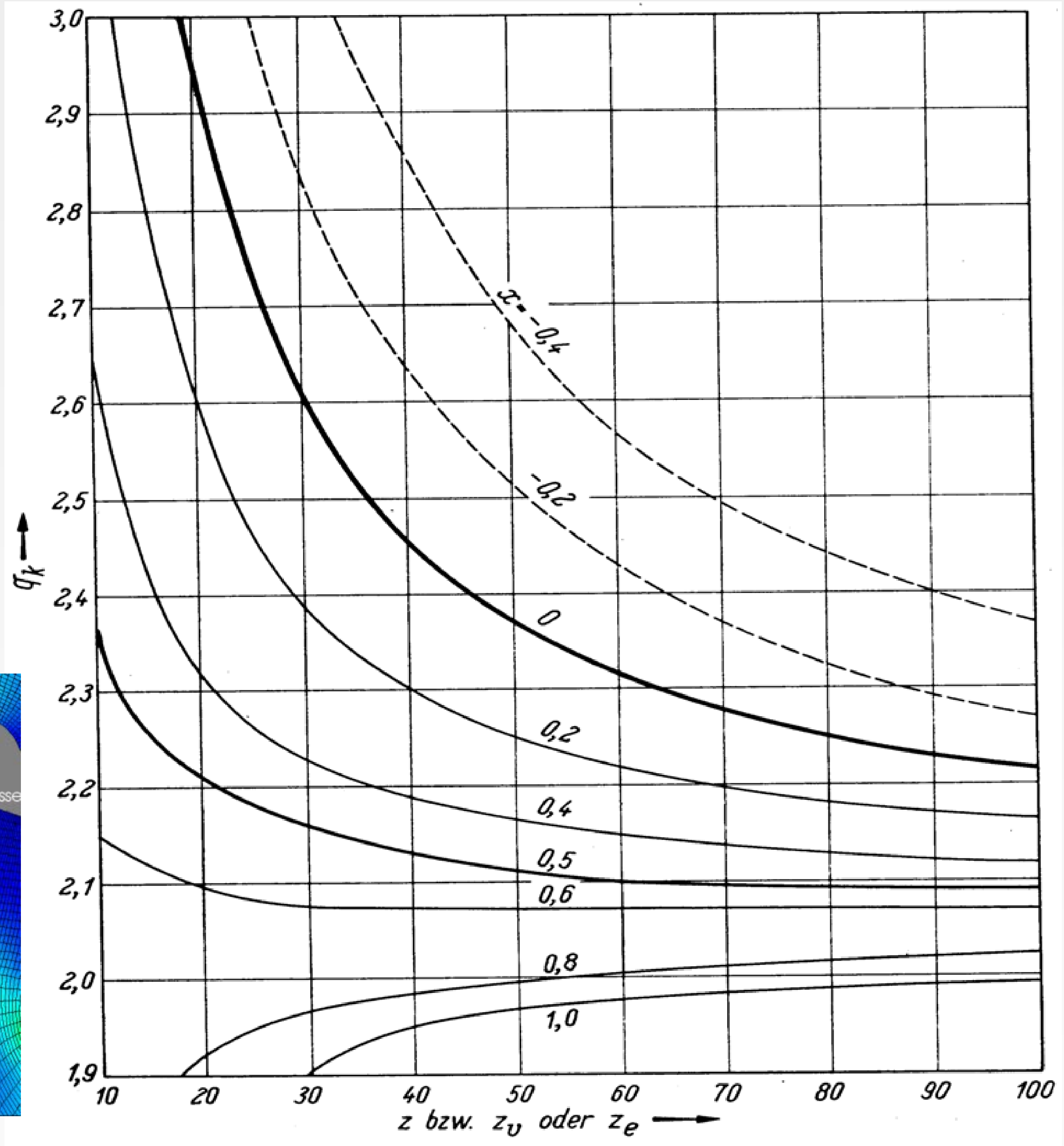
$$\sigma_{e\breve{g}1} = \frac{2M_{d1} q_{k1}}{(b/d_1) m d_1^2} = \frac{2M_{d1} q_{k1}}{(b/d_1) m^3 z_1^2} \leqq \sigma_{e\breve{g}em1}$$

Buradan gerekli diş modülü $$m \geqq \sqrt[3]{\frac{q_{k1}}{z_1^2}} \sqrt[3]{\frac{2M_{d1}}{\frac{b}{d_1} \sigma_{e\breve{g}em1}}}$$

bulunur.

DIN 867 'ye göre olan referans profil için q_k diş form faktörü





Diş sayıları

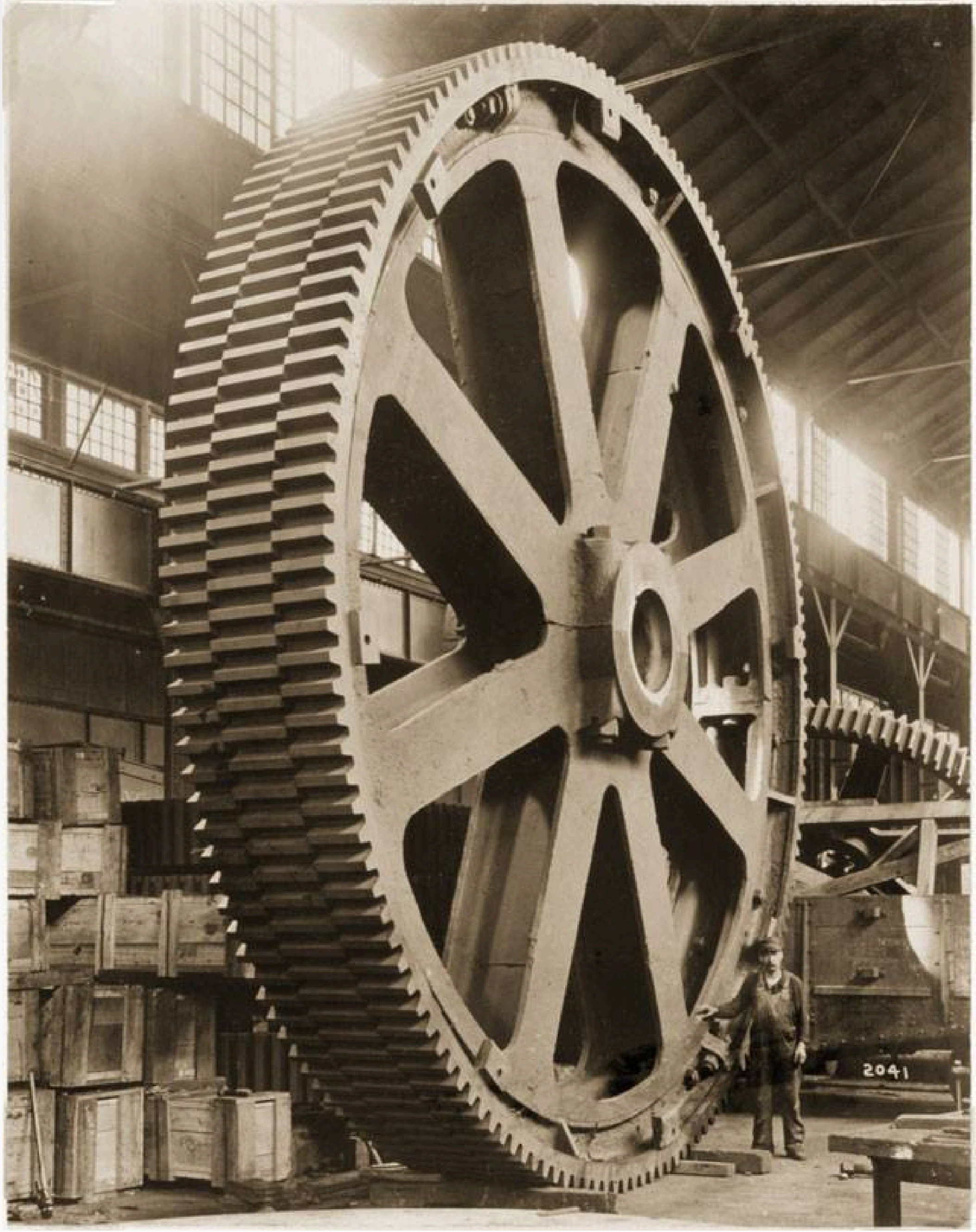
Modülün mümkün olduğu kadar küçük ve diş sayısı $z_1 \geq 20$ olmalıdır.

Düşük devir ve motor tahriğinde $z_1=12$,

el tahriğinde $z_1=7$'ye kadar azaltılabilir.

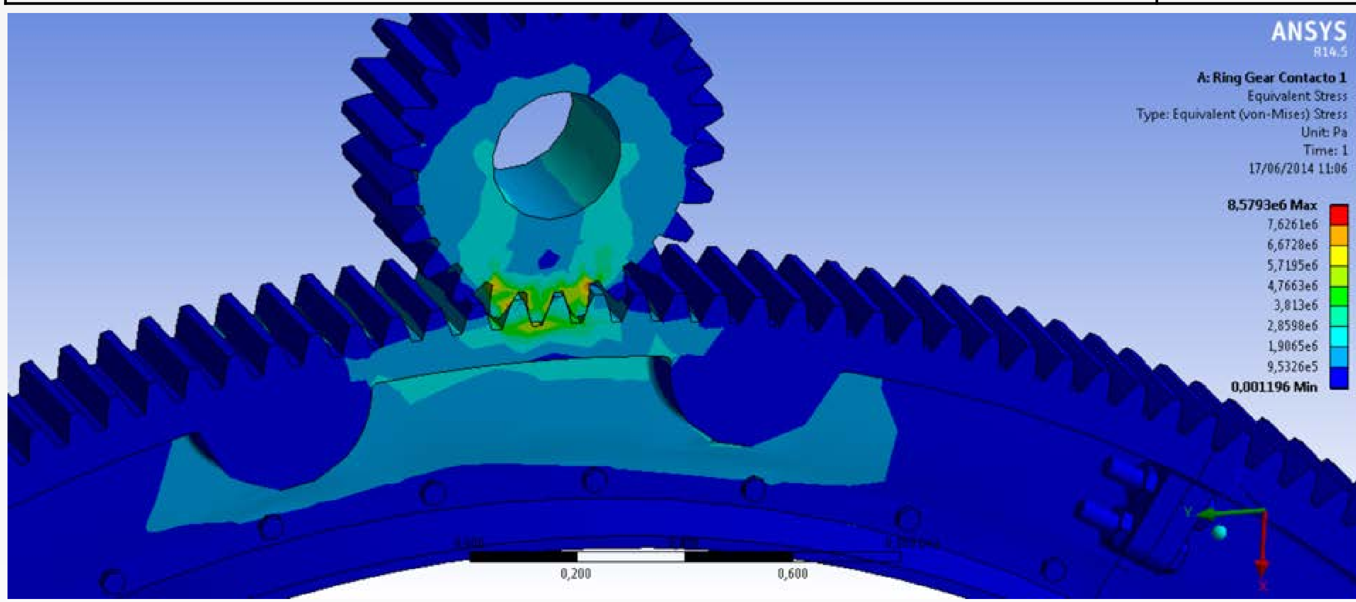
$z_1 \leq 14$`de tashih gereklidir.

$z_1>14$'de $x_1 \approx 0,5$ ve $x_1+x_2 \approx 1$ tavsiye edilir.



Tavsiye edilen b/d_1 değerleri

	b/d_1 değerleri
Dış yüzeyi sertleştirilmiş dişli çarklar	(0,1...0,3...0,5)+i/20
Islah edilmiş, sertleştirilmemiş dişli çarklar	(0,2...0,5...0,8)+i/10
Mil tek taraflı yataklanmış	≤ 0,7
Mil iki taraftan yataklanmış	≤ 1,2



Tavsiye edilen b/m değerleri

	b/m değerleri
İtinalı döküm dişler işlenmemiş	6...10
Dişler işlenmiş, çelik konstrüksiyonda yataklanmış veya tek taraflı yataklanmış	10...15
Dişler itinalı işlenmiş, dişli kutusu içinde yataklanmış	15...25
Dişler çok hassas işlenmiş, dişli kutusu içinde hassas yataklanmış ve yağlanmış, $n_1 \leq 3000$ d/d	25...45
Aynı şekilde $n_1 \geq 3000$ d/d	45...1000
Dişler sertleştirilmiş ve taşlanmış	5...15

Dişli çark üzerindeki moment

Maksimum moment $$M_{d1} = f_B \frac{P}{\omega_1}$$

Burada,

P, dişliler ile aktarılan güç (anma gücü),

ω_1 , pinyon (döndüren dişli) açısal hızı (1/s), $\omega_1 = 2\pi n_1/60$

n_1, giriş (pinyon) deviri (dev/dk)

f_B , tahrik eden ve edilen makinalara bağlı ortaya çıkan darbeleri ve çalışma sıklığını karakterize eden bir faktördür.

	f_B işletme katsayısı için değerler		Tahrik		
Grup •	Makina	Günlük Çalışma Süresi (saat)	Elektrik Motoru	Türbin, Çok Silindirli Motor	Tek Silindirli, Pistonlu Motor
I	**Darbesiz;** Jeneratör, Kayışlı Konveyör, Bandlı Konveyör,	0,5	0,5	0,8	1
	Vidalı Konveyör, Hafif Yüklü Asansörler,Vinçler, Takım	3	0,8	1	1,25
	Tezgahlarının İlerleme Mekanizmaları, Fanlar, Turbo	8	1	1,25	1,5
	Kompresörler, Homojen Yoğunluklar için Mikserler	24	1,25	1,5	1,75
II	**Az Darbeli;** Takım Tezgahlarının Ana Tahrik Mekanizmaları, Ağır	0,5	0,5	1	1,25
	Yüklü Asansörler, Krenlerin Yürütme ve Döndürme Mekanizmaları,	3	0,8	1,25	1,5
	Homojen Olmayan Yoğunluklar için Mikserler, Çok Silindirli Pompalar	8	1	1,5	1,75
		24	1,25	1,75	2
III	**Çok Darbeli;** Dövme ve Kesme Makinaları, Plastik Yapıştırma	0,5	1,25	1,5	1,75
	Presleri,	3	1,5	1,75	2
	Haddeleme ve Demir Çelik Tesisleri, Kepçeli Yükleyiciler, Yüksek Güçlü	8	1,75	2	2,25
	Santrifüjler ve Tevzi Pompaları,				
	Sondaj Makinaları, Briket Presleri,	24	2	2,25	2,5
	Kırıcılar				

Malzeme Emniyeti

Yüzey Basıncı

DIN 3990'da diş yüzeyindeki basınç gerilmesi şu şekilde ifade edilmiştir.

$$\sigma_p = Z_H Z_E Z_\varepsilon Z_\beta \sqrt{\frac{F_t}{d_1 b} \frac{i+1}{i}}$$

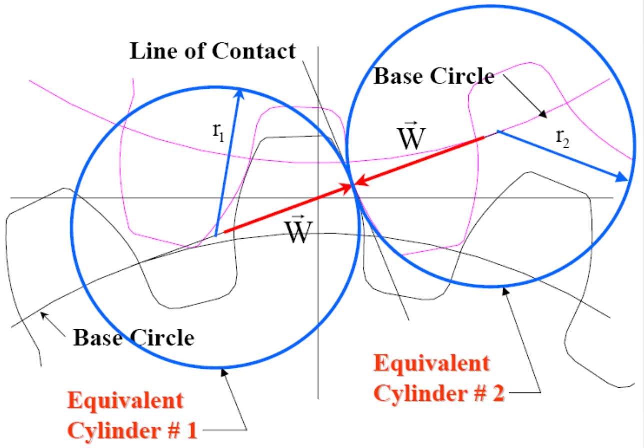
Burada:

Z: DIN Faktörleri

i: çevrim oranı

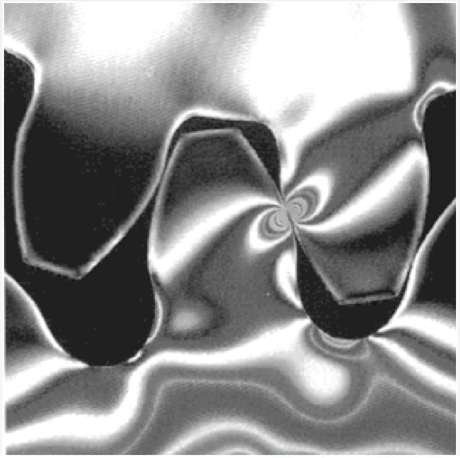
Malzeme		$\sigma_{eğem}$ daN /mm^2	P_{em} daN/mm^2
Dökme Demir	GG-20	4,5	22
	GG-25	5.5	27
Dökme Çelik	GS-52	9	31
	GS-60	10	39
Yapı Çeliği	St 50	11	34
	St 60	12.5	38
	St 70	14	44
Islah Çeliği	C 45	13.5	45
	C60	15	50
	34 Cr 4	18	60
	37 MnSi 5	19	55
	42 CrMo 4	20	63
	35 NiCr 18	20	90
Sementasyonla Sertleştirilmiş Çelik	C 15	12	150
	16 MnCr 5	20	150
	20 MnCr 5	22	150
	15 CrNi 6	21	150
	18 CrNi 8	22	150
Alev veya İnduksiyonla Sertleştirilmiş Çelik	Ck 45	18	135
	37 MnSi 5	20	125
	53 MnSi 4	20	140
	41 Cr 4	20	130
	42 CrMo 4	21	150
Siyanür Banyosunda Sertleştirilmiş	37 MnSi 5	20	125
	35 NiCr 18	22	135
Banyoda Nitrürasyon	C 45	16	75
	16 MnCr 5	17	72
	42 CrMo 4	29	85
Gaz Nitrürasyonu	16 MnCr 5	21	88

Yüzey Mukavemeti



Yüzey mukavemeti hesabı için temas eden iki silindirdeki Hertz basıncı dikkate alınır.

$$p = \sqrt{\frac{1}{2\pi}\frac{m^2}{(m^2-1)}E\frac{F_N}{b}\left(\frac{1}{\rho_1}+\frac{1}{\rho_2}\right)}$$



Burada, m, Poisson sayısı, $E = 2E_1E_2 / (E_1+E_2)$

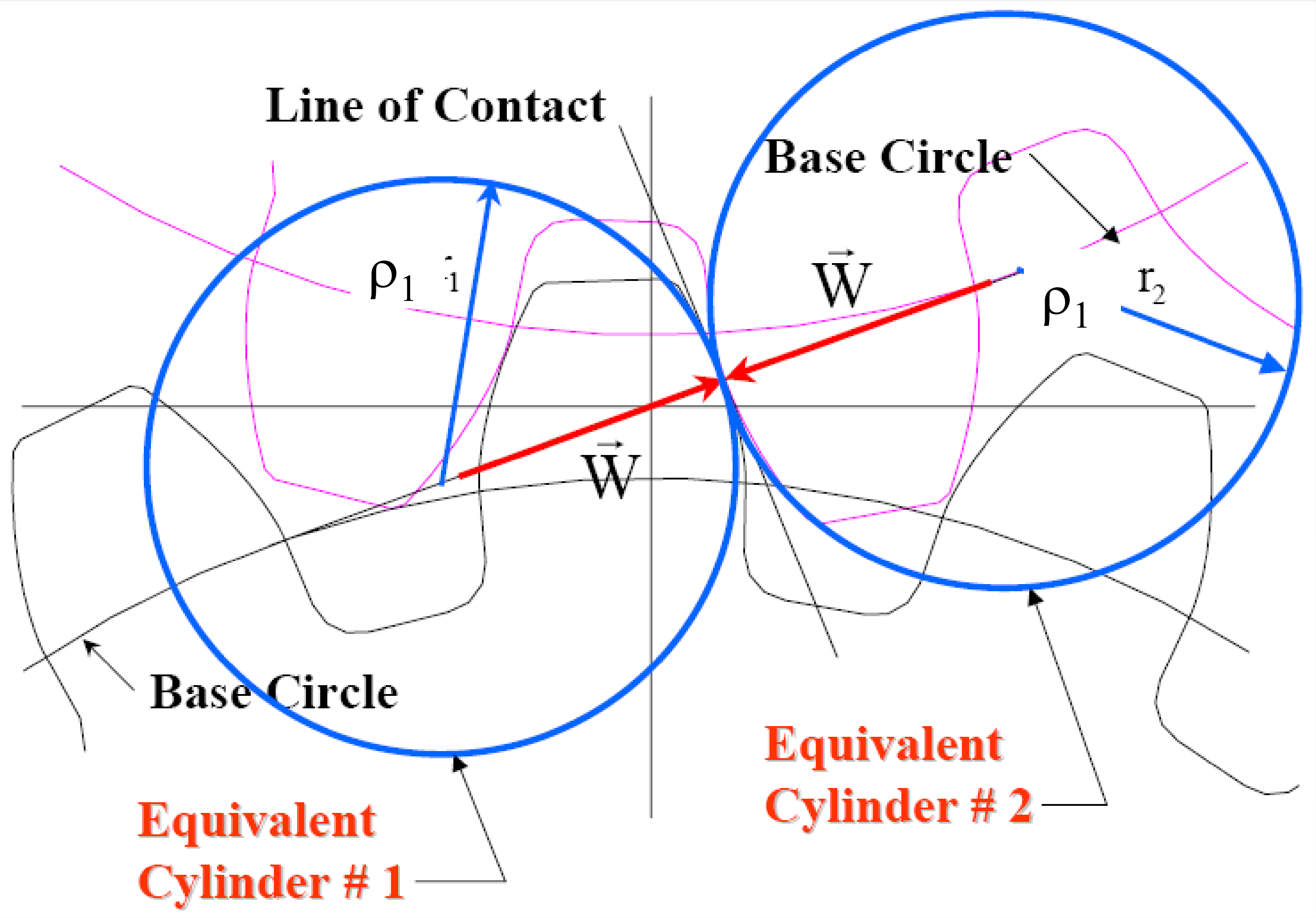
E, malzemeye bağlı ortalama elastisite modülü

