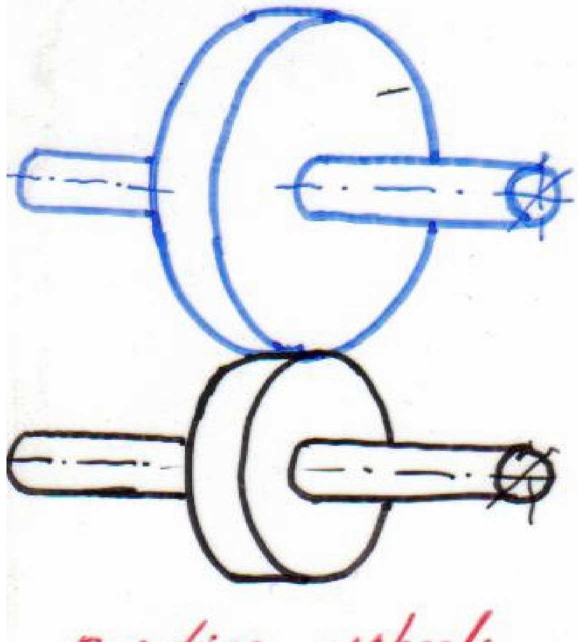
Dişli Çarklar
(Gears)

GEAR DRIVES

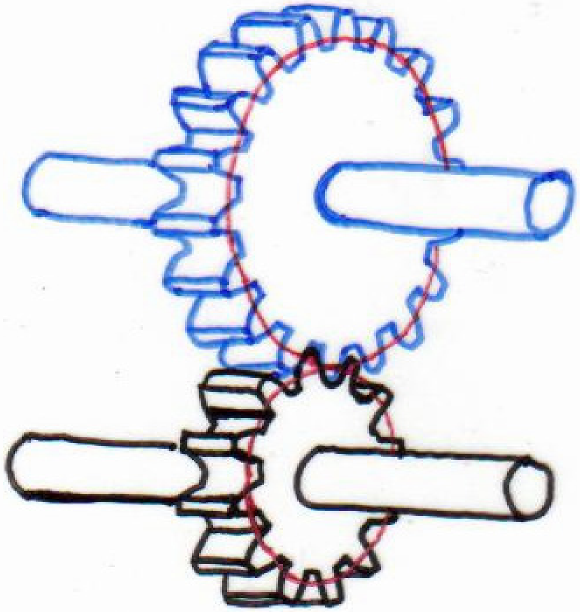
The function of a gear is to transmit motion, rotating or reciprocating, from one machine part to another and where necessary to reduce or increase the revolutions of a shaft. *Gears* are rolling cylinders or cones having teeth on their contact surfaces to ensure positive motion

There are many kinds of gears, and they may be grouped according to the position of the shafts that they connect. **Spur gears** connect parallel shafts, **bevel gears** connect shafts whose axes intersect, and **worm gears** connect shafts whose axes do not intersect. A spur gear with a rack converts rotary motion to reciprocating or linear motion. The smaller of two gears is known as the **pinion**.