F_N , Normal kuvvet

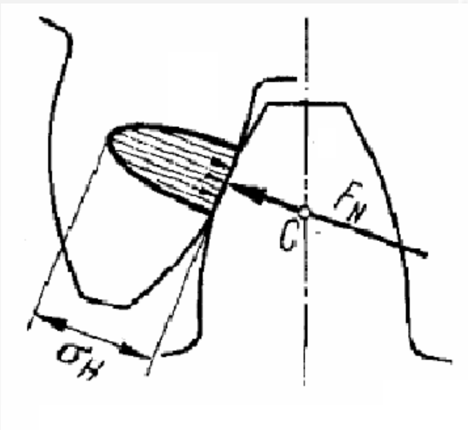
b : Temas eden silindirlerin uzunluğu (diş genişliği)

ρ_1 , ρ_2 : Silindir yarıçapı (temas noktalarında diş profilinin eğrilik yarıçapı)

Dişte ezilme (yüzey mukavemeti)



Yüzey Mukavemeti



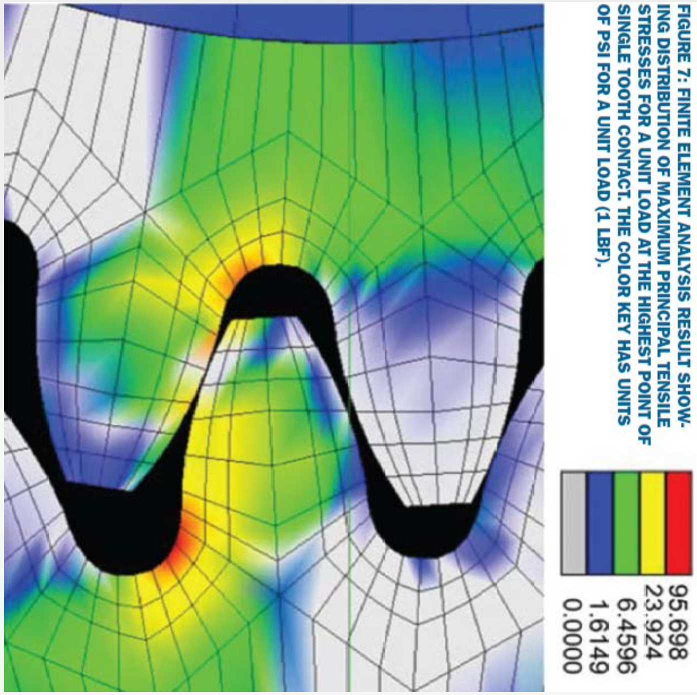
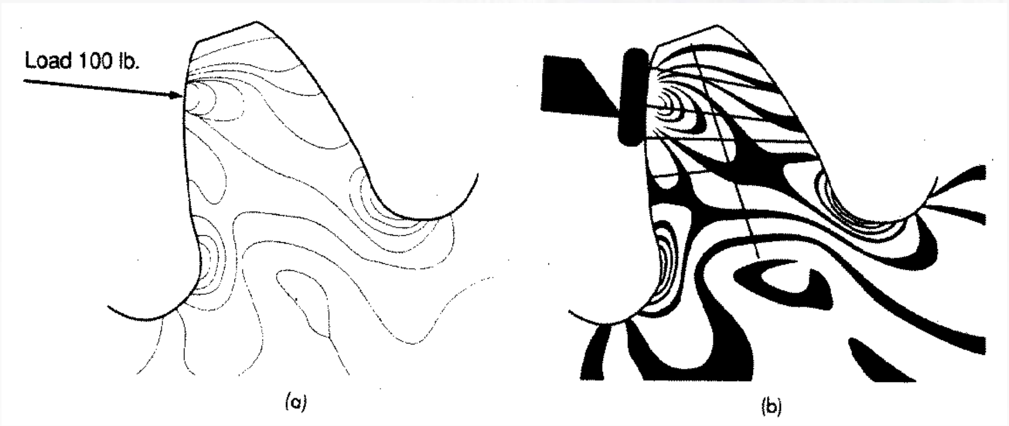
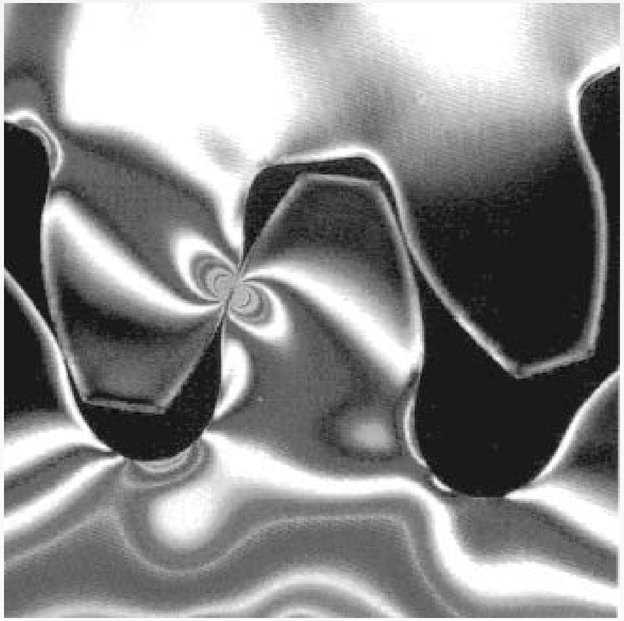
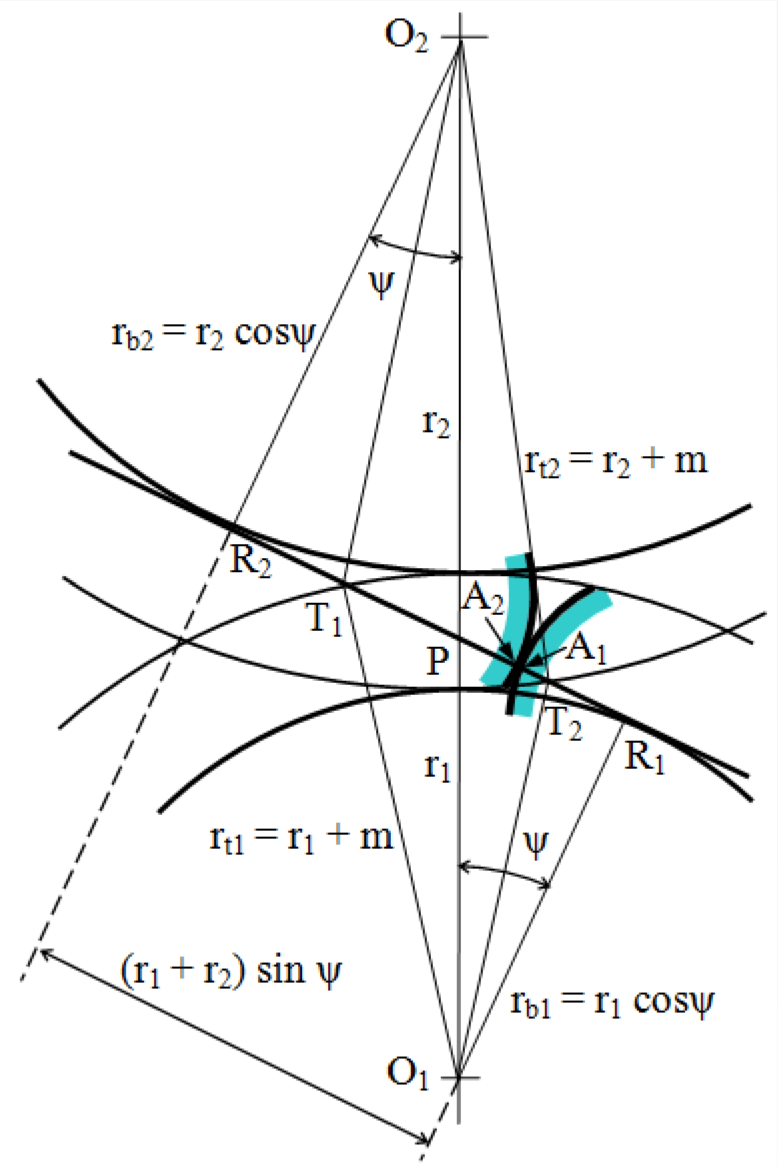


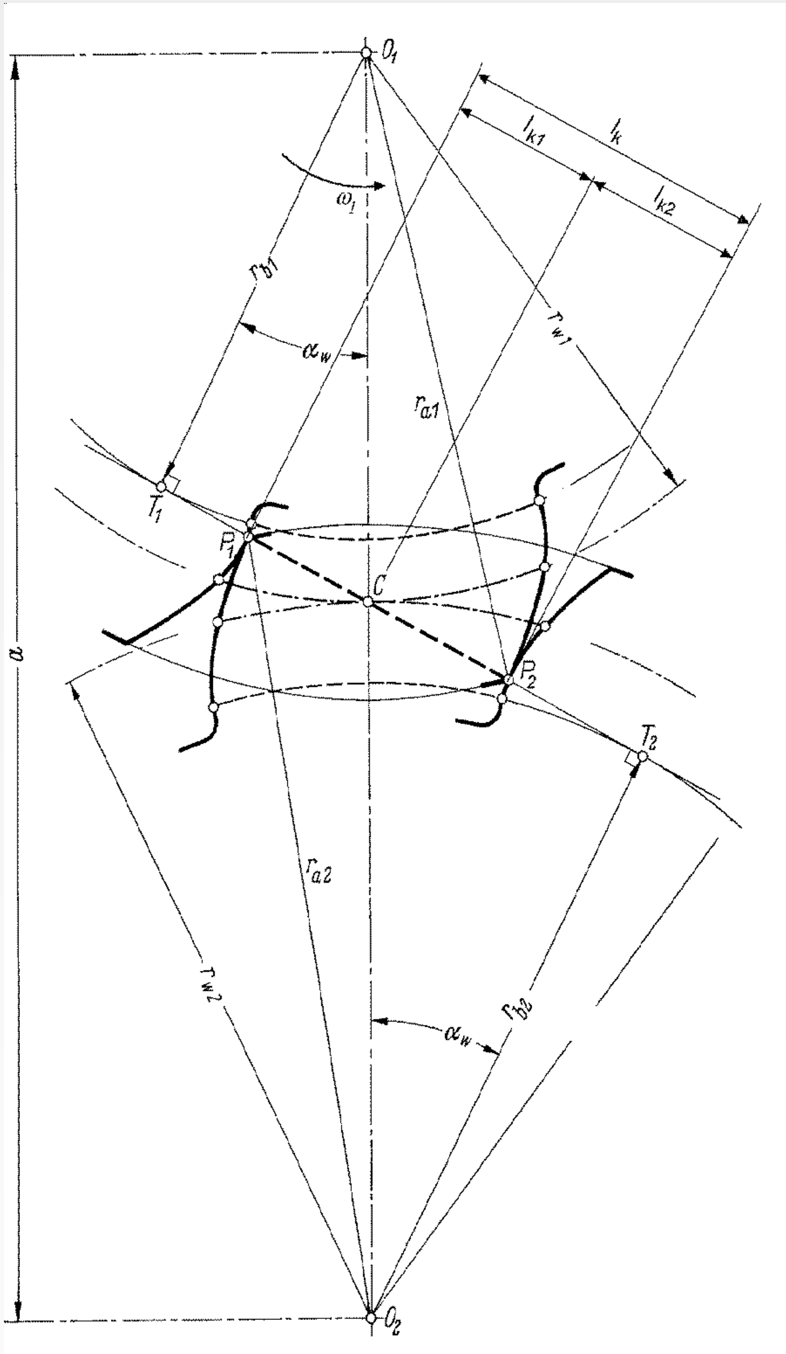
FIGURE 7: FINITE ELEMENT ANALYSIS RESULT SHOWING DISTRIBUTION OF MAXIMUM PRINCIPAL TENSILE STRESSES FOR A UNIT LOAD AT THE HIGHEST POINT OF SINGLE TOOTH CONTACT. THE COLOR KEY HAS UNITS OF PSI FOR A UNIT LOAD (1 LBF).





Kavrama kıtası





Yüzey Mukavemeti

Kavrama doğrusu üzerinde keyfi bir Y noktası için

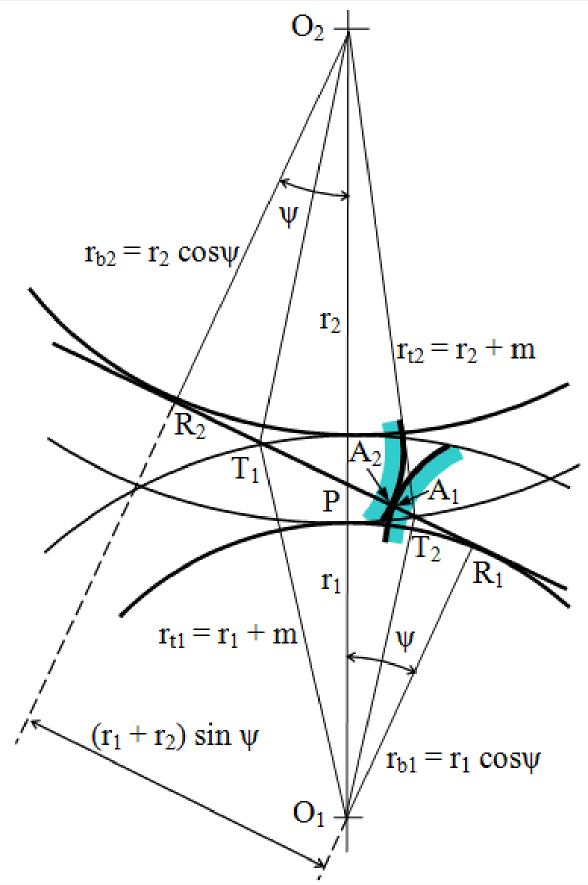
$$\rho_1 = \overline{R_1 Y}$$

$$\rho_2 = \overline{R_2 Y}$$

$$\sqrt{\overline{R_1 R_2} \quad \frac{1}{\rho_1} + \frac{1}{\rho_2}}$$

$$\rho_1 + \rho_2 = \overline{R_1 R_2} = a \sin \alpha_b$$

$$p = \sqrt{\frac{1}{2\pi} \frac{m^2}{(m^2 - 1)} E \frac{F_N}{b} \left(\frac{1}{\rho_1} + \frac{1}{\rho_2}\right)}$$



Yüzey Mukavemeti

$$p = \sqrt{\frac{1}{2\pi}\frac{m^2}{(m^2-1)}E\frac{F_N}{b}\left(\frac{1}{\rho_1}+\frac{1}{\rho_2}\right)}$$

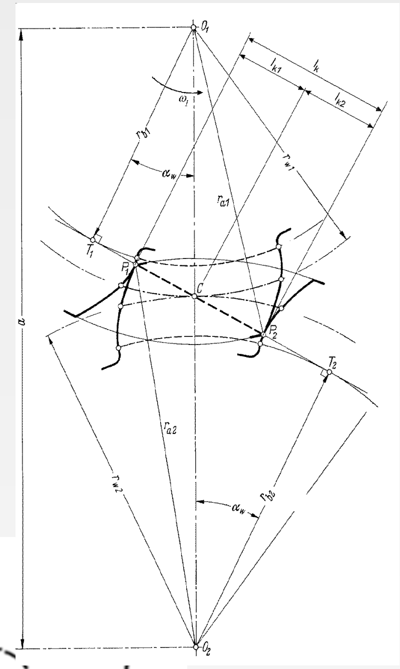
m=10/3 olduğu taktirde: $p = \sqrt{0{,}175E\frac{F_N}{b}\left(\frac{1}{\rho_1}+\frac{1}{\rho_2}\right)}$

Minimum noktası ortadadır, R_1 ve R_2 noktalarının yakınında ise p çok büyük olur.

$z_1 \geq 20$ olması durumunda genel olarak yuvarlanma noktası P 'deki Hertz basıncı p_c hesaplanır.

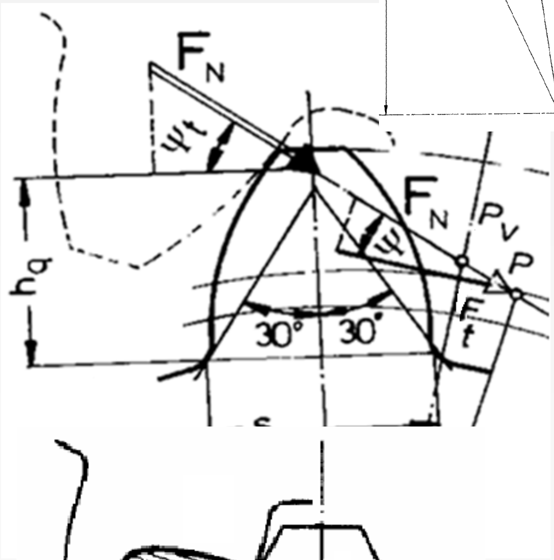
Bu durumda $\rho_{1c} = r_{e1}\sin\psi_v; \quad \rho_{2c} = r_{e2}\sin\psi_v$

Yüzey Mukavemeti

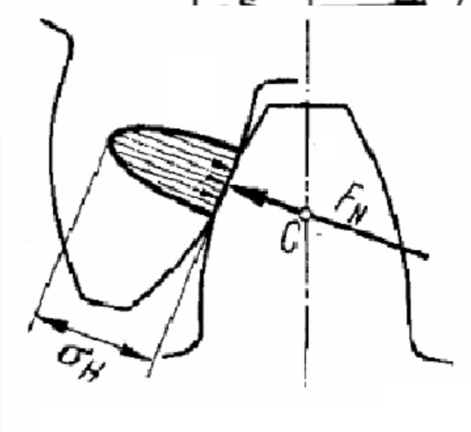


$$\frac{1}{\rho_{1c}}+\frac{1}{\rho_{2c}}=\frac{1}{\sin\psi_v}\left(\frac{1}{r_{e1}}+\frac{1}{r_{e2}}\right)=\frac{1}{r_{e1}\sin\psi_v}\left(1+\frac{1}{i}\right)$$

$$r_{e1}=r_1\frac{\cos\psi}{\cos\psi_v}$$



$$\frac{1}{\rho_{1c}}+\frac{1}{\rho_{2c}}=\frac{\cos\psi_v}{r_1\cos\psi\sin\psi_v}\frac{i+1}{i}.$$

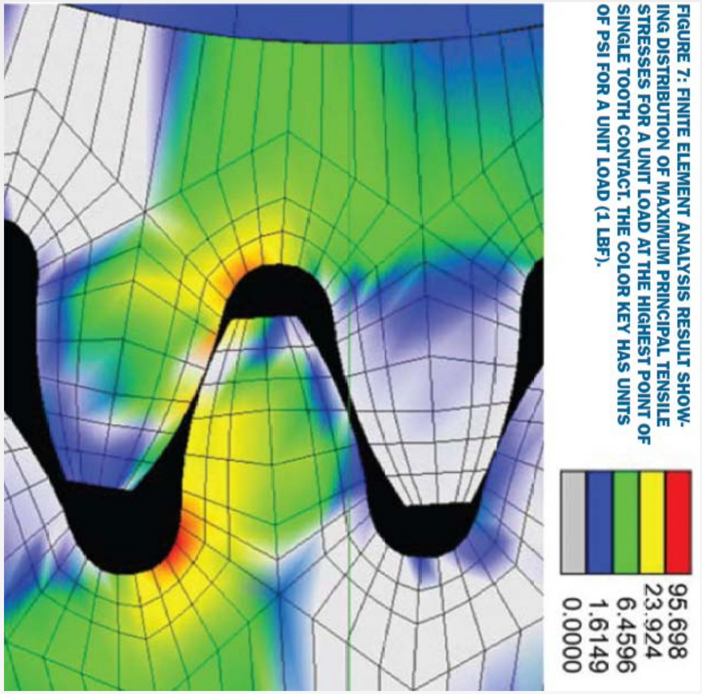


$$F_N=\frac{F_t}{\cos\psi}$$ $$F_R=F_t\tan\psi$$

Yüzey Mukavemeti

$$p_c = \sqrt{0{,}175E \frac{Ft}{b\cos\psi} \frac{\cos\psi_v}{r_1 \cos\psi \sin\psi_v} \frac{i+1}{i}}$$

$$p_c = \sqrt{0{,}35E} \sqrt{\frac{Ft}{bd_1} \frac{i+1}{i}} \sqrt{\frac{\cos\psi_v}{\sin\psi_v \cos^2\psi}}$$

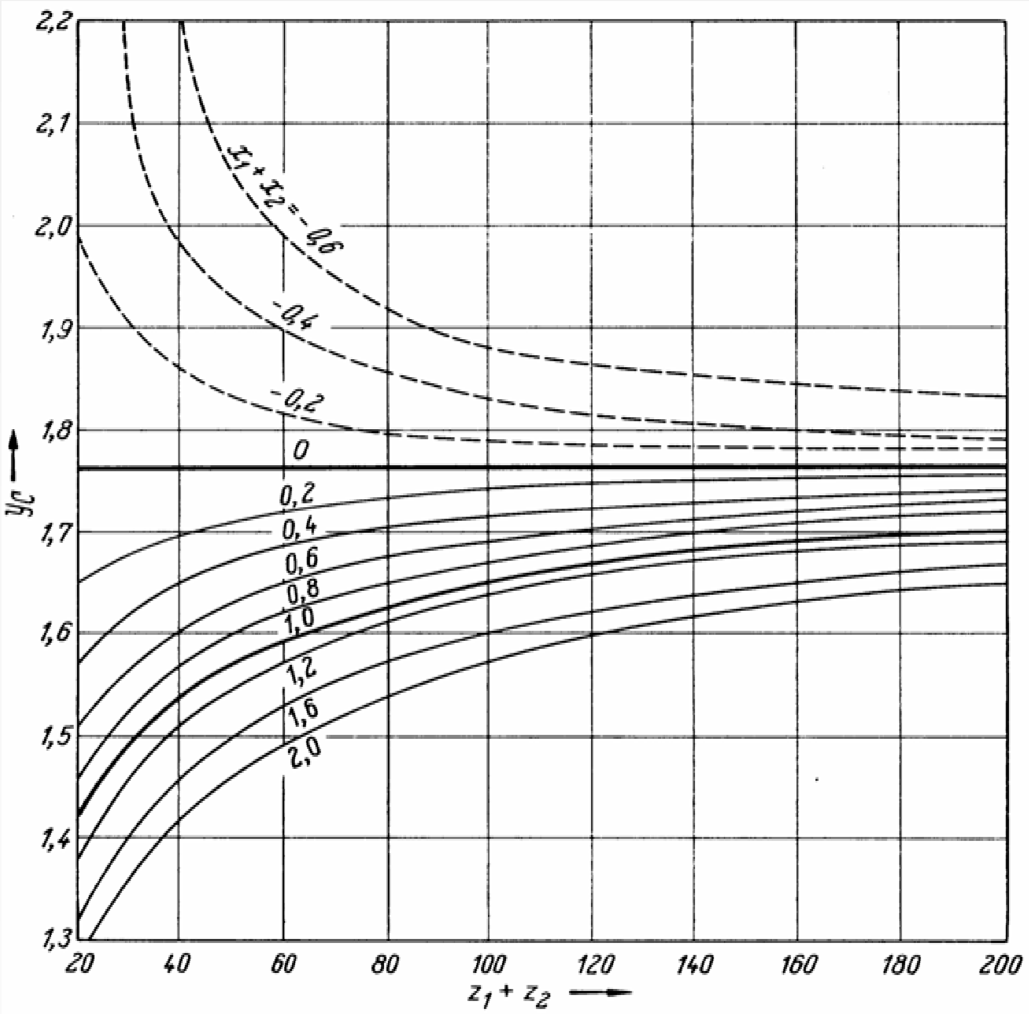


İfadedeki sonuncu köke (y_c) yuvarlanma noktası faktörü adı verilir.

$$p_c = \sqrt{0{,}35E} \sqrt{\frac{F_t}{bd_1} \frac{i+1}{i}}\, y_c$$

$$\sigma_p = Z_H Z_E Z_\varepsilon Z_\beta \sqrt{\frac{F_t}{d_1 b} \frac{i+1}{i}}$$

y_c yuvarlanma noktası faktörü



V-mekanizmalarındaki Hertz gerilmelerinin az olmasını, diyagramdaki (x_1+x_2) değerlerinin artması ile y_c değerlerinin azalışı göstermektedir. Negatif (x_1+x_2) değerleri, bilhassa küçük diş sayılarında, büyük y_c değerleri vermektedir.

Zaman mukavemeti (yorulma) ve pitting

Yorulma deneylerinden (Wöhler-Eğrileri) belirli tekrar sayısında ($N \geq 10^8$ yük değişim sayısı) taşınabilen basınç P_D bulunabilir.

$$P_{em} = P_D / S$$

Pitting hadisesinde hasar ani olarak meydana gelip mekanizmayı durdurmayacağı için, diş dibi mukavemetine nazaran daha düşük emniyet değerleri ($S \approx 1,2...1,3$, nitrürasyon işlemine tabi çarklarda 1,8) yeterlidir.

Zaman mukavemeti (yorulma)

Zaman mukavemetinin (DUDLEY ve WINTER 'e göre) hesaplanmasında aşağıdaki faktörler kullanılabilir.

$N=10^6...10^7$ yük tekrarında $S=1,1$

$N=10^5...10^6$ yük tekrarında $S=1,25$

$N<10^5$ yük tekrarında $S=1,4$

Yük tekrar sayısı N_1 tam yükte ömür L_h [saat], devir sayısı n [devir/dak.] ve bir çarktaki diş temas sayısı z'ye bağlı olarak şu şekilde hesaplanır:

$$N_1 = 60\ n\ z\ L_h$$

$$p_c = \sqrt{0{,}35E} \quad \sqrt{\frac{F_t}{bd_1}\frac{i+1}{i}}\ y_c$$

Yüzey Mukavemeti

$F_t = M_{d1} / r_1 = 2\ M_{d1} / d_1$ ve $d_1 = mz_1$ alınırsa, dişlerin yuvarlanma noktasındaki ezilme basıncı,

$$p_c^{\ 2} = 0{,}35E\frac{2M_{d1}}{d_1 bd_1}\frac{i+1}{i}y_c^{\ 2} = \frac{0{,}7EM_{d1}}{(\frac{b}{d_1})m^3 Z_1^3}\frac{i+1}{i}y_c^{\ 2} \le p_{em}^2$$

olur. Buradan dişlerin ezilme emniyeti için gerekli diş modülü

$$m \ge \frac{1}{Z_1}\sqrt[3]{\frac{0{,}7EM_{d1}}{(\frac{b}{d_1})\,p^2_{em}}\frac{i+1}{i}y_c^{\ 2}}$$

bulunur.

Metal olmayan dişli çarklar

Metal olmayan malzemelerden, dişli çark imalinde Vulkanize fiber (Dynopas) ve preslenmiş suni reçine (Lignofol, Ferrozell, Resitex, Novotext vb.), ayrıca Polyamid (mesela Ultramid A ,B ve S) Acetal reçineler (Polyoxymetylen, Delrin) kullanılır.

Dişlilerin taşıyabileceği

teğetsel kuvvet değeri $F_t = cbm\pi y$

denklemiyle hesaplanır.

Burada, $(b/m) \approx 10$ ve

malzeme faktörü c- değeridir.

Metal olmayan dişli çarklar

Üstün özelliklerine misal olarak düşük özgül ağırlık (1,2 - 1,4 gr / cm^3), mükemmel sönümleme kabiliyeti (sessiz çalışma), korozyona ve aşınmaya karşı mukavemet, uygun sürtünme davranışı (yağsız ve az yağlı çalışabilme) ve talaşsız imalat imkanı (presleme ve enjeksiyon) söylenebilir

Metal olmayan dişli çarklar

$F_t = cbm\pi y$ teğetsel kuvvet denklemindeki

c- değerleri (Malzeme faktörü)

ν	0,5	1	2	3	4	5	6	8	10	12	15	m/s
Dyno	0,35	0,30	0,23	0,20	0,18	0,17	0,16	0,145	0,13	0,125	0,125	kg/mm^2
Ferr	0,26	0,24	0,22	0,20	0,18	0,16	0,15	0,13	0,115	0,105	0,10	kg/mm^2

y- değerleri (Diş sayısı faktörü)

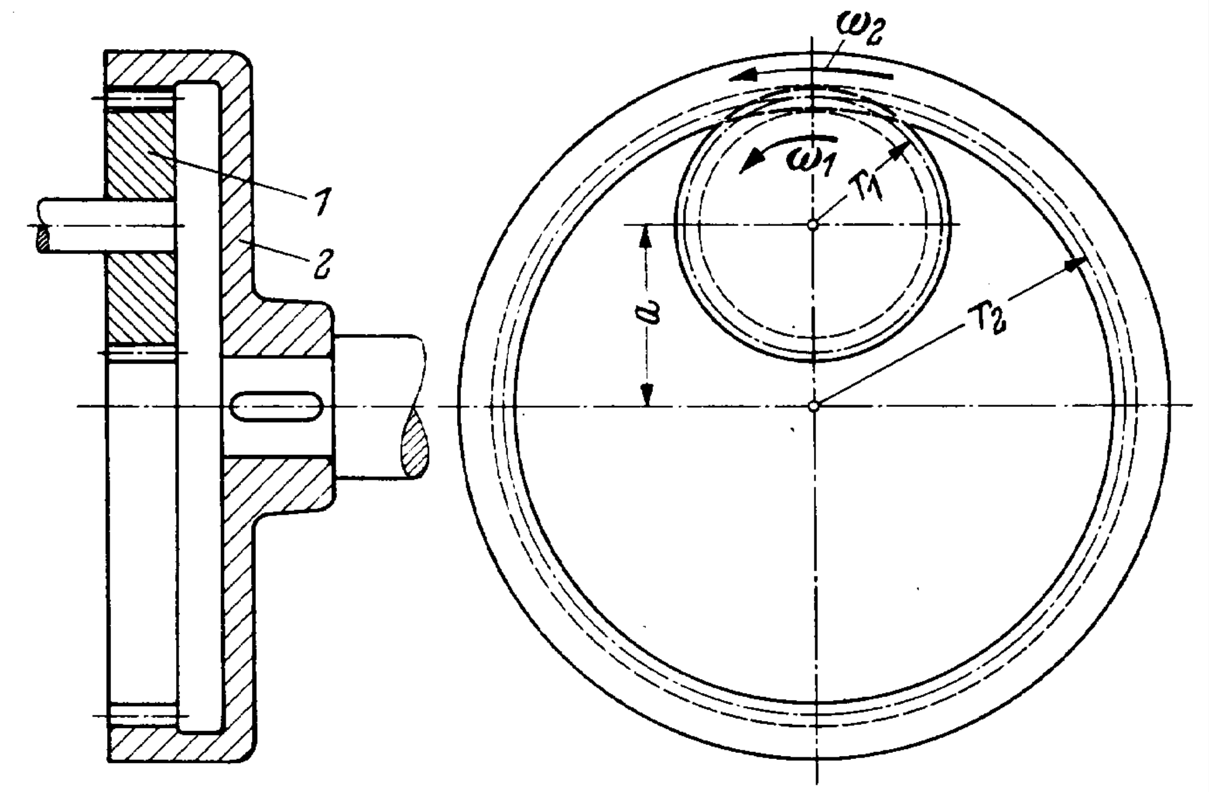
z =	12	14	16	20	30	40	50	70	100	150	200
Dynopas	0,64	0,75	0,85	1,00	1,25	1,40	1,50	1,63	1,73	1,81	1,86
Ferrozel	0,70	0,80	0,88	0,95	1,05	1,10	1,15	1,28	1,35	1,40	

İç Dişliler

çok büyük boyutlu bir iç dişli çarkının merkezlenmesi için, büyük çaplı bir yatak yerine, plastik kam tekerlek parçalar kullanılması

Düz İç Alın Dişliler

İç dişliler sadece bıçak dişli ile yuvarlanma metodu ile işlenebilir, taşlama sadece form metodu ile mümkündür

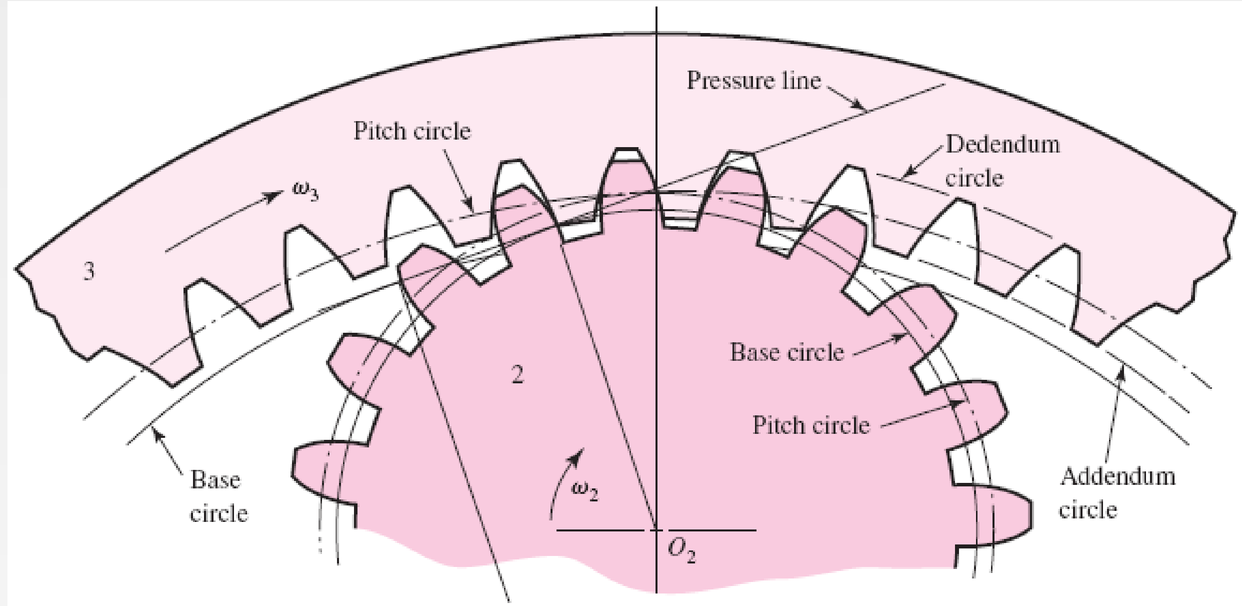


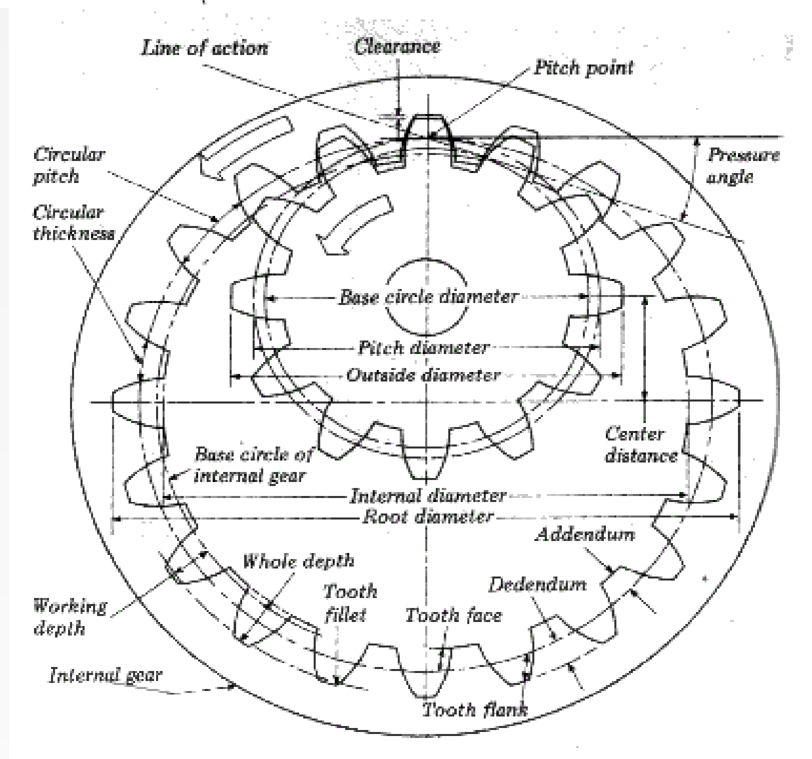
İç dişli çarklar

İki diş başının girişimi (takılması): Baş dairelerinin kesişme noktasına çark dişi, pinyon dişinden evvel erişmelidir.

$z_2 - z_1 \geq 12$

r_{t2} takılma önlenmiş diş başı yarıçapıdır.





Düz İç Alın Dişliler

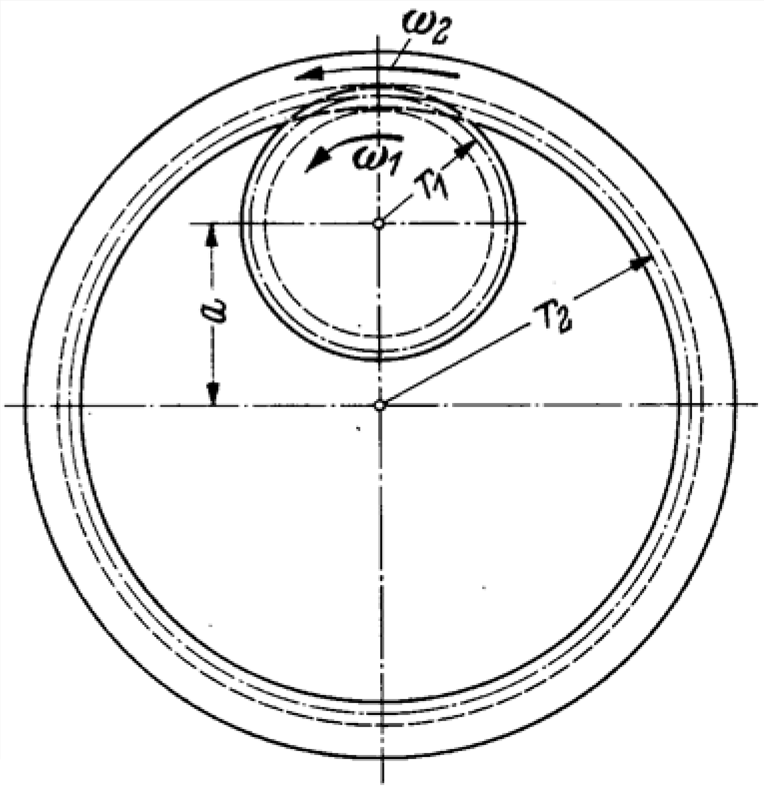
Çevrim (tahvil) oranı

$$i = \frac{n_1}{n_2} = \frac{r_{e2}}{r_{e1}} = \frac{r_2}{r_1} = \frac{r_{b2}}{r_{b1}} = \frac{Z_2}{Z_1}$$

Eksenler arası mesafe

$$a = r_2 - r_1 = \frac{m}{2}(z_2 - z_1)$$

$$a = r_{e2} - r_{e1} \qquad r_{e1} = \frac{a}{i-1} \qquad r_{e2} = \frac{a\,i}{i-1}$$