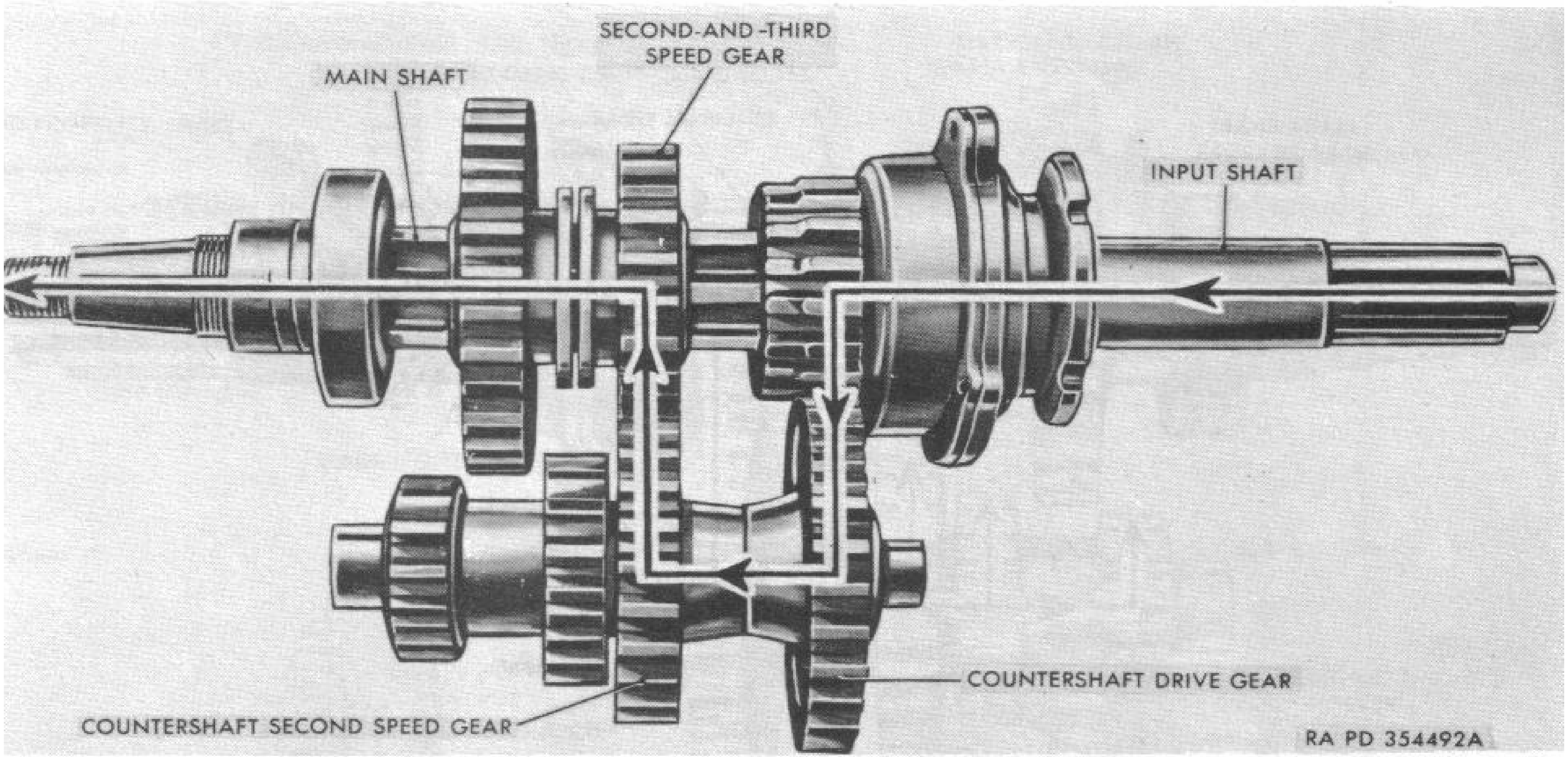
Friction wheels

(a simple means of transmitting rotary motion from one shaft to another)