Eş çalışma kavrama açısı $\cos\psi_v = \frac{a}{a_v}\cos\psi$

Düz İç Alın Dişliler

dış dişlide $x_1+x_2 = (z_1+z_2)\,(\varphi_v - \varphi) / (2\,tg\psi)$

$\varphi_v = 2\,[(x_1+x_2)/(z_1+z_2)]\;tg\psi + \varphi$

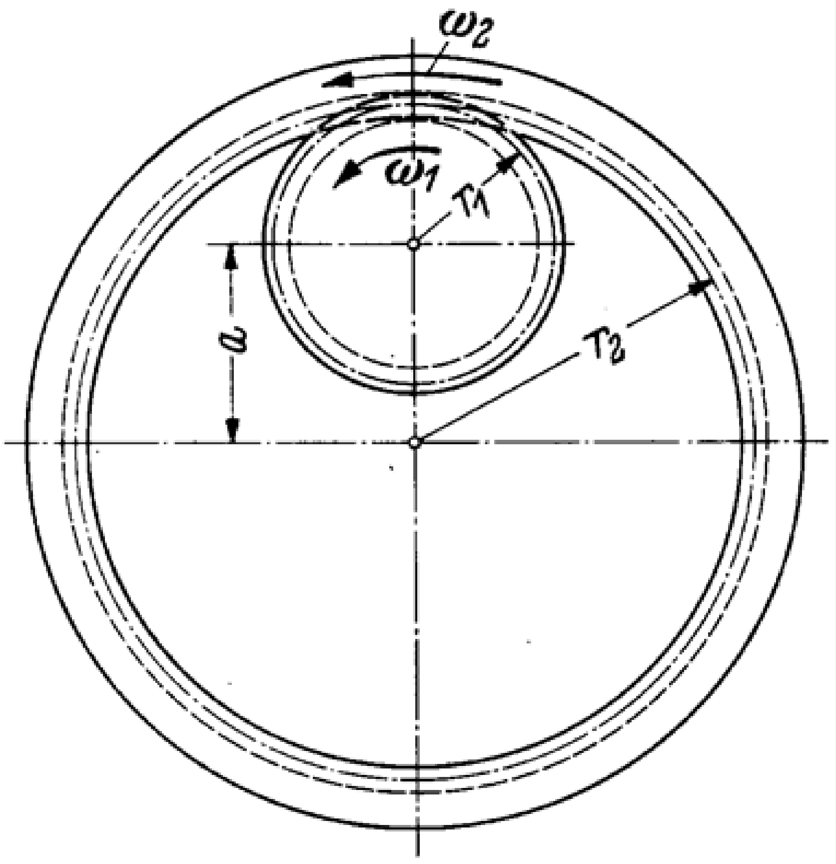
İç dişli profil kaydırma oranı

$$x_2 - x_1 = \frac{(z_2 - z_1)(ev\psi_v - ev\psi)}{2\tan\psi}$$

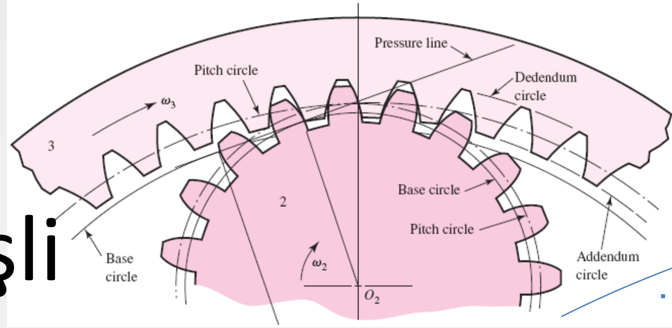
$$ev\psi_v = 2\frac{x_2 - x_1}{z_2 - z_1}\tan\psi + ev\psi$$

Eksenler arası mesafe

$$a_v = a\frac{\cos\psi}{\cos\psi_v} = \frac{m}{2}(z_2 - z_1)\frac{\cos\psi}{\cos\psi_v}$$



İç Dişli



Kavrama Oranı

$$\varepsilon = g \,/\, t_b,$$

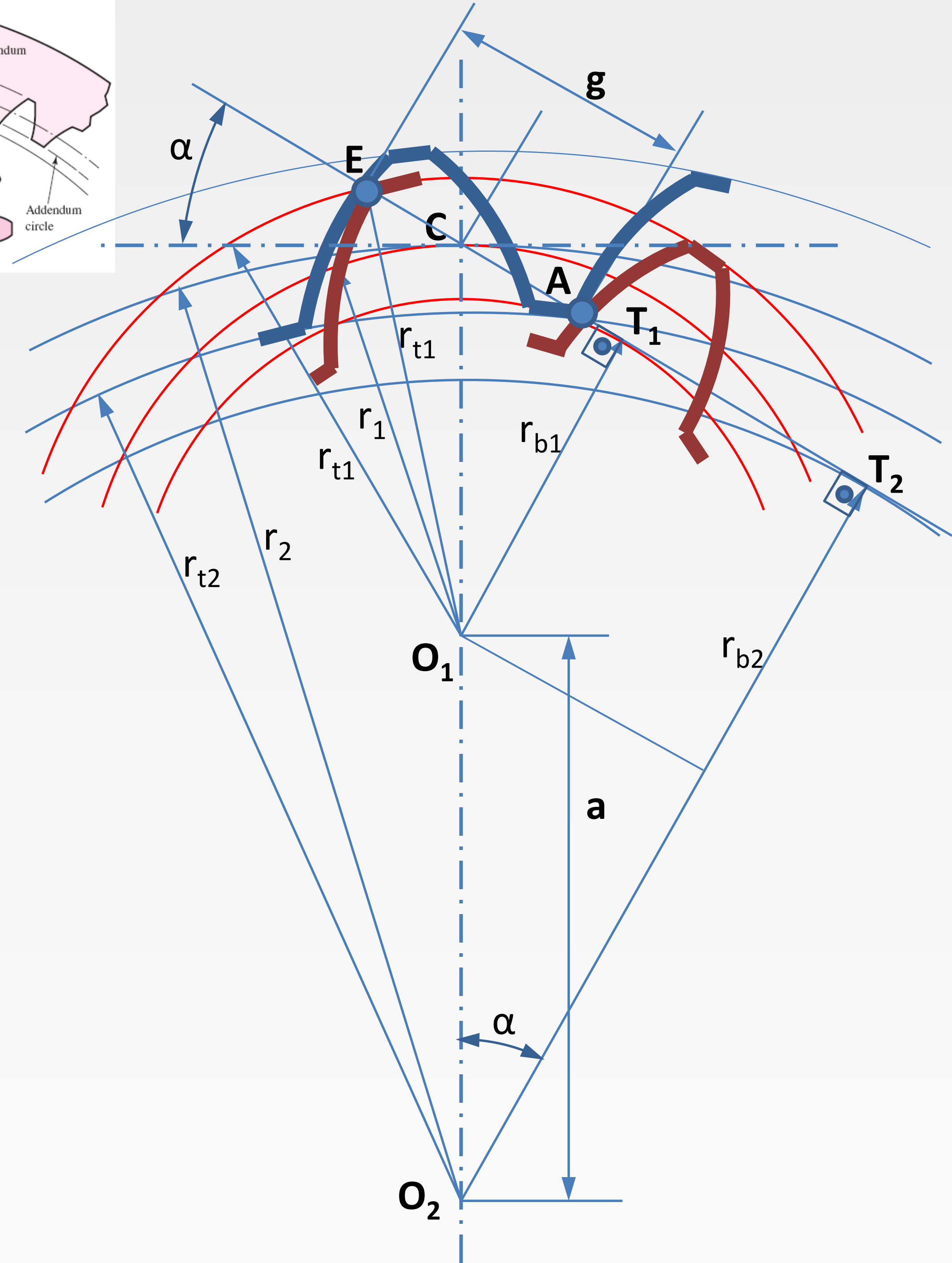
$$g = \overline{AE} = \overline{T_1E} - \overline{T_1A}$$

$$g = \overline{T_1E} - \overline{T_2A} + \overline{T_1T_2}$$

$$\overline{T_1E} = \sqrt{r_{t1}^2 - r_{b1}^2};$$

$$\overline{T_2A} = \sqrt{r_{t2}^2 - r_{b2}^2};$$

$$\overline{T_1T_2} = \sqrt{a^2 - (r_{b2} - r_{b1})^2}$$



$$p_c = \sqrt{0{,}35E}\ \sqrt{\frac{F_t}{bd_1}\frac{i+1}{i}}\,y_c$$

Düz İç Alın Dişliler

Dişlerin (c) yuvarlanma noktasındaki ezilme basıncı,

$$p_c = \sqrt{0{,}175E\frac{F_N}{b}\left(\frac{1}{\rho_{1c}} - \frac{1}{\rho_{2c}}\right)}$$

yuvarlanma noktası faktörü (y_c) ile

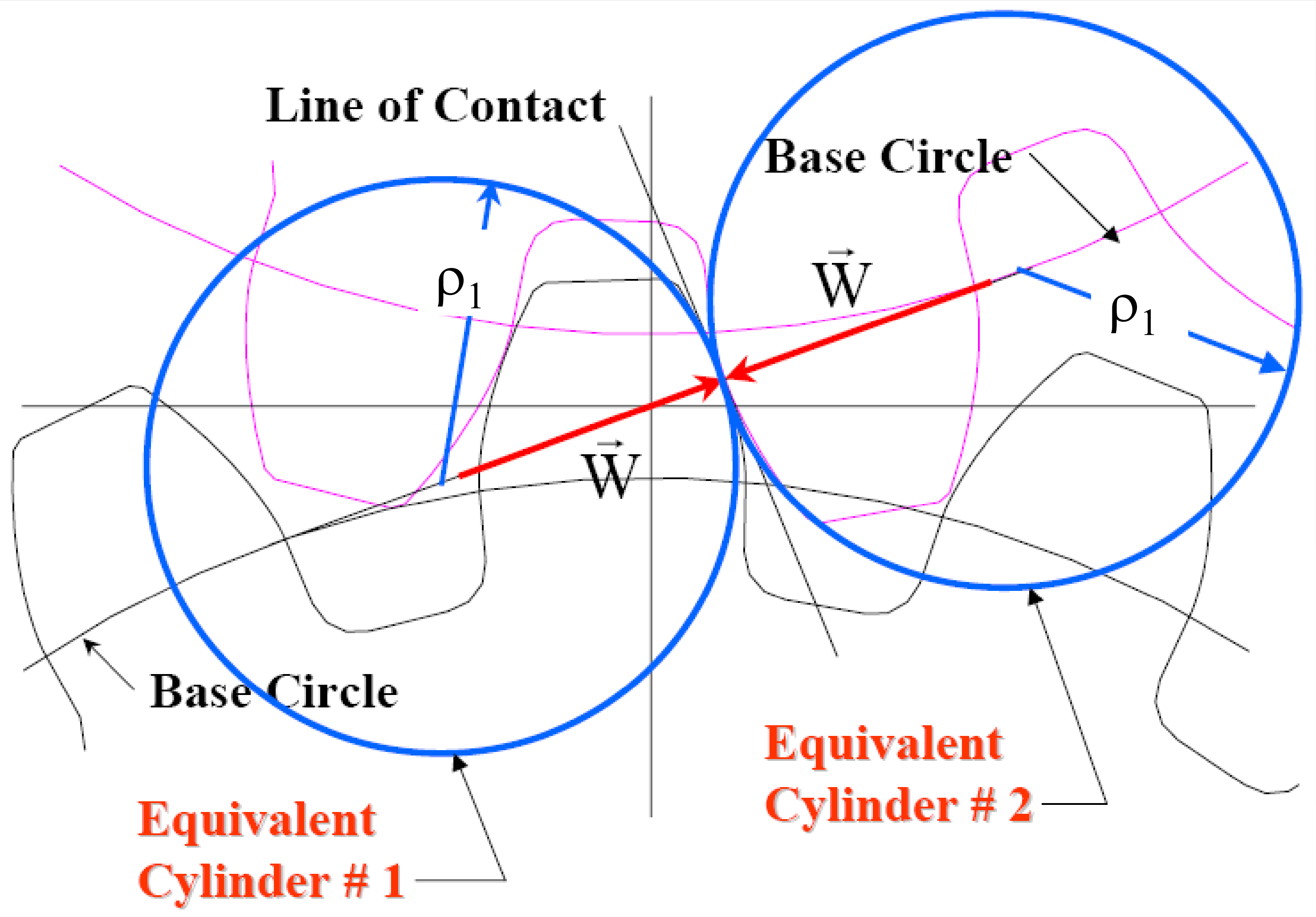
$$y_c = \sqrt{\frac{1}{\sin\psi\cos\psi}} = 1{,}764$$

$$p_c = \sqrt{0{,}35E}\sqrt{\frac{F_t}{bd_1}\frac{i-1}{i}}\,y_c$$

olur. Buradan dişlerin ezilme emniyeti için gerekli diş modülü

$$m \geqq \frac{1}{z_1}\sqrt[3]{\frac{0{,}7EM_{d1}}{(b/d_1)p^2_{em}}\frac{i-1}{i}y_c^2} = \frac{1}{Z_1}\sqrt[3]{\frac{2{,}18EM_{d1}}{(b/d_1)\,p^2_{em}}\frac{i-1}{i}}$$ bulunur.

Dişte ezilme (yüzey mukavemeti)



Dişte ezilme (yüzey mukavemeti)

Eşdeğer yuvarlanan silindirler ve yarıçaplar ρ_1, ρ_2

ρ_1=(r_1.sinψ) , ρ_2=(r_2.sinψ)

burada r_1 ve r_2 pinyon ve çark yuvarlanma (taksimat) dairesi yarıçaplarıdır.

Elastiklik katsayısı

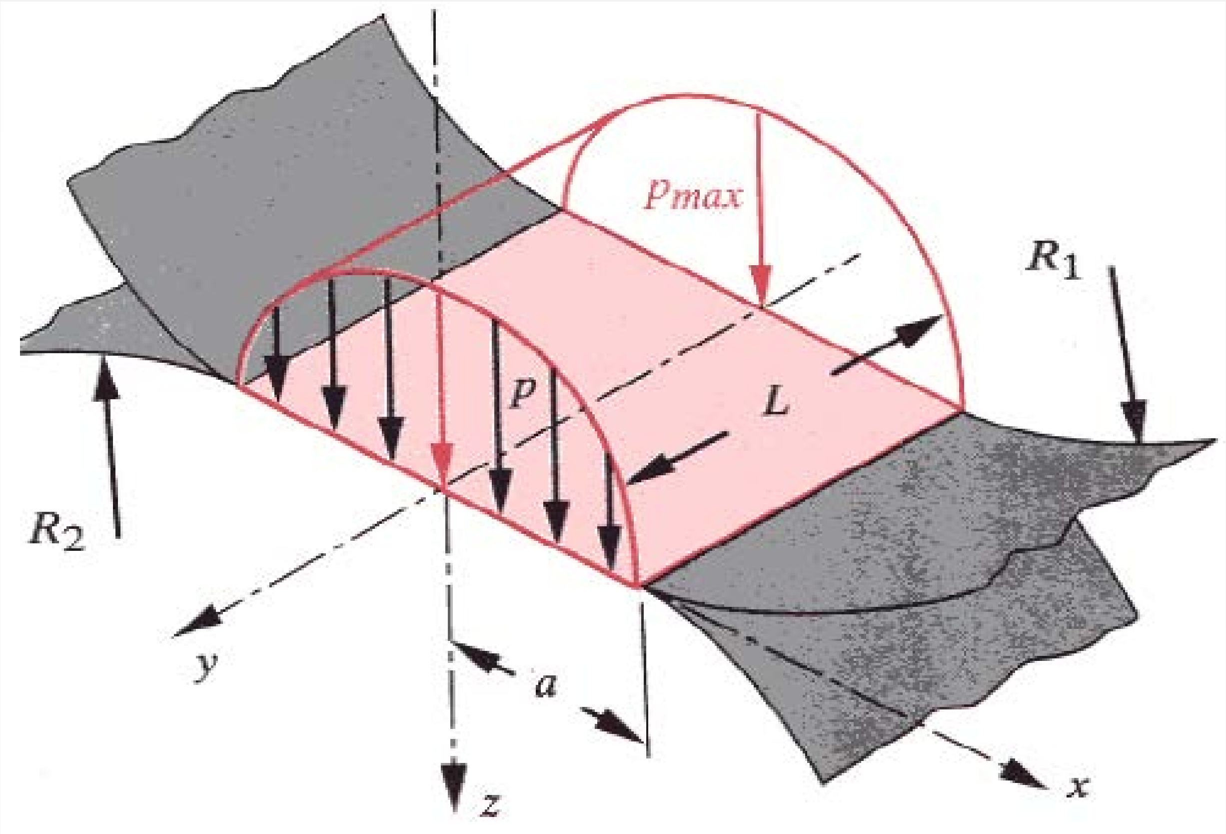
$$C_p = \left[\frac{1}{\pi \left(\frac{1-\nu_p^2}{E_p} + \frac{1-\nu_g^2}{E_g} \right)} \right]^{\frac{1}{2}}$$

Hertz ezilme basıncı

$$p_{max} = -C_p \left[\frac{F}{L} \left(\frac{1}{\rho_1} + \frac{1}{\rho_2} \right) \right]^{1/2}$$

Hertz Contact Stress Equations

Ellipsoidal-prism pressure distribution



Hertz Contact Stress Equations

Contact width,

$$a = \sqrt{\frac{2F}{\pi L}\frac{(1-\upsilon_1^{\ 2})/E_1 + (1-\upsilon_2^{\ 2})/E_2}{1/d_1 + 1/d_2}}$$

d_1 , d_2 represent the pinion and gear pitch diameters.

The maximum contact stress, $P_{max} = 2F / \pi aL$

Total contact force is F,

$$F = 2L\int_0^a p(x)\,dx$$

The maximum surface (Hertz) stress:

$$P_{\max} = \sigma_H = 0.564\sqrt{\frac{F(\frac{1}{R_1}+\frac{1}{R_2})}{\frac{1-\upsilon_1^2}{E_1}+\frac{1-\upsilon_2^2}{E_2}}}$$

Hertz Contact Stress Equations

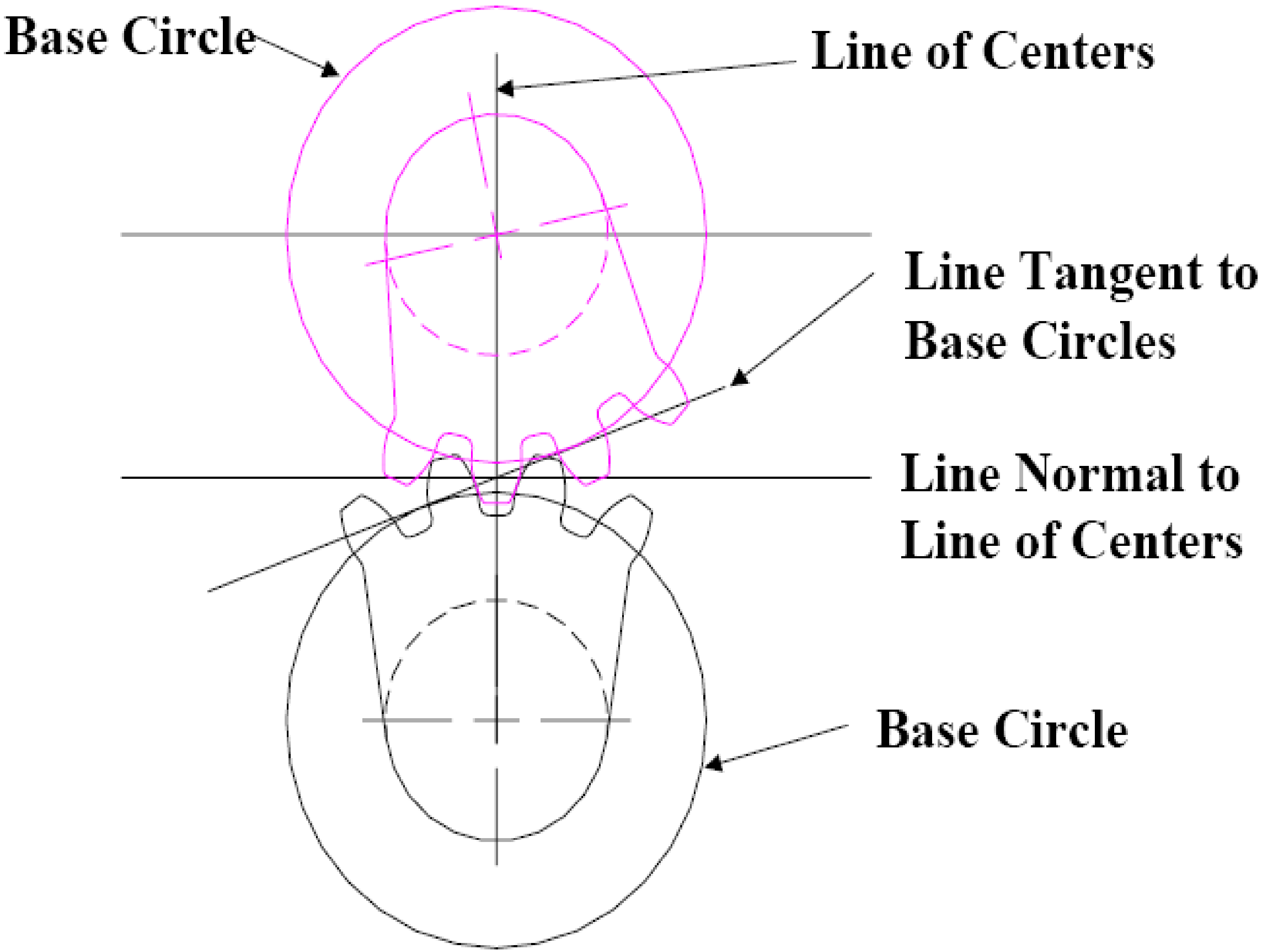
F is the load per unit width

R is the radius of cylinder i, $R_i = d_i \sin\varphi / 2$ for the gear teeth

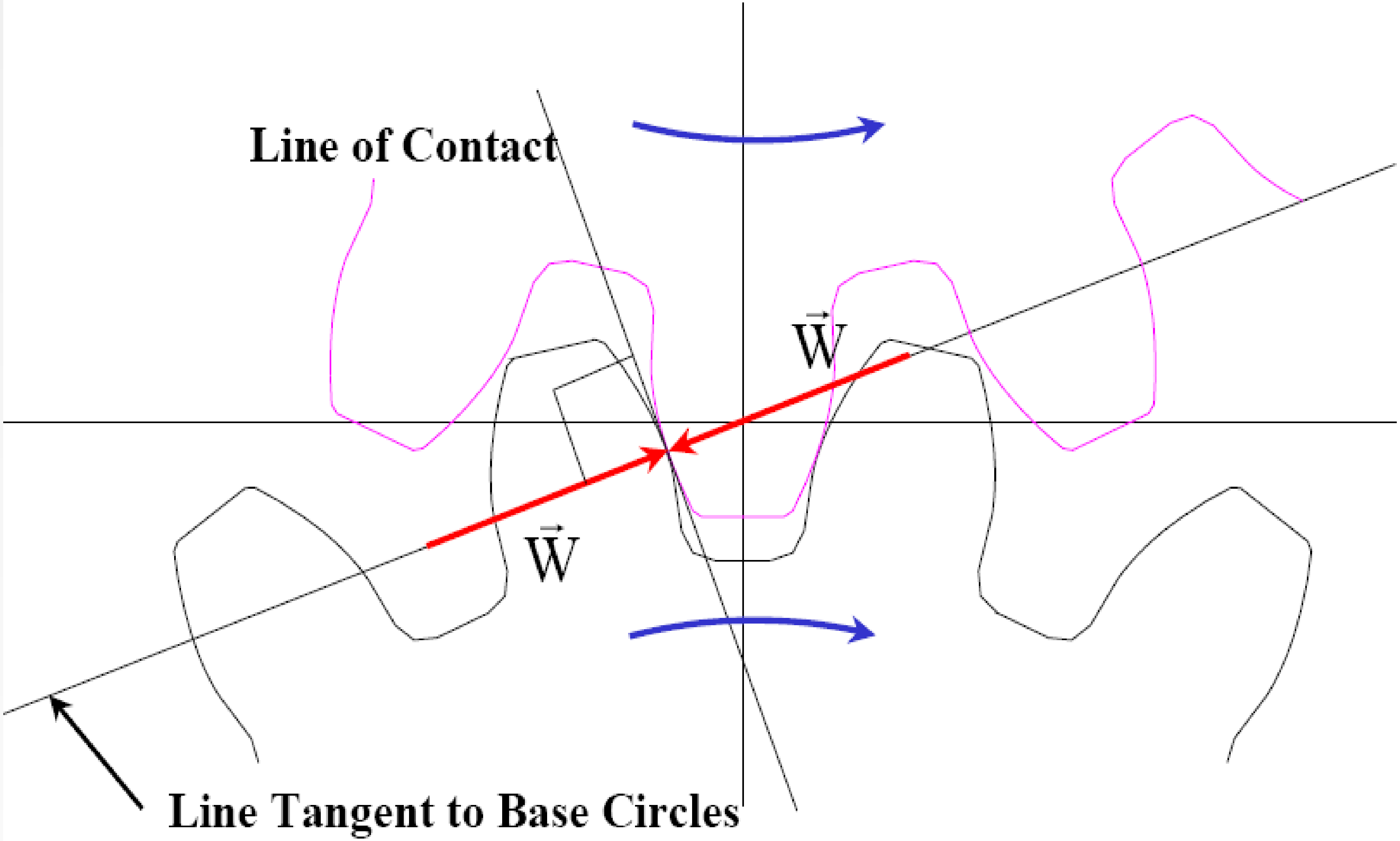
φ is pressure angle, ν_i is Poisson's ratio for cylinder i

E_i is Young's modulus for cylinder i

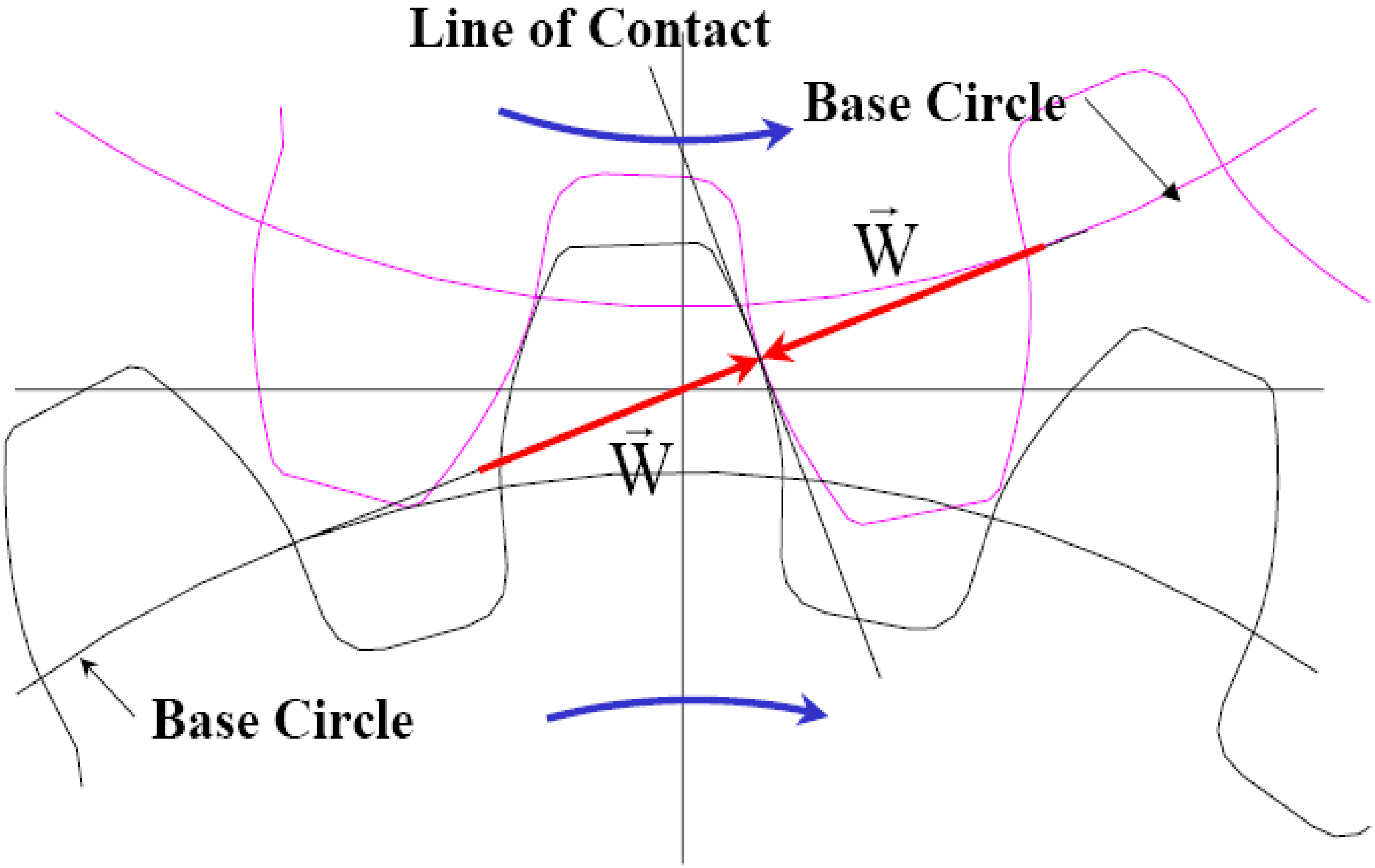
Gear Interaction



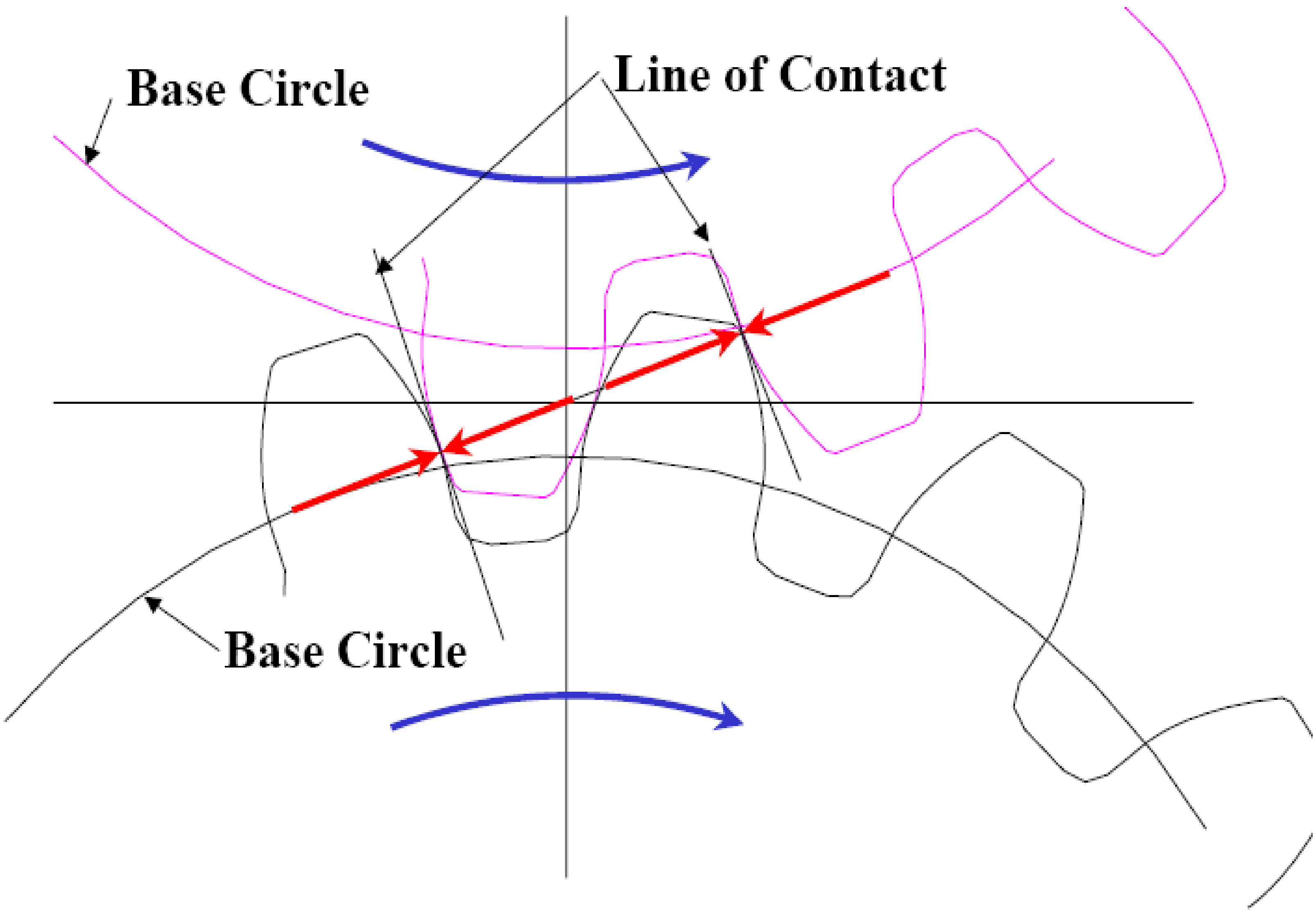
1st Close Up of Meshed Teeth



2^{nd} Close Up of Meshed Teeth



3rd Close Up of Meshed Teeth

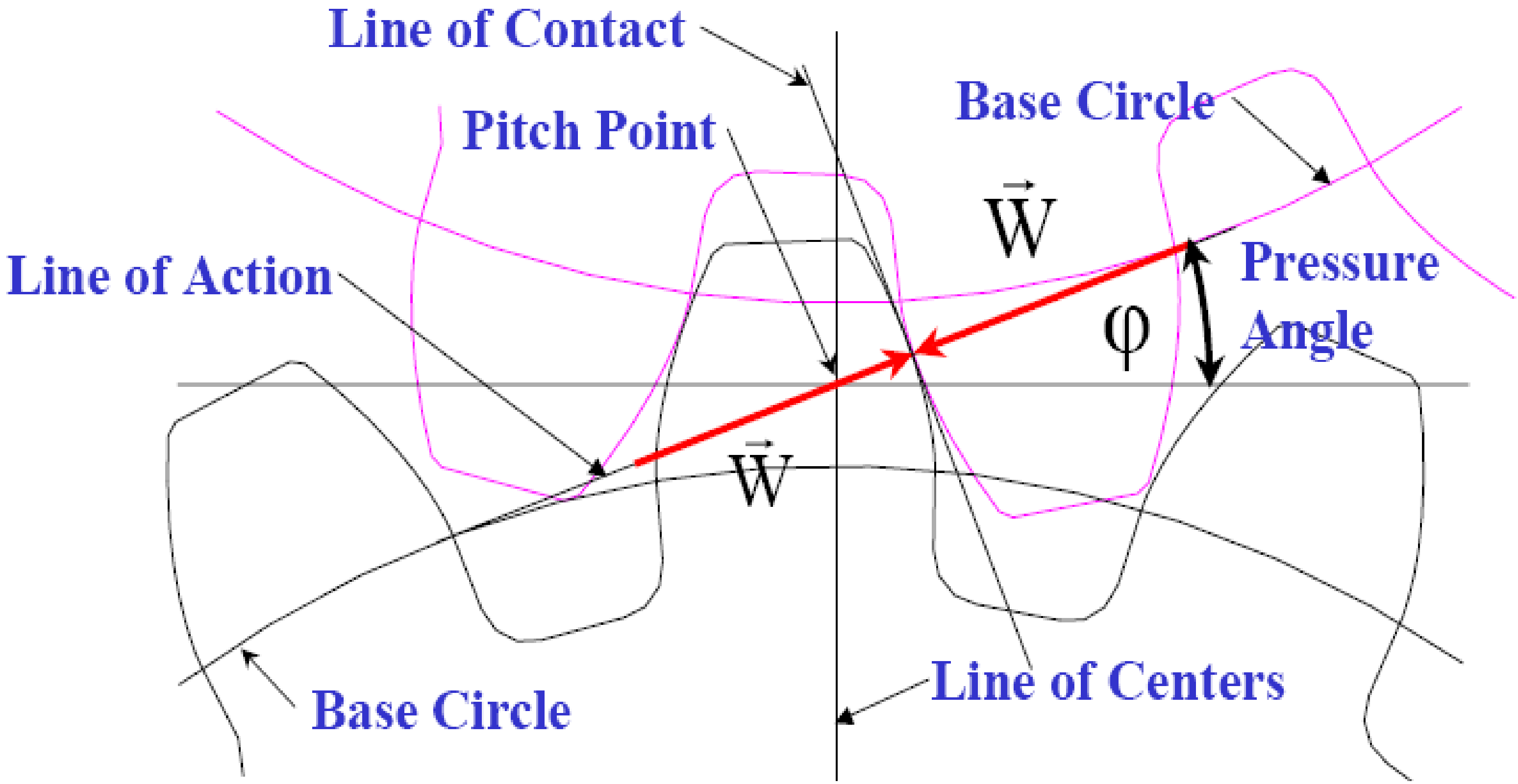


Line of Action/Pressure Angle

Line of Action – Line tangent to both base circles

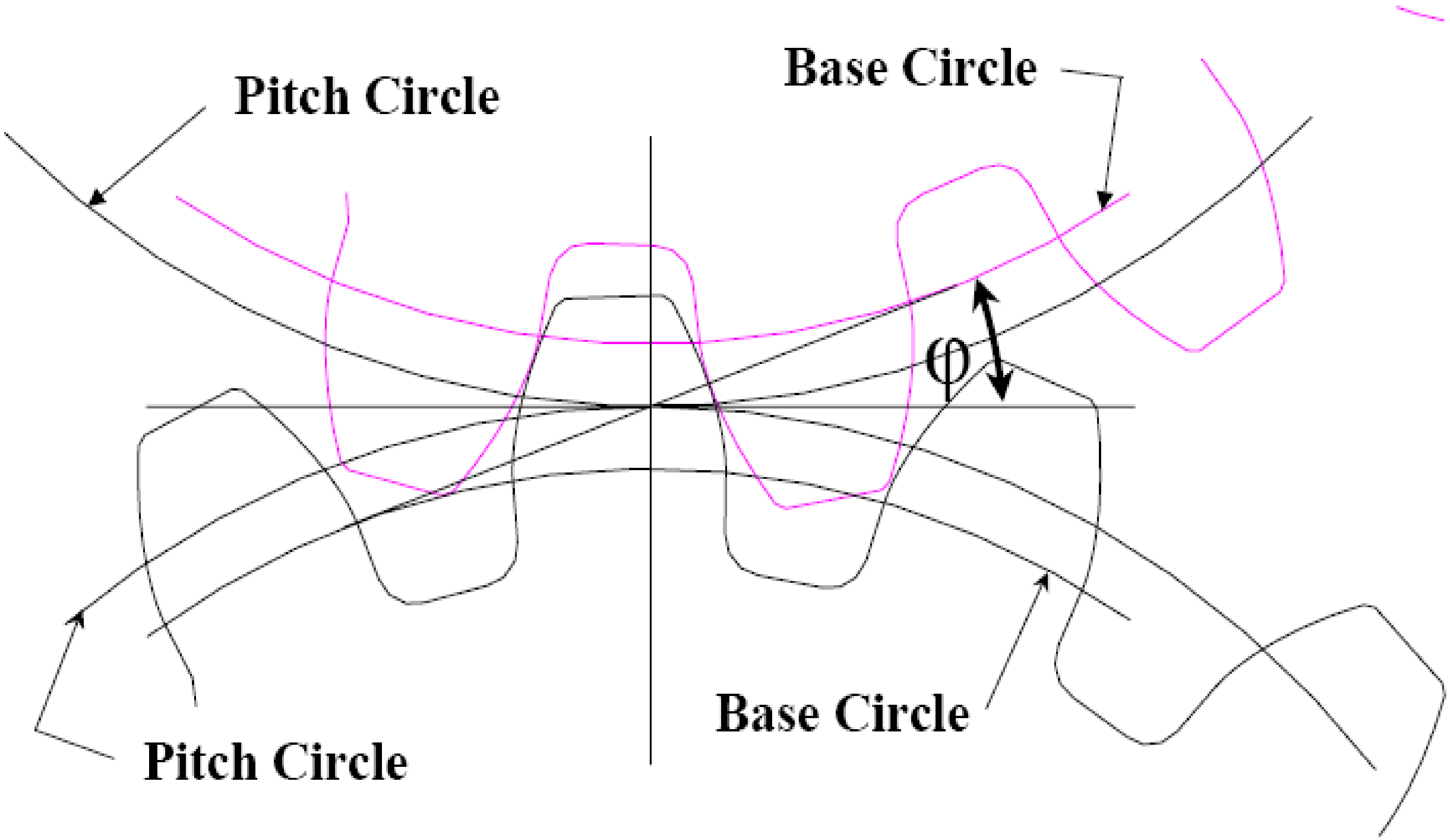
Pressure Angle – Angle between the line normal to the line of centers and the line of action.

Pitch Point – Intersection of the line of centers with the line of action



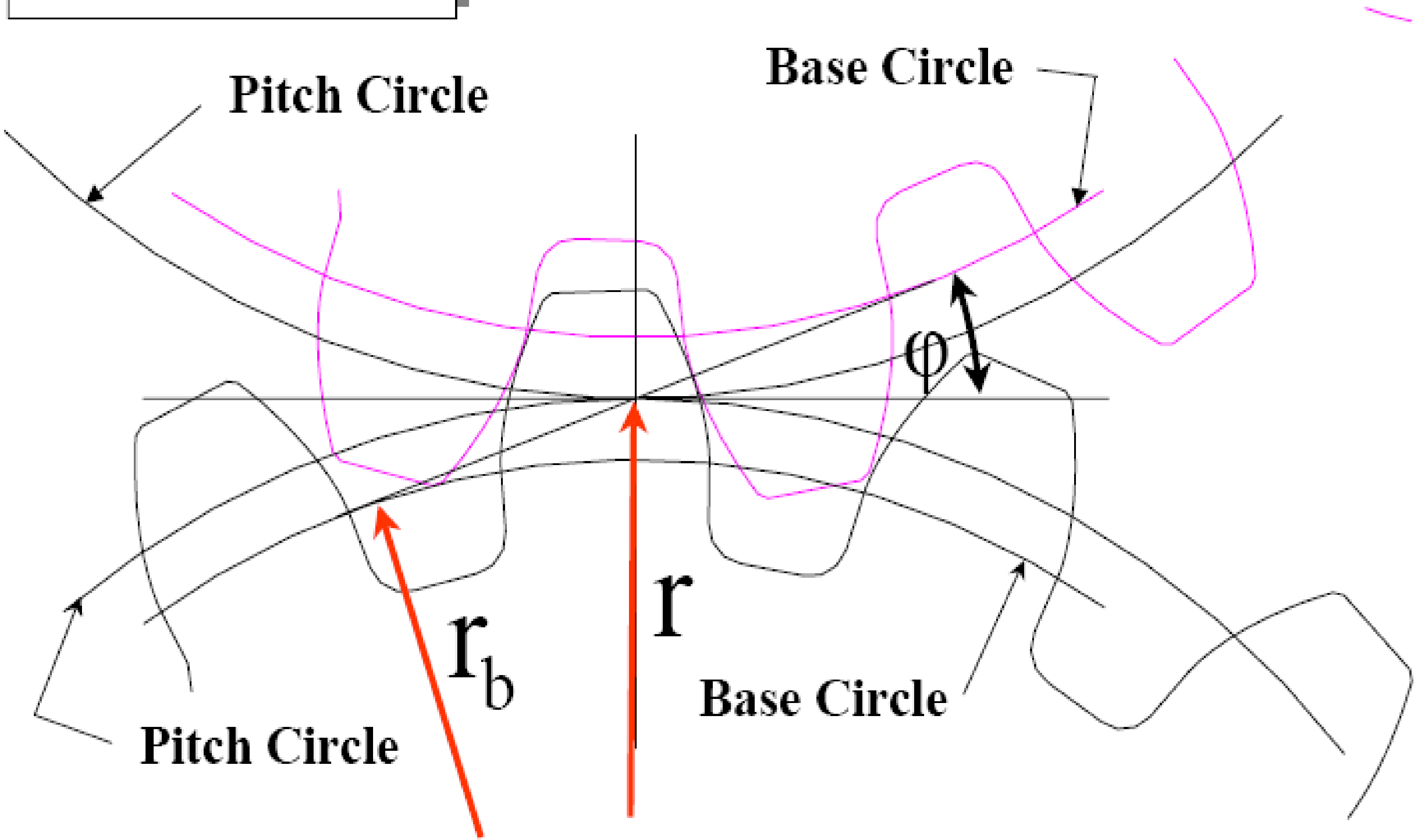
Pitch Circle

Pitch Circle – Circle with origin at the gear center and passing through the pitch point.