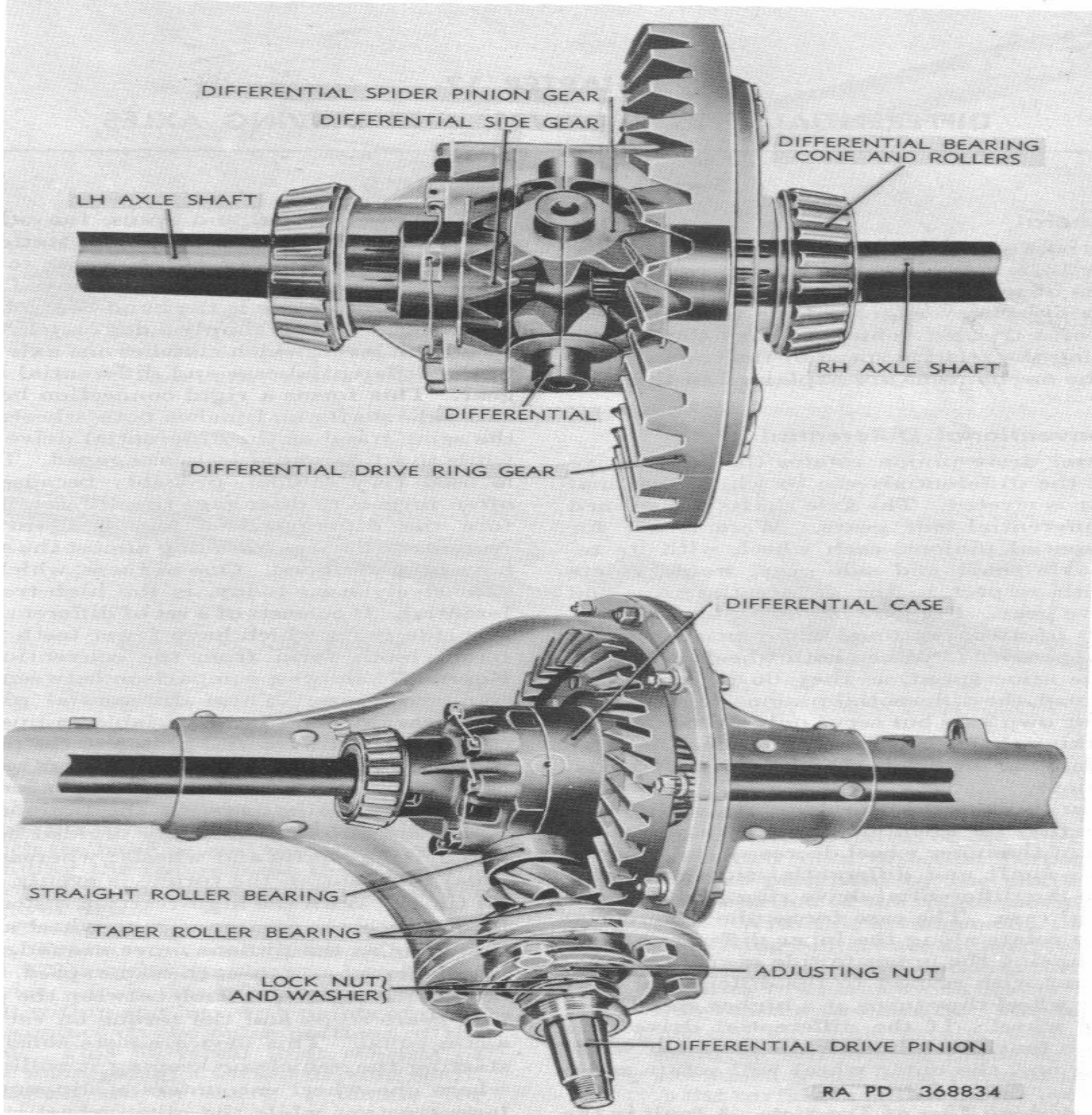


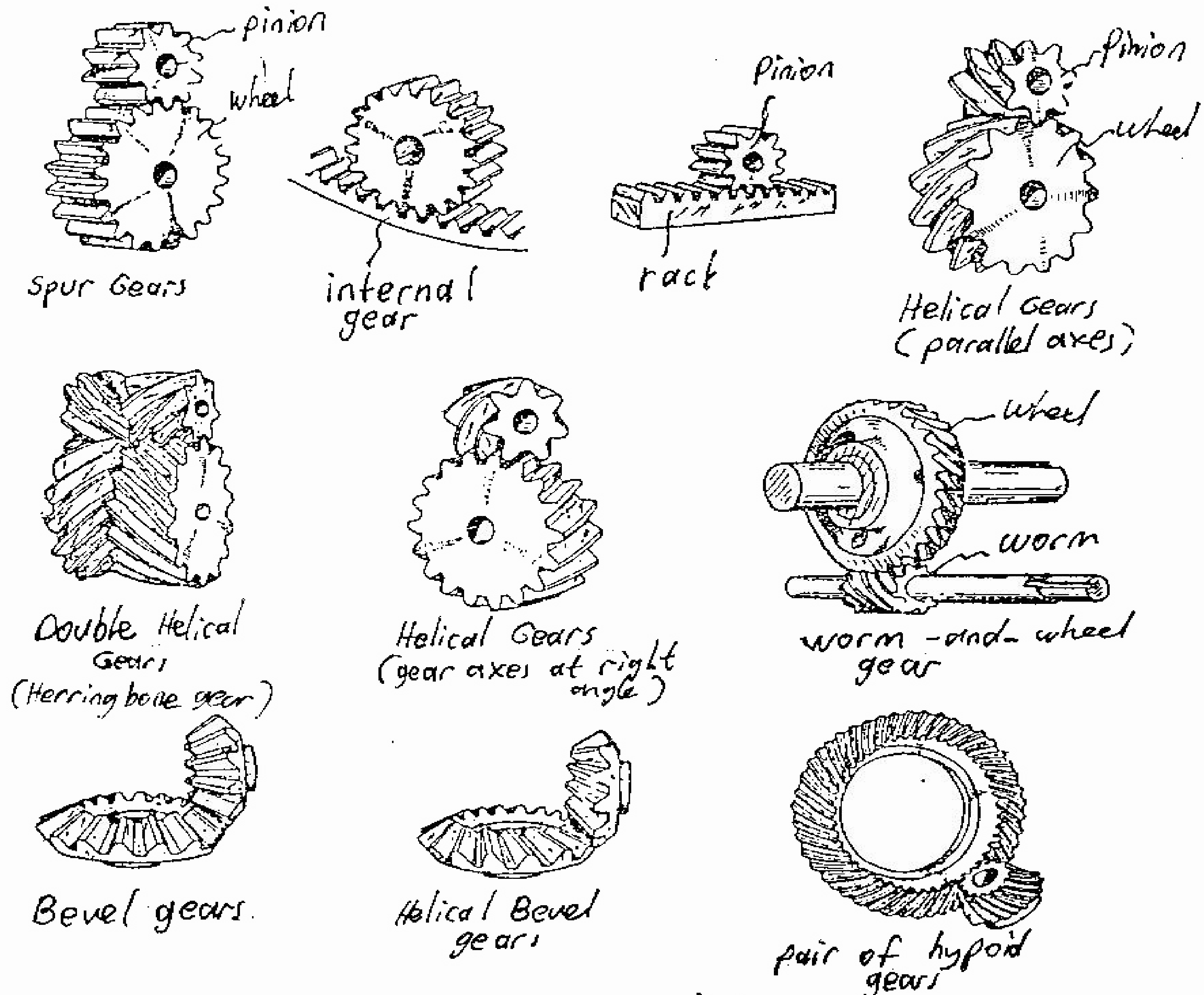
Spur Gears

(Teeth added to friction wheels provide a more efficient means of transmitting rotary motion)