Relationship Between Pitch and Base Circles

$$r_b = r\cos(\varphi)$$



Torque Relationship

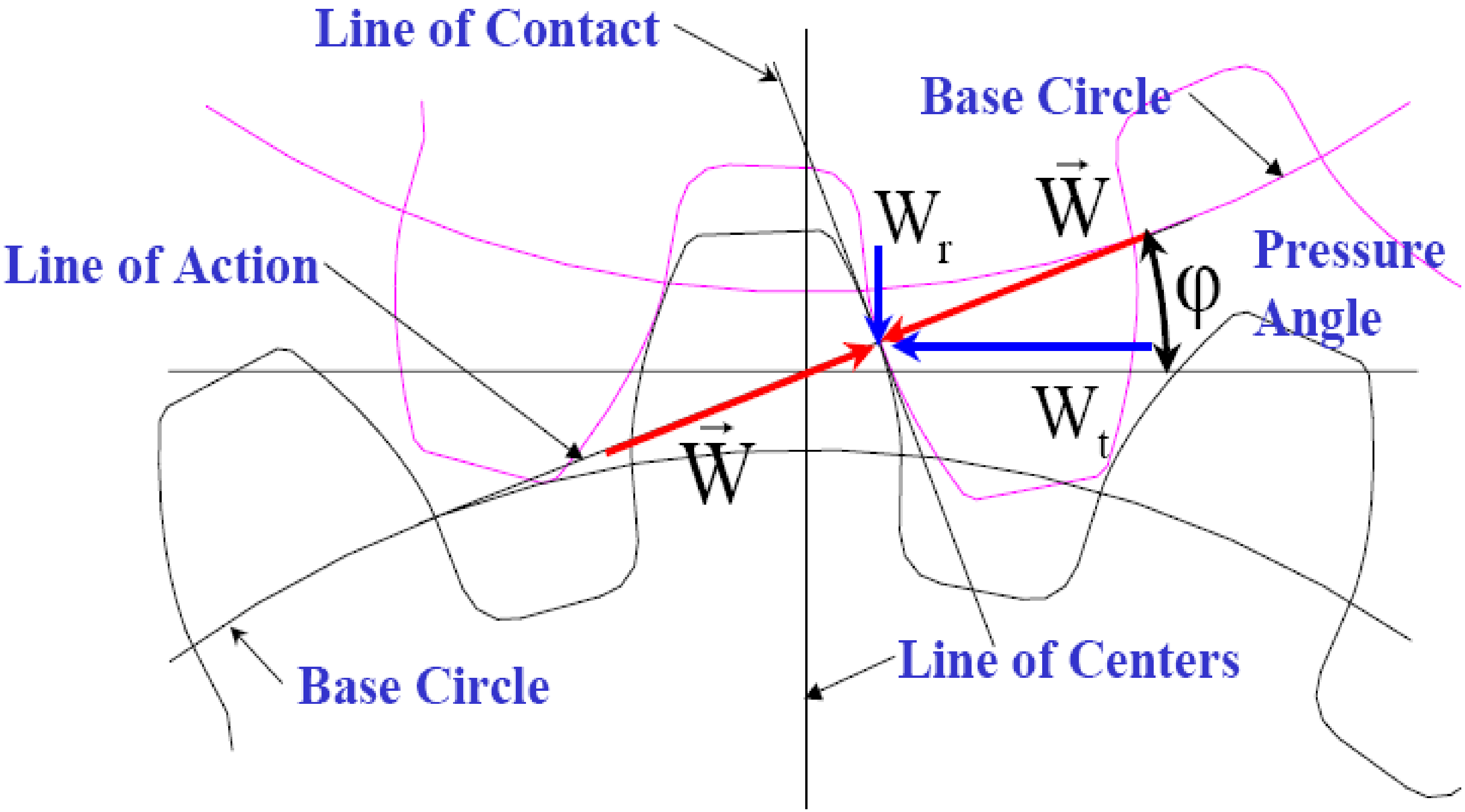
$$T \equiv \frac{\text{Power}}{\text{Angular Velocity}} = \frac{P}{\omega}$$

$$T = \frac{P(\text{hp})}{n\ (\text{rev/min})} \cdot \frac{550\ \text{lb} \cdot \text{ft/sec}}{1.0\ \text{hp}} \cdot \frac{1.0\ \text{rev}}{2\pi\ \text{rad}} \cdot \frac{60\ \text{sec}}{\text{min}} \cdot \frac{12\ \text{in}}{\text{ft}}$$

$$T = 63{,}000 \frac{P}{n} (\text{lb} \cdot \text{in})$$

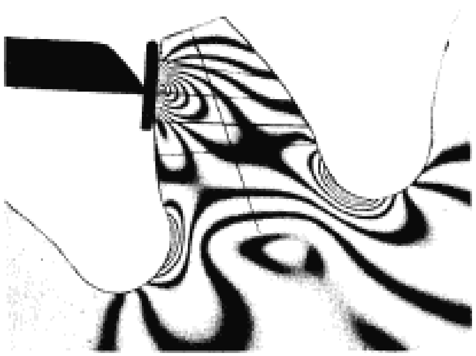
Tooth Load Equations

$$W_t = \frac{T}{d/2} \qquad W_r = W_t \cdot \tan\varphi \qquad \left|\vec{W}\right| = W_t / \cos\varphi$$

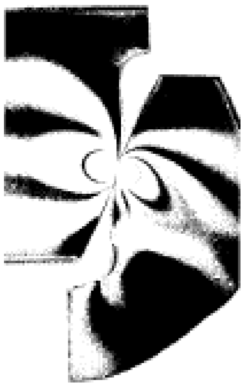


Gear Tooth Failure Mechanisms

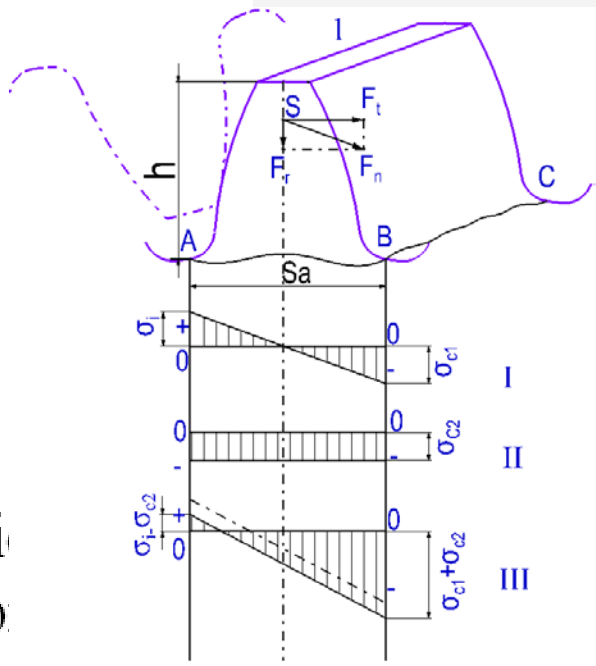
The primary failure mechanisms for involute gear teeth are: 1) excessive bending stresses at the base of the tooth and, 2) excessive bearing or contact stress at the point of contact.



Deutschman, Fig. 10-20

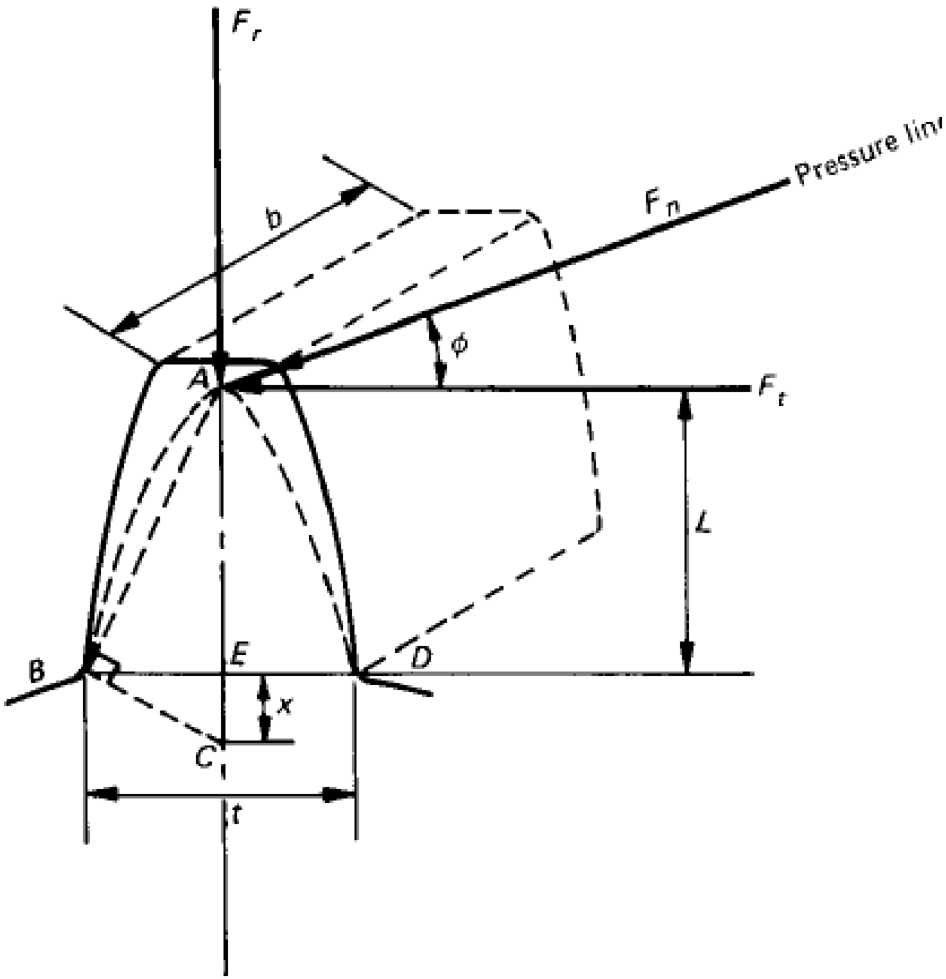


Mott, Fig. 9-14



The American Gear Manufacturers Associati
(AGMA) has developed standard methods fo
addressing both failure mechanisms.

Lewis Equation



$$\sigma = \frac{M}{I/c}$$

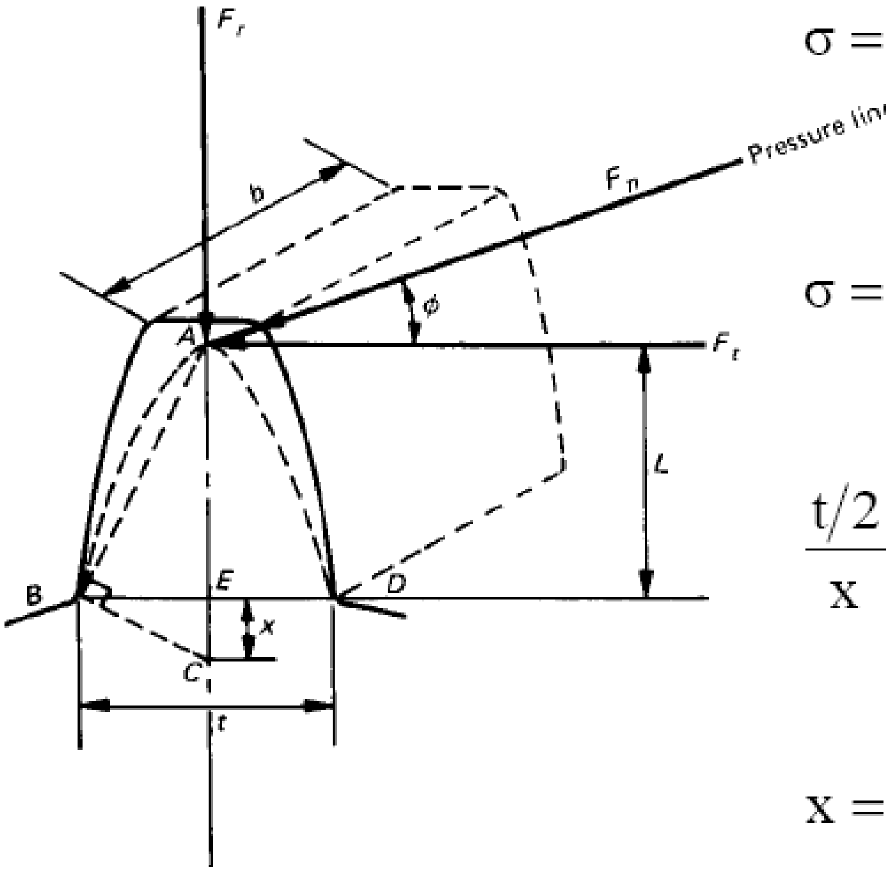
$$M = W_t \cdot L$$

$$I/c = \frac{1}{12} bt^3 \Big/ \tfrac{t}{2} = \frac{bt^2}{6}$$

$$\sigma = \frac{6 W_t L}{bt^2}$$

Deutschman, Fig. 10-18

Lewis Equation
(Continued)



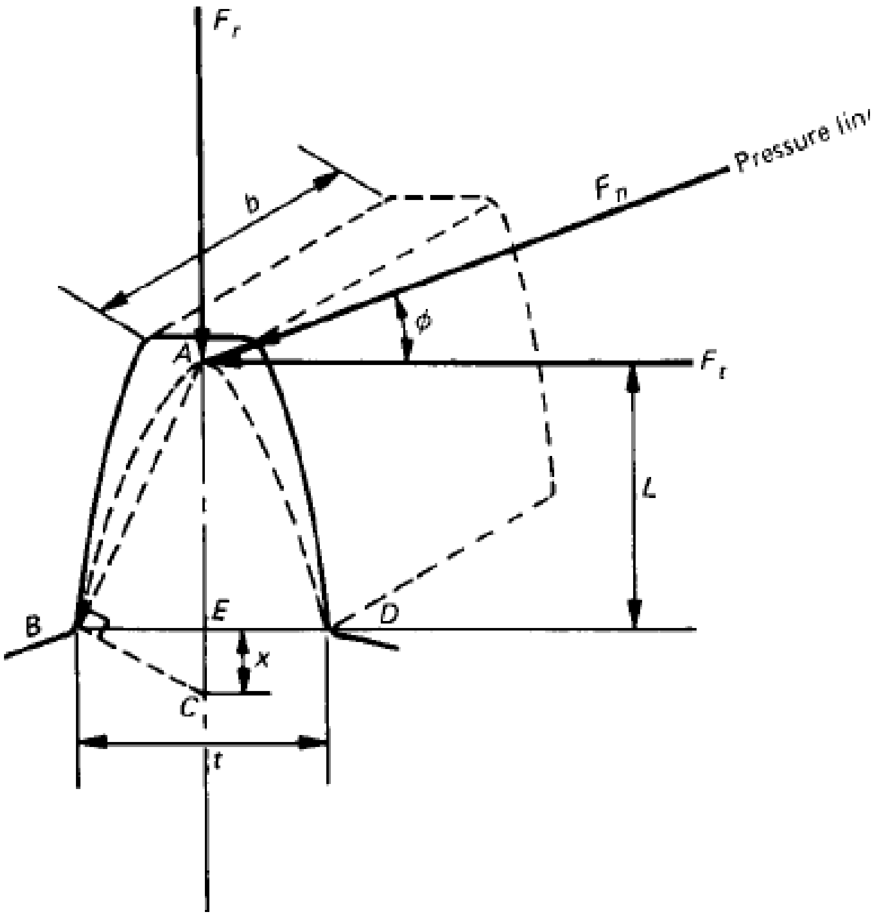
$$\sigma = \frac{6W_t L}{bt^2}$$

$$\sigma = \frac{W_t}{b}\frac{1}{t^2/6L} = \left(\frac{W_t}{b}\right)\left(\frac{1}{t^2/4L}\right)\left(\frac{1}{4/6}\right)$$

$$\frac{t/2}{x} = \frac{L}{t/2}$$

$$x = \frac{t^2}{4L}$$

Lewis Equation
(Continued)



$$\sigma = \left(\frac{W_t}{b}\right)\left(\frac{1}{t^2/4L}\right)\left(\frac{1}{4/6}\right)$$

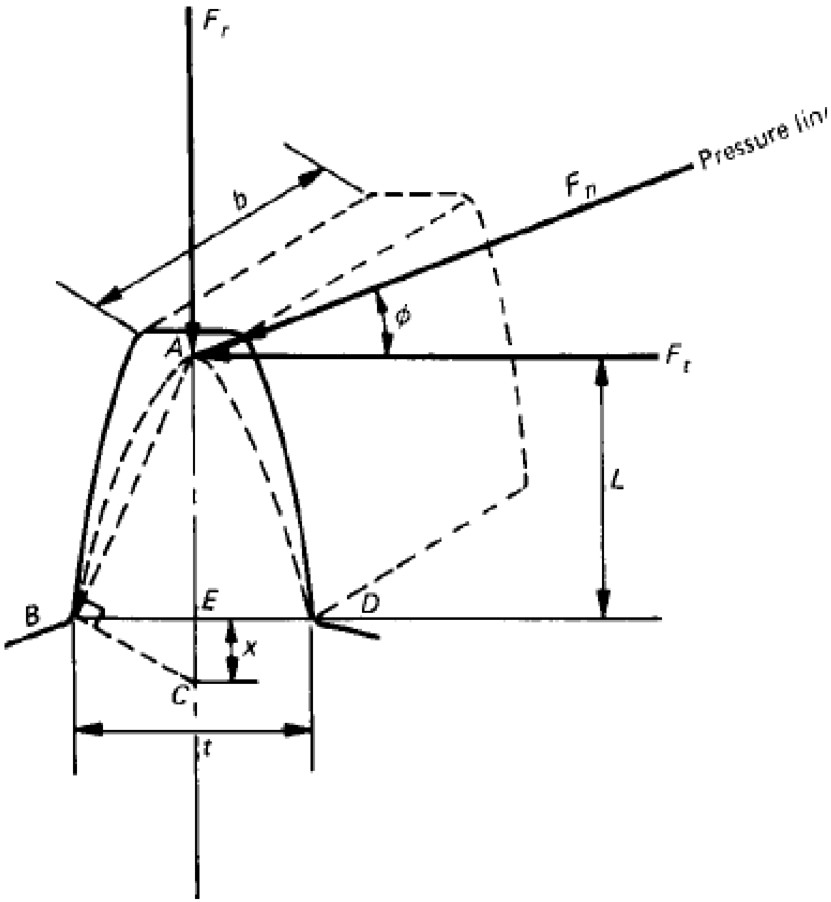
$$x = \frac{t^2}{4L} \qquad p \equiv \text{circular pitch}$$

$$\sigma = \left(\frac{W_t}{b}\right)\left(\frac{1}{x}\right)\left(\frac{1}{2/3}\right)\left(\frac{p}{p}\right)$$

$$y = \frac{2x}{3p}$$ Lewis Form Factor

$$\boxed{\sigma = \frac{W_t}{bpy}}$$

Lewis Equation
(Continued)



$$y = \frac{2x}{3p}$$

$$\sigma = \frac{W_t}{bpy}$$

$P \equiv$ Diametral Pitch $= \pi/p$

$Y \equiv \pi y$

$$\boxed{\sigma = \frac{W_t P}{bY}}$$

Most commonly used form of Lewis Equation

Y can be determined graphically or by a computer.