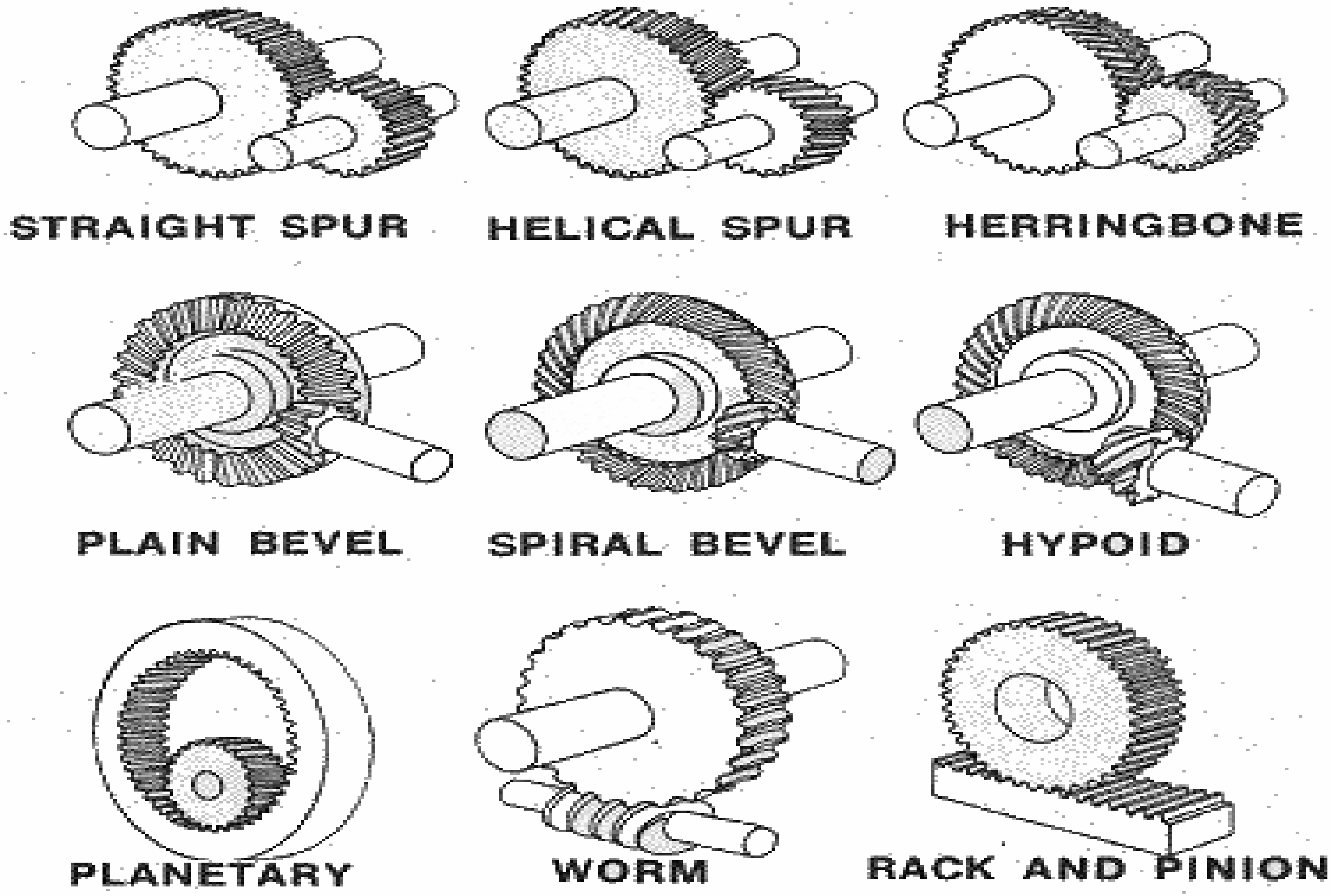
Transmission gears in second-speed position.

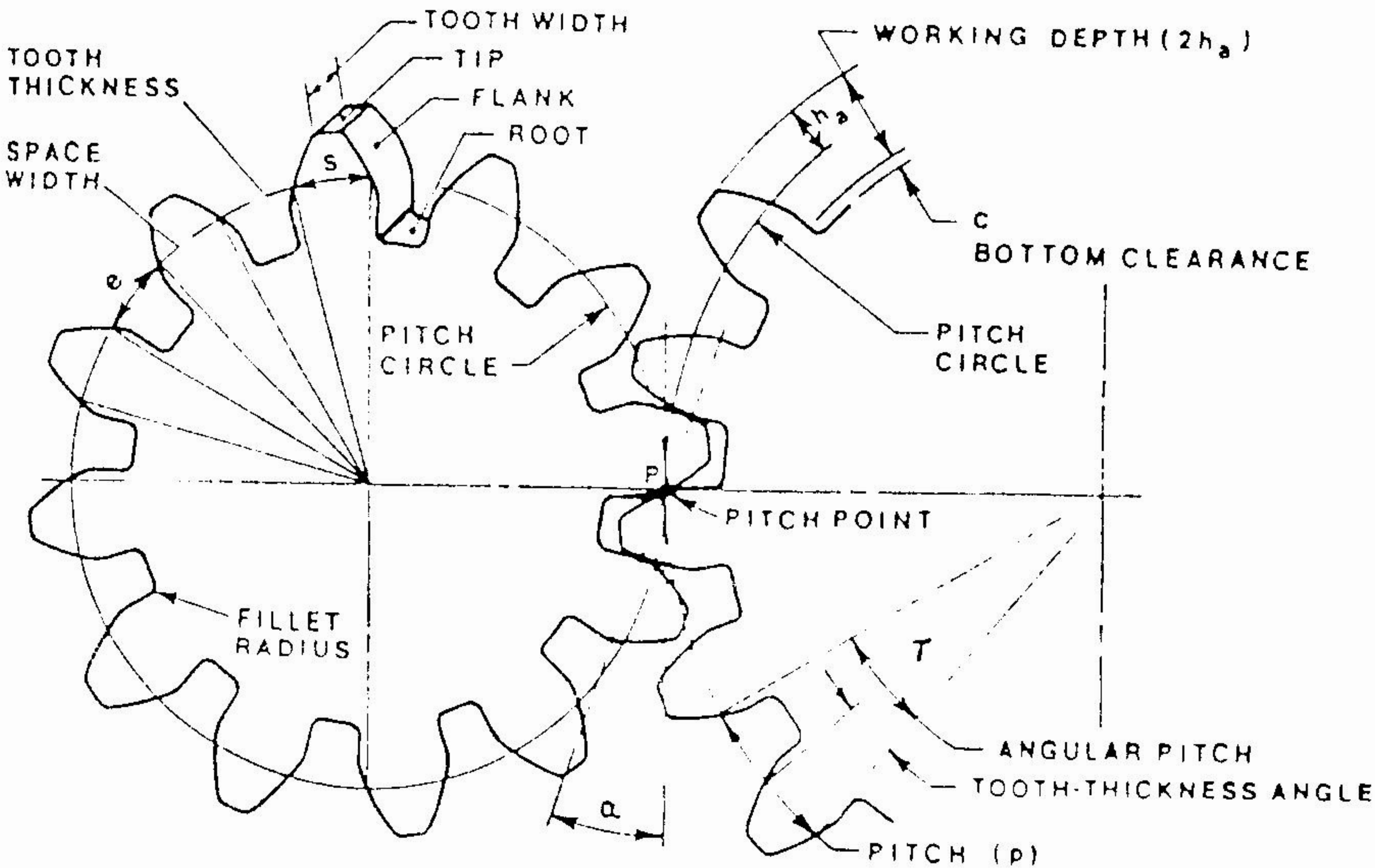

DIFFERENTIAL SPIDER PINION GEAR
DIFFERENTIAL SIDE GEAR
DIFFERENTIAL BEARING CONE AND ROLLERS
LH AXLE SHAFT
RH AXLE SHAFT
DIFFERENTIAL
DIFFERENTIAL DRIVE RING GEAR
DIFFERENTIAL CASE
STRAIGHT ROLLER BEARING
TAPER ROLLER BEARING
ADJUSTING NUT
LOCK NUT
AND WASHER
DIFFERENTIAL DRIVE PINION
RA PD 368834