Lewis Form Factor

(Example Values)

NUMBER OF TEETH	Y	NUMBER OF TEETH	Y
12	0.245	28	0.353
13	0.261	30	0.359
14	0.277	34	0.371
15	0.290	38	0.384
16	0.296	43	0.397
17	0.303	50	0.409
18	0.309	60	0.422
19	0.314	75	0.435
20	0.322	100	0.447
21	0.328	150	0.460
22	0.331	300	0.472
24	0.337	400	0.480
26	0.346	Rack	0.485

Values are for a normal pressure angle of 20 degrees, full-depth teeth, and a diametral pitch of one.

$$\sigma = \frac{W_t P}{bY}$$

Shigley, Table 14-2

Limitations of the Lewis Equation

1. Assumes that maximum bending load occurs at the tip. Maximum load occurs near the pitch circle when one tooth carries all of the torque induced load.
2. Considers only bending component of the force acting on the tooth. The radial force will cause a compressive stress over the base cross section.
3. Doesn’t consider contact stresses.
4. Assumes that the loads are static.

The AGMA has developed a number of factors to be used with the Lewis Equation that will lead to an acceptable design.

The AGMA Equations

$$\sigma = \frac{W_t P_d}{FJ} \cdot K_a \cdot K_s \cdot K_m \cdot K_v$$

$$\sigma_{all} = \frac{S_{at} K_L}{K_T K_R}$$

$F \equiv$ face width (b)

$K_a \equiv$ Application factor

$K_s \equiv$ Size factor

$K_m \equiv$ Load distribution factor

$K_v \equiv$ Dynamic factor

$J \equiv$ Geometry factor

$P_d \equiv$ Diametral Pitch $= N/d$

$d \equiv$ Pitch Diameter

$N \equiv$ Number of Teeth

$W_t \equiv$ Tangential Load

$S_{at} \equiv$ AGMA Allowable Stress Number

$K_L \equiv$ Life factor

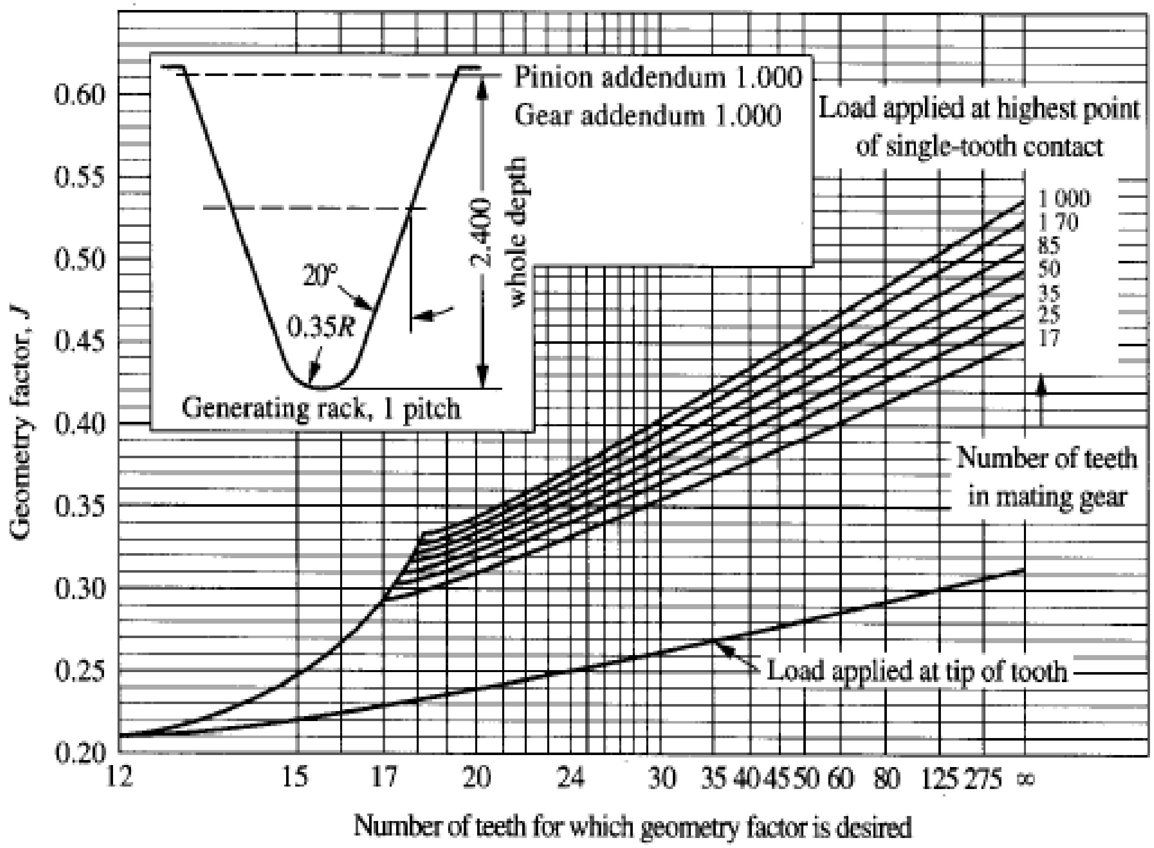
$K_T \equiv$ Temperature Factor

$K_R \equiv$ Reliability Factor

Factors are used to adjust the stress computed by the Lewis equation. Factors are also used to adjust the strength due to various environmental conditions.

Shigley contains tables and charts for many of these factors.

AGMA Form Factor

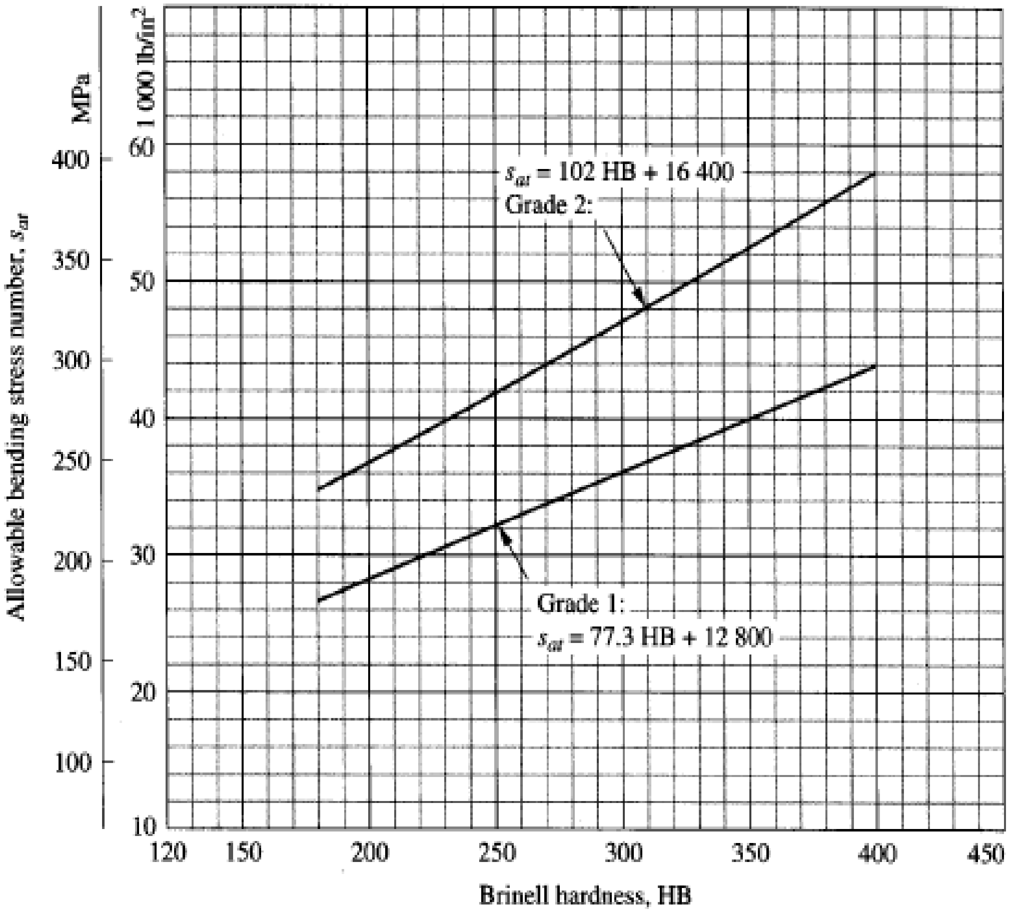


(a) 20° spur gear: standard addendum

Note that the AGMA Form Factor will result in a lower stress than the Lewis Equation.

Mott, Fig. 19-5

AGMA Allowable Bending Stress Numbers



Grade 1 is the basic or standard material classification.

Grade 2 requires better than normal microstructure control.

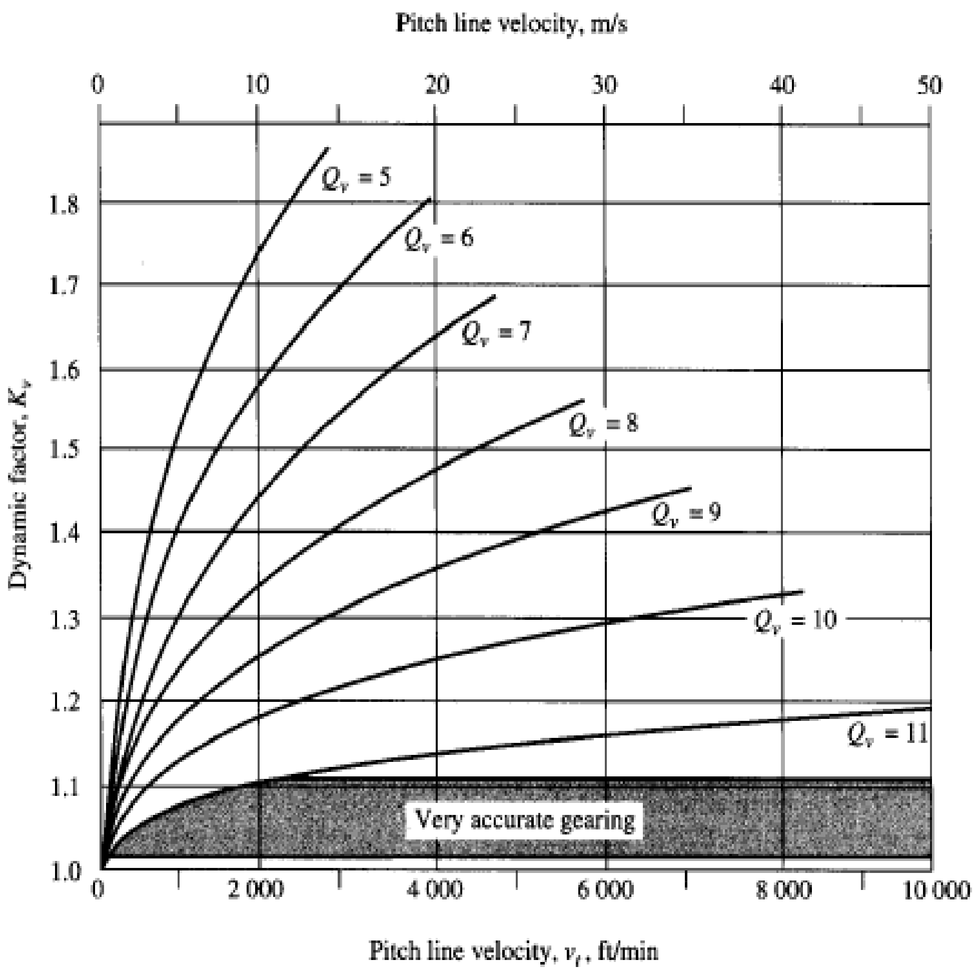
Mott, Fig. 9-8

AGMA Dynamic Factor

The AGMA Dynamic Factor is used to correct the bending stress number for dynamic effects associated with:

1. Inaccuracies in tooth profile, tooth spacing, profile lead, and run-out,
2. Vibration of the tooth during meshing due to tooth stiffness;
3. Magnitude of the pitch-line velocity,
4. Dynamic unbalance of the rotating members,
5. Wear and permanent deformation of contacting surfaces,
6. Shaft misalignment and deflection, and
7. Tooth friction.

Dynamic Factor Chart



$Q_v \equiv$ AGMA Quality Number

The AGMA standards contain tolerances for each quality number.

Pitch Line Velocity = rω

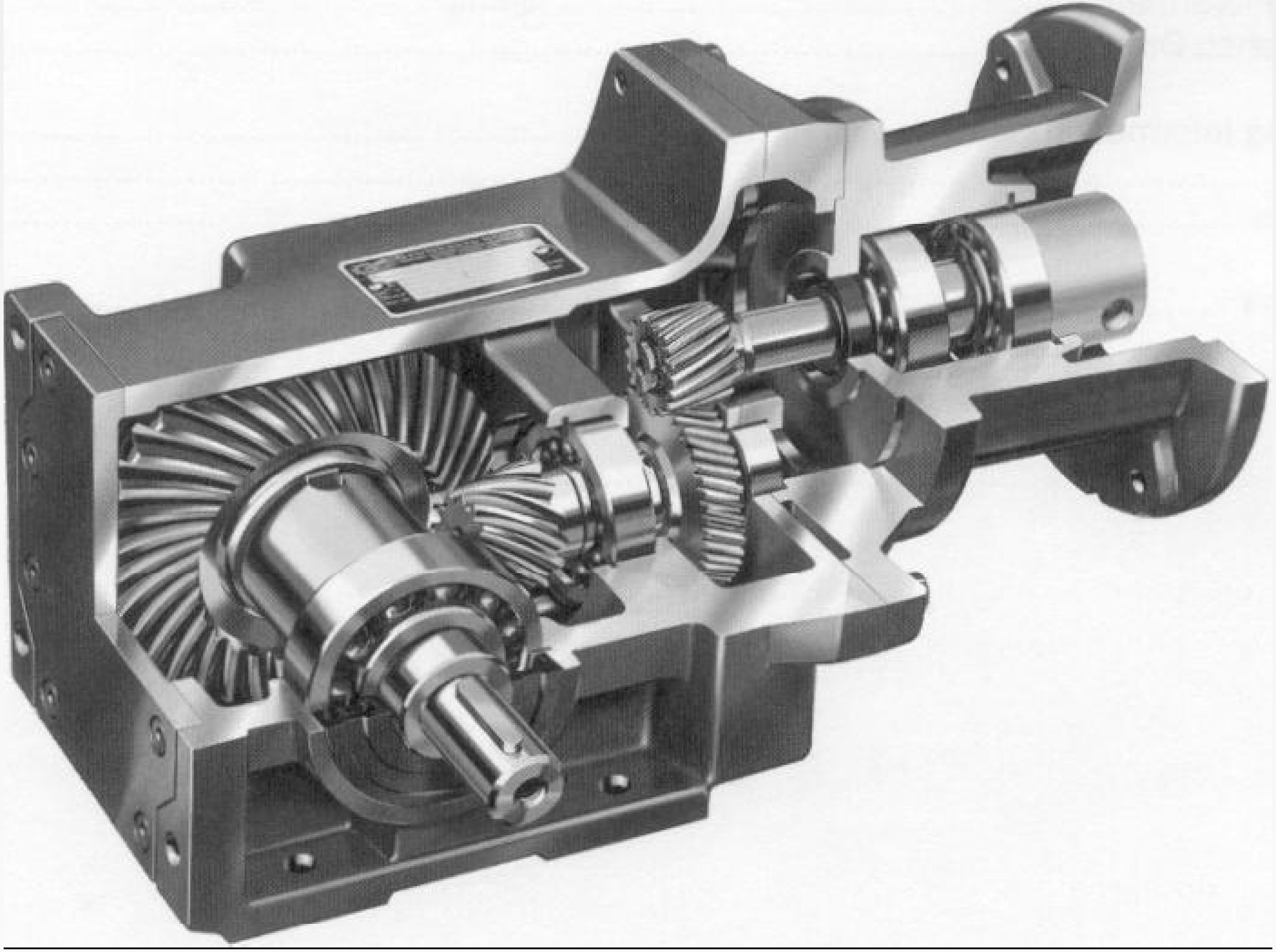
The dynamic factor in Shigley is equal to the reciprocal of the dynamic factor given in this chart.

Mott, Fig. 9-19

Assignment

1. A spur pinion has a pitch of 6 teeth/in, 22 full-depth teeth, and a 20 degree pressure angle. This pinion runs at a speed of 1200 rev/min and transmits 15 hp to a 60-tooth gear. If the face width is 2 in, estimate the bending stress.
2. A steel spur pinion has a module of 1.25 mm, 18 full depth teeth, a pressure angle of 20 degrees, and a face width of 12 mm. At a speed of 1800 rev/min, this pinion is expected to carry a steady load of 0.5 kW. Determine the resulting bending stress.

İki kademeli helisel ve konik dişli kutusu



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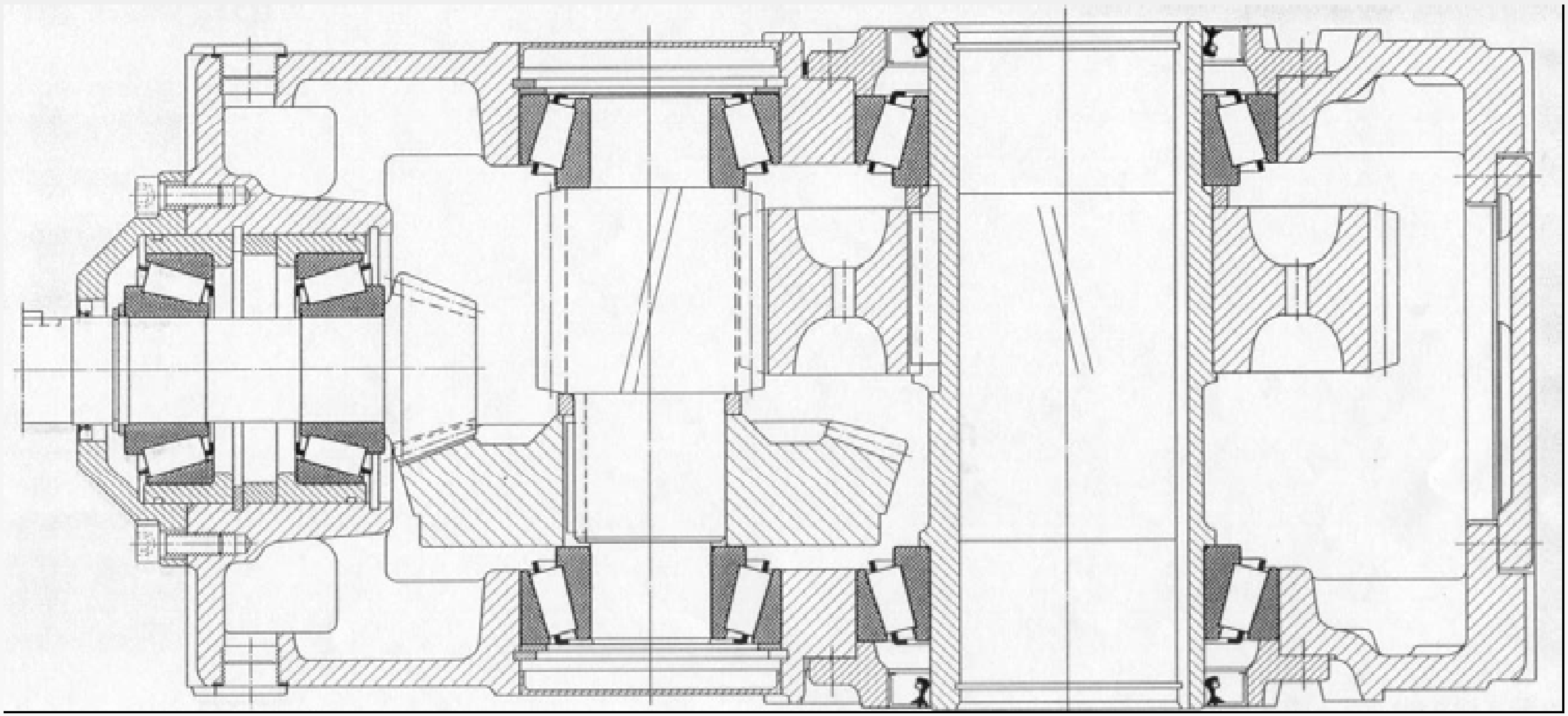