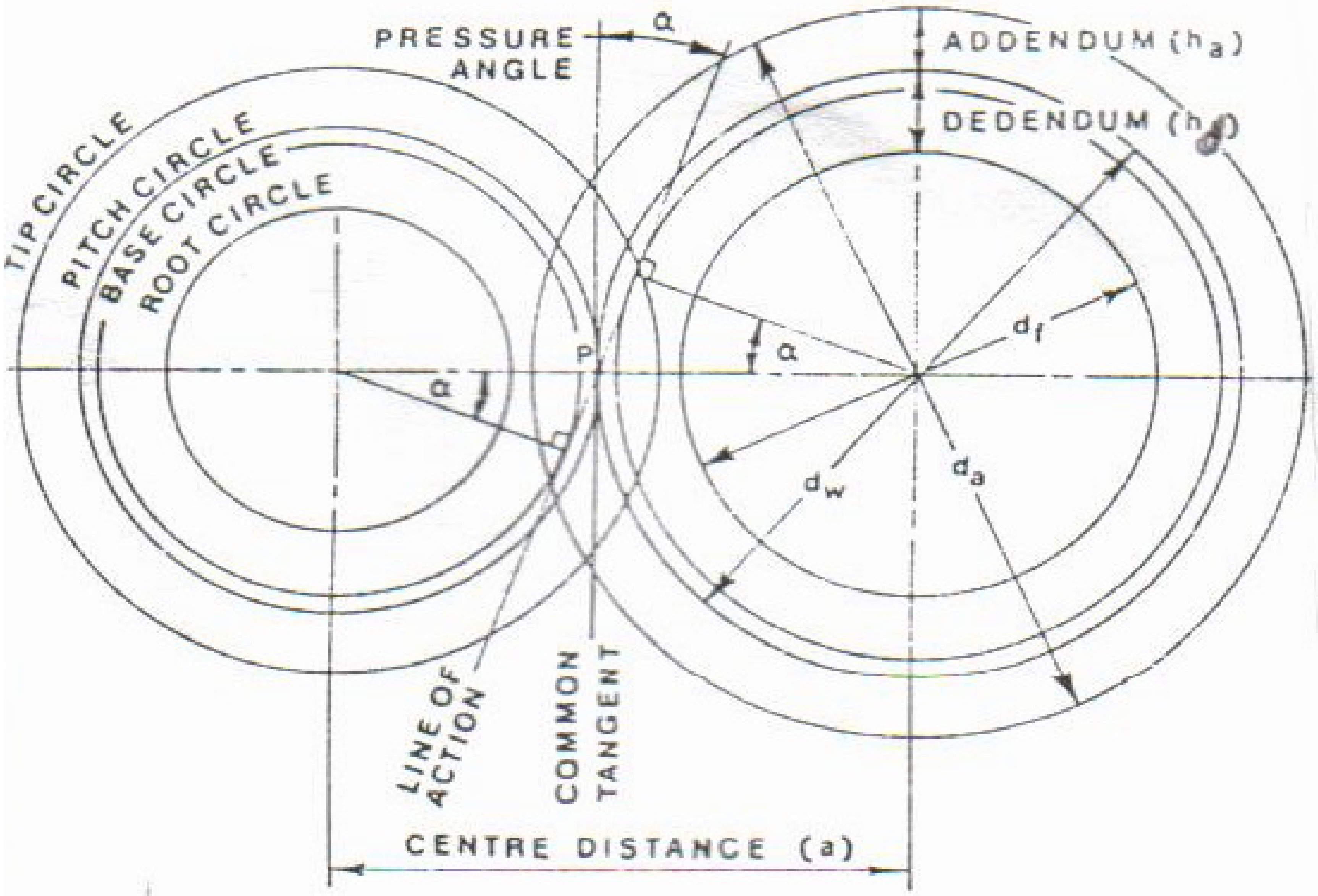
Various Types of Gears

TYPES OF GEARS

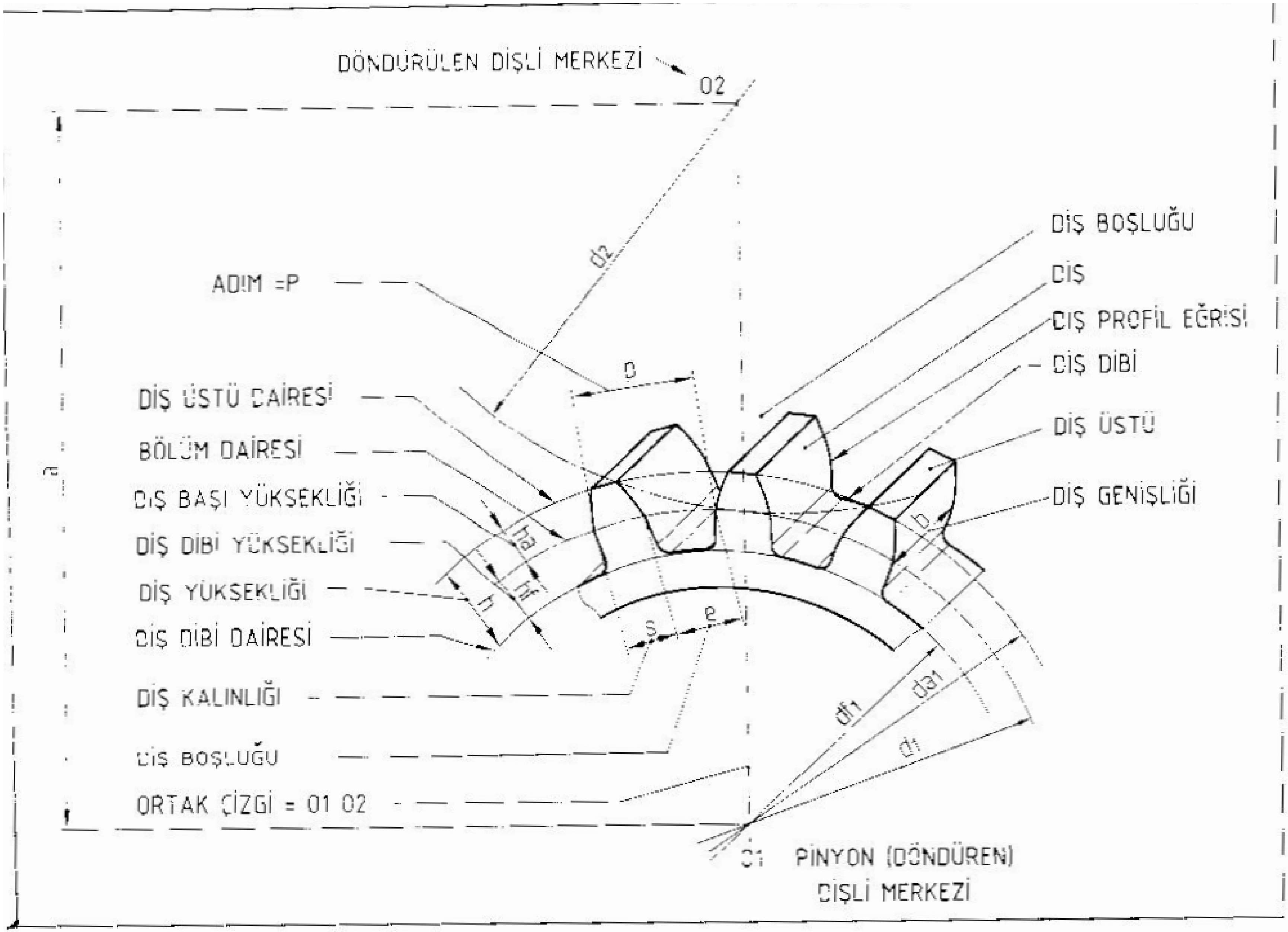




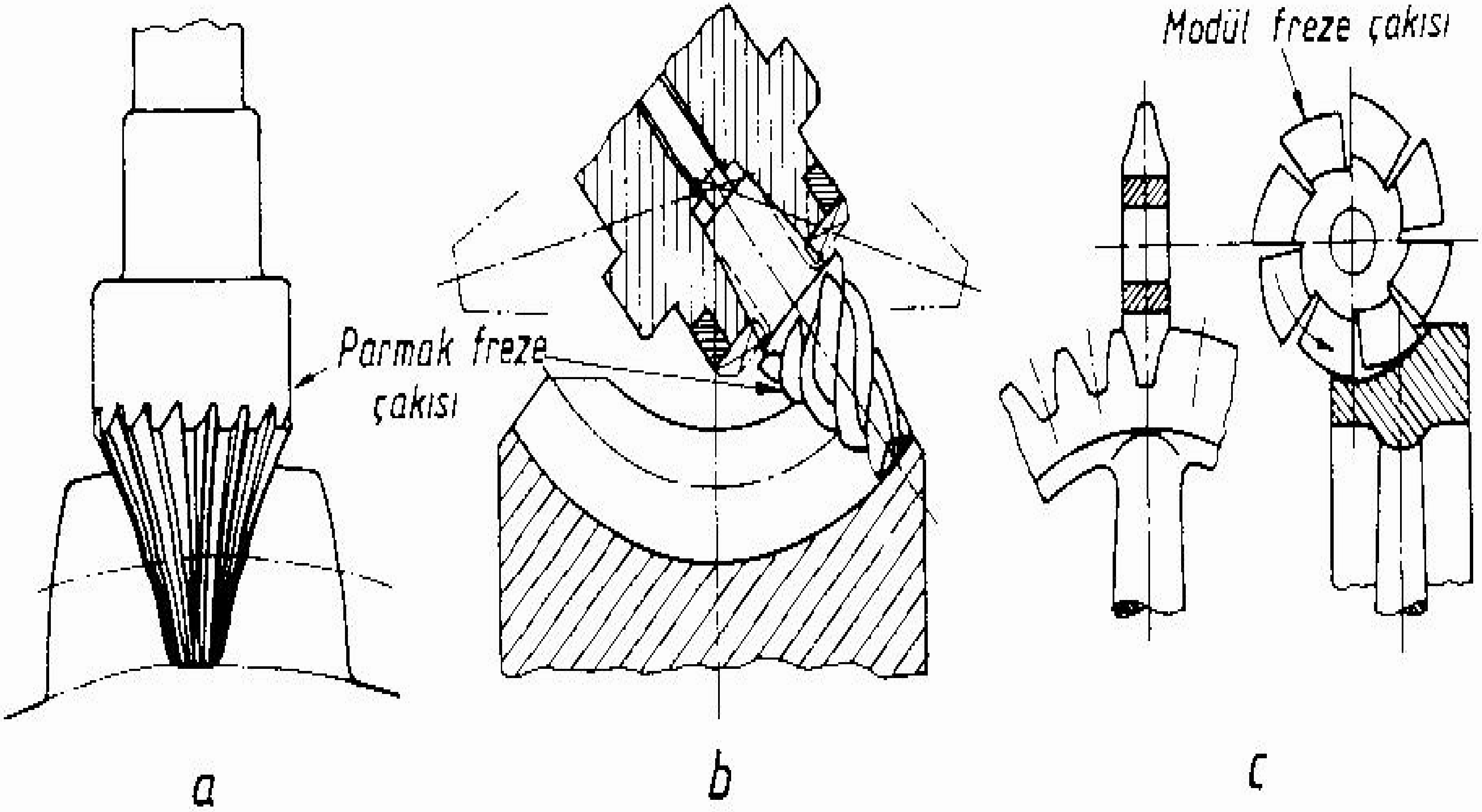
Involute-gear definitions