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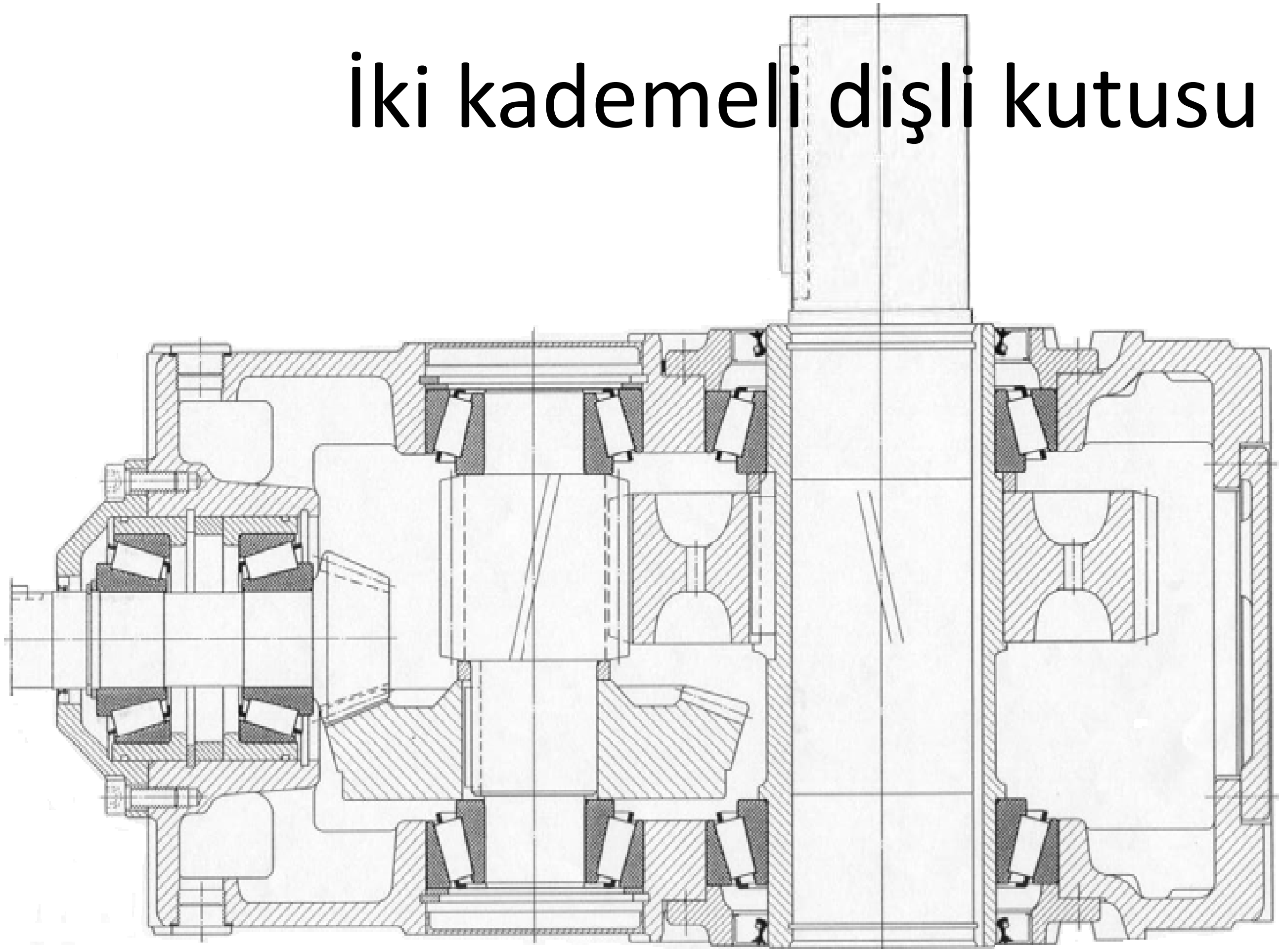

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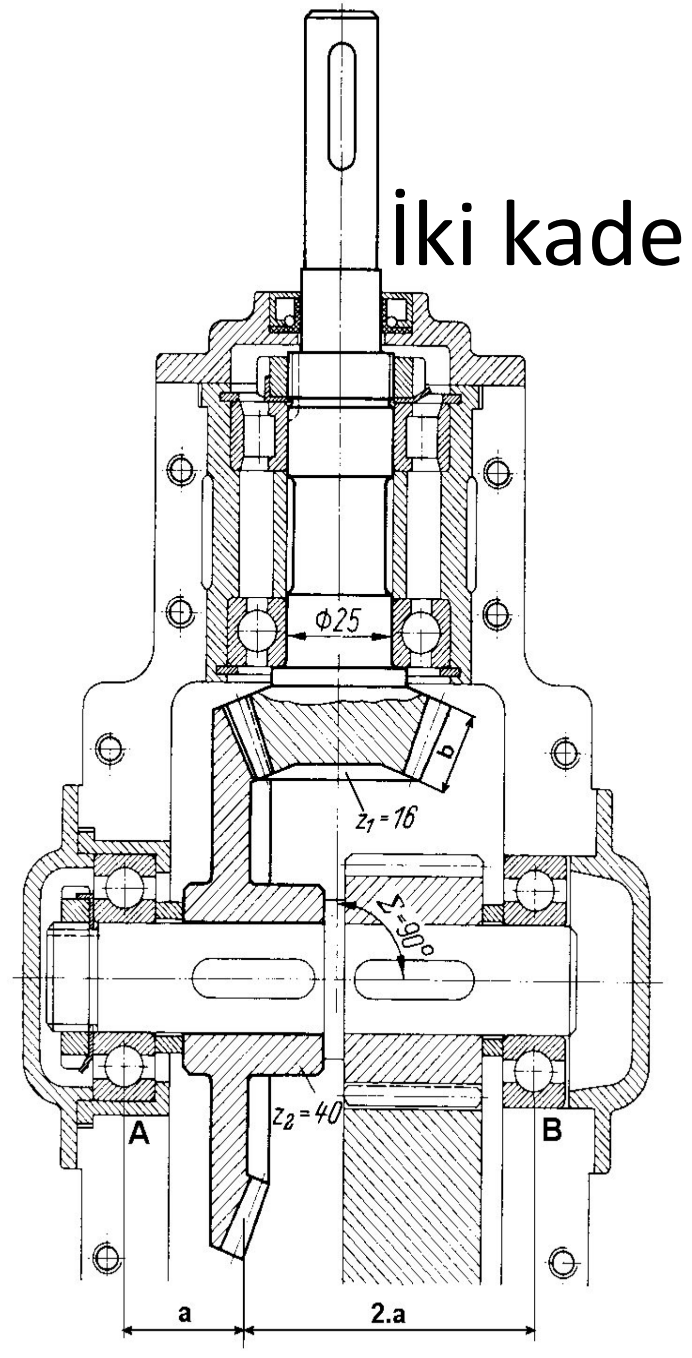

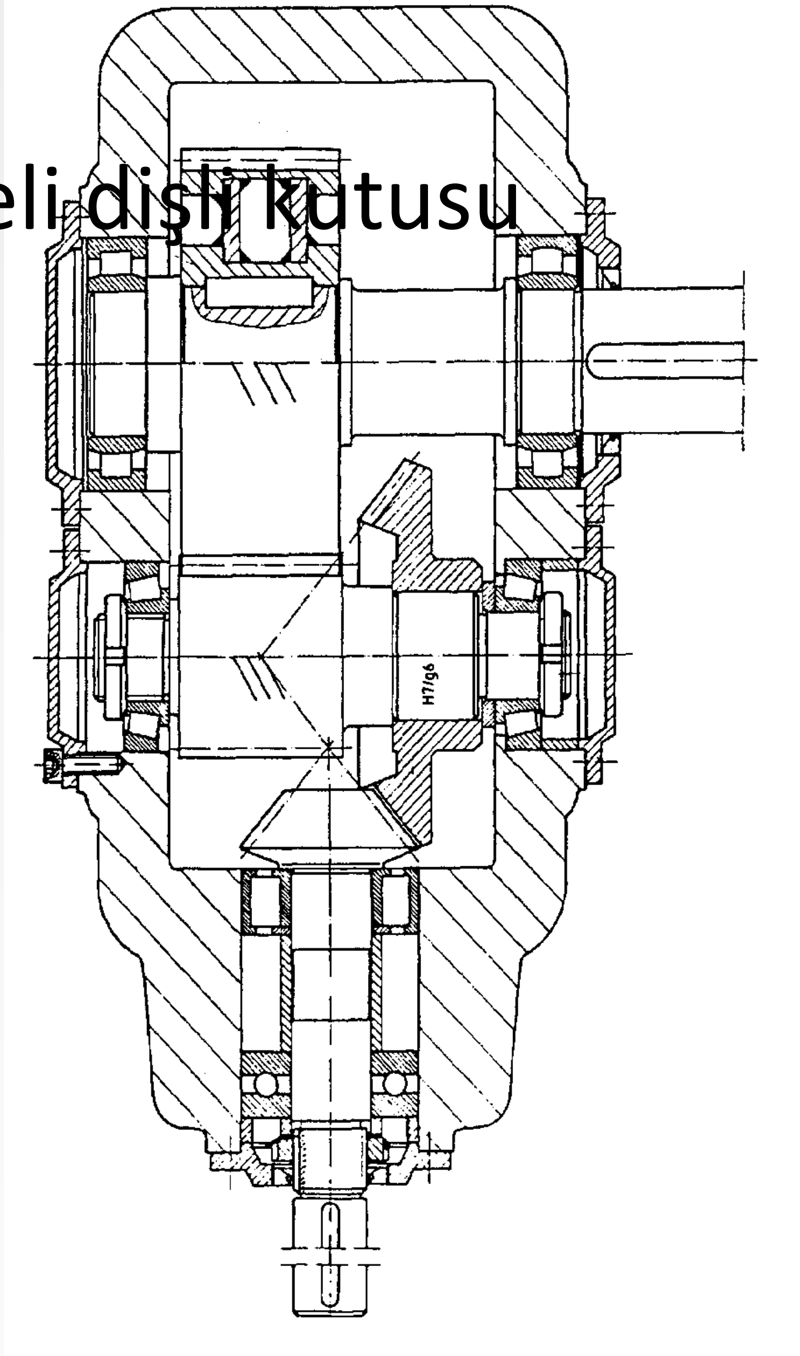

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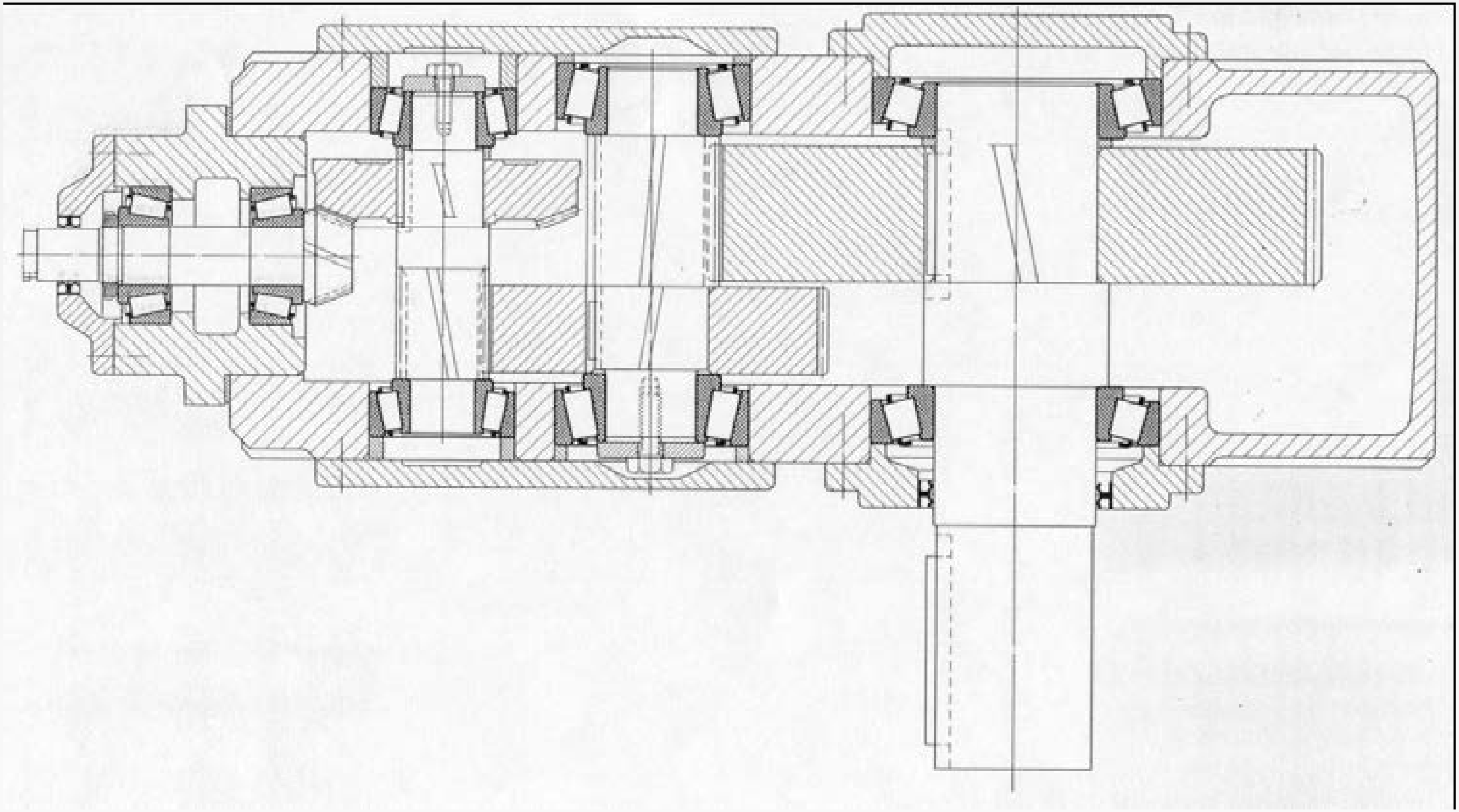

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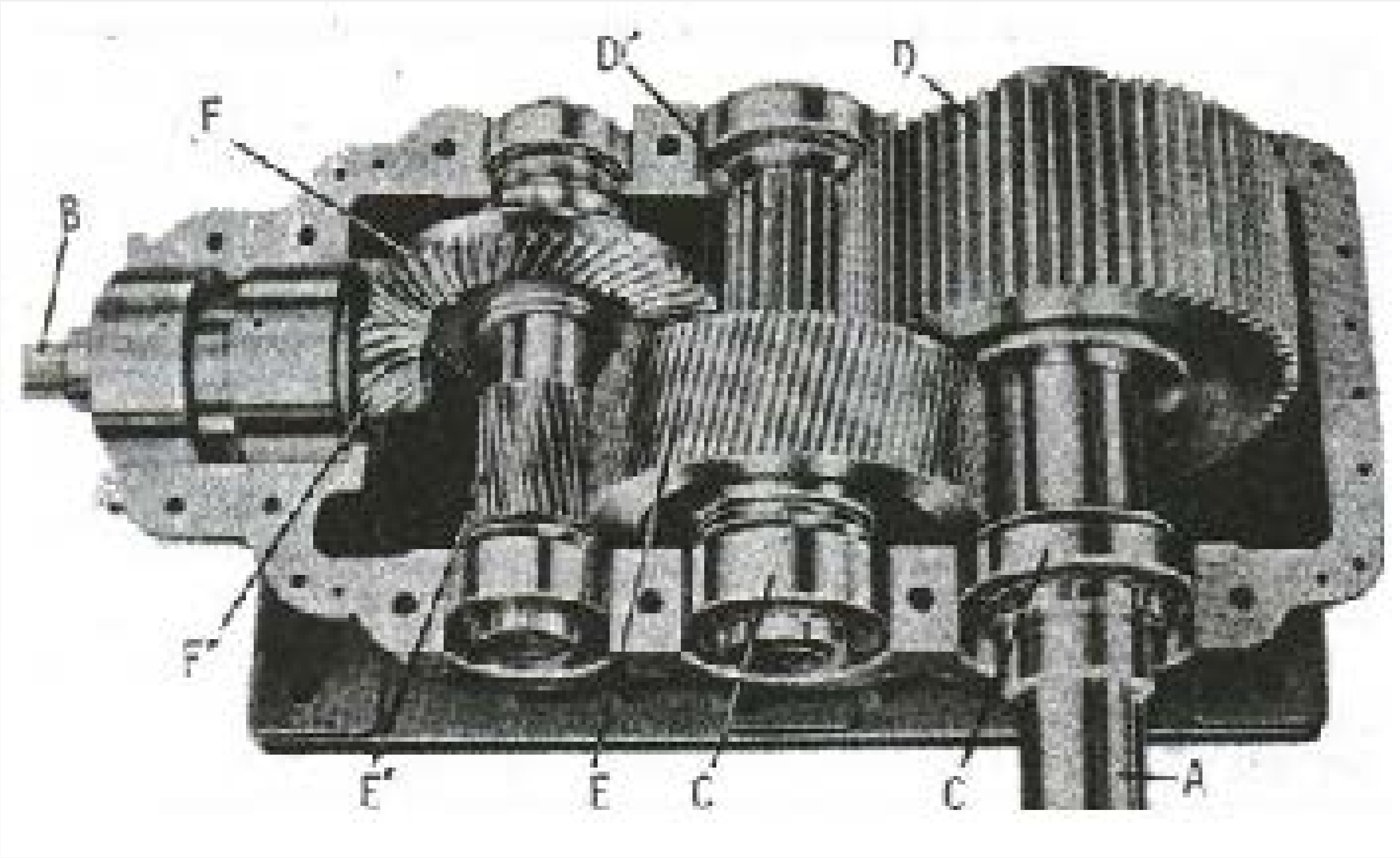
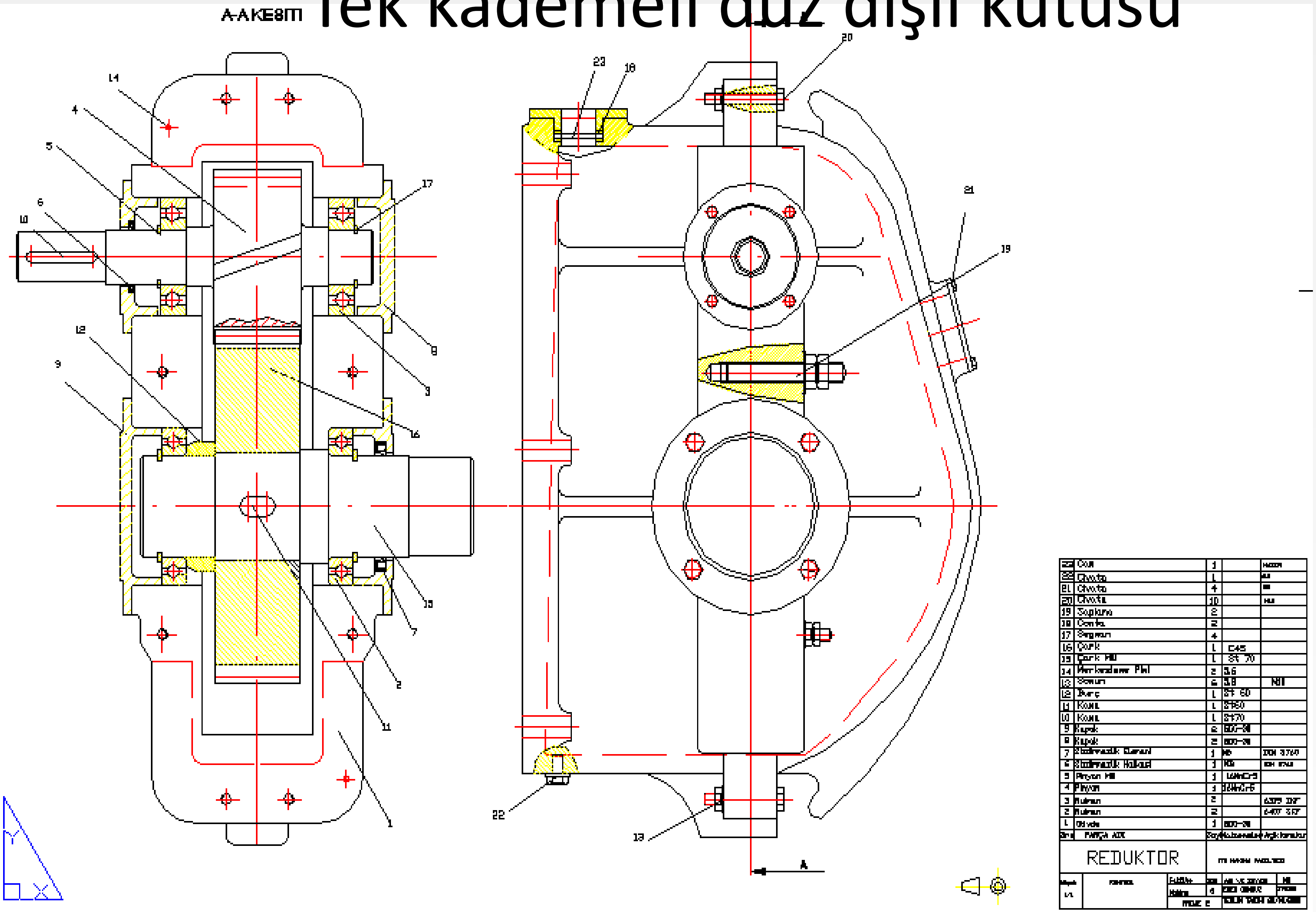


Tek kademeli düz dişli kutusu



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