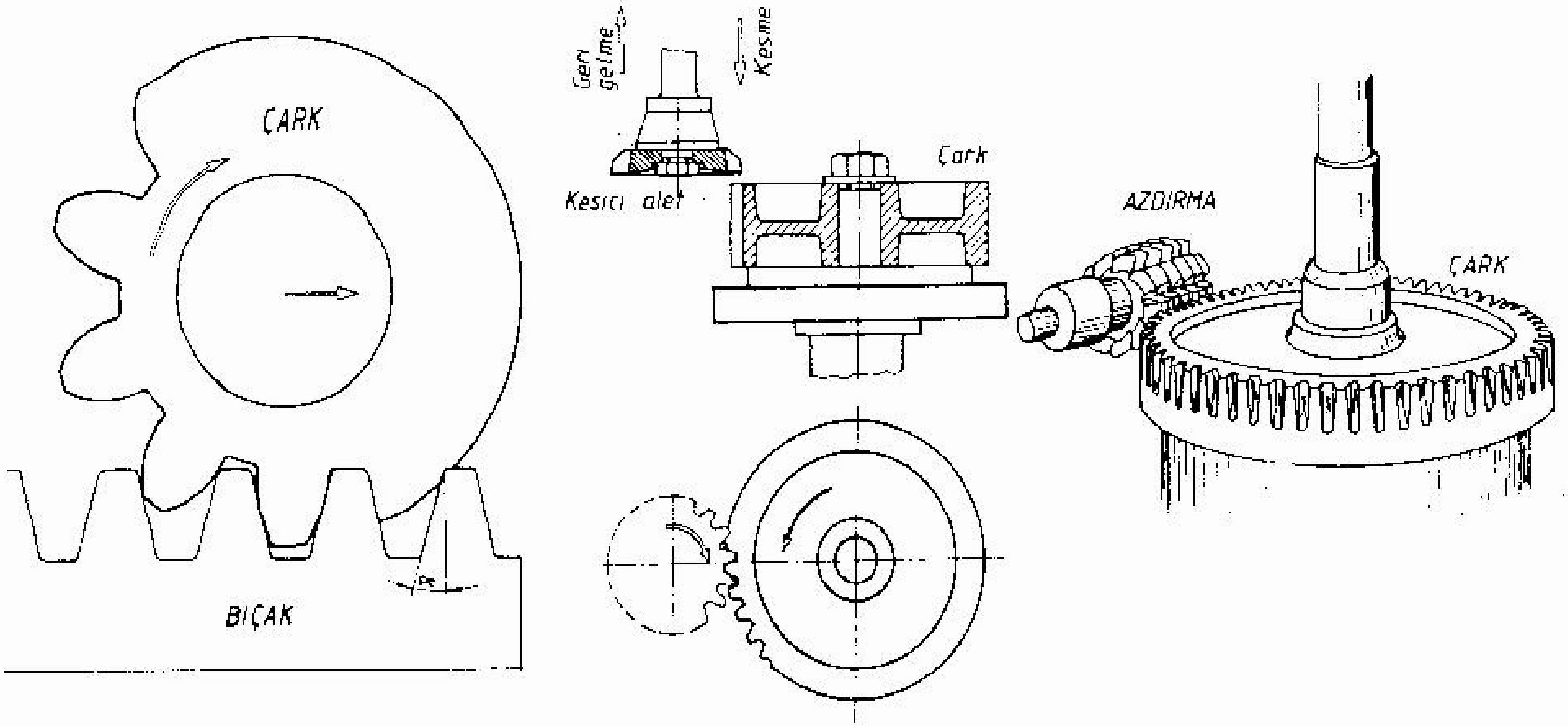
PRESSURE ANGLE
α
ADDENDUM (h_a)
DEDENDUM (h_f)
TIP CIRCLE
PITCH CIRCLE
BASE CIRCLE
ROOT CIRCLE
P
α
α
d_f
d_w
d_a
LINE OF ACTION
COMMON TANGENT
CENTRE DISTANCE (a)



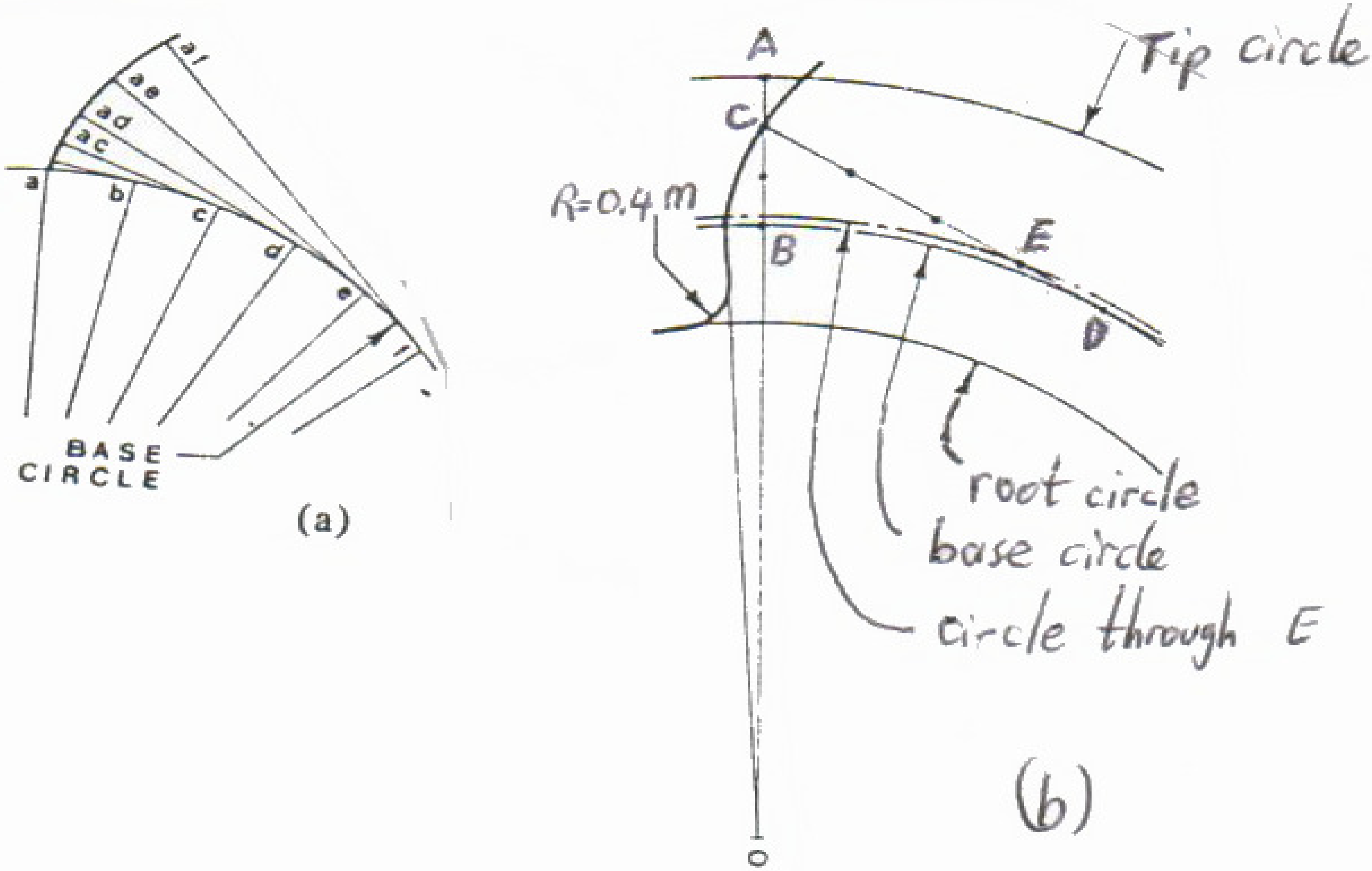
Genel dişli çark terimleri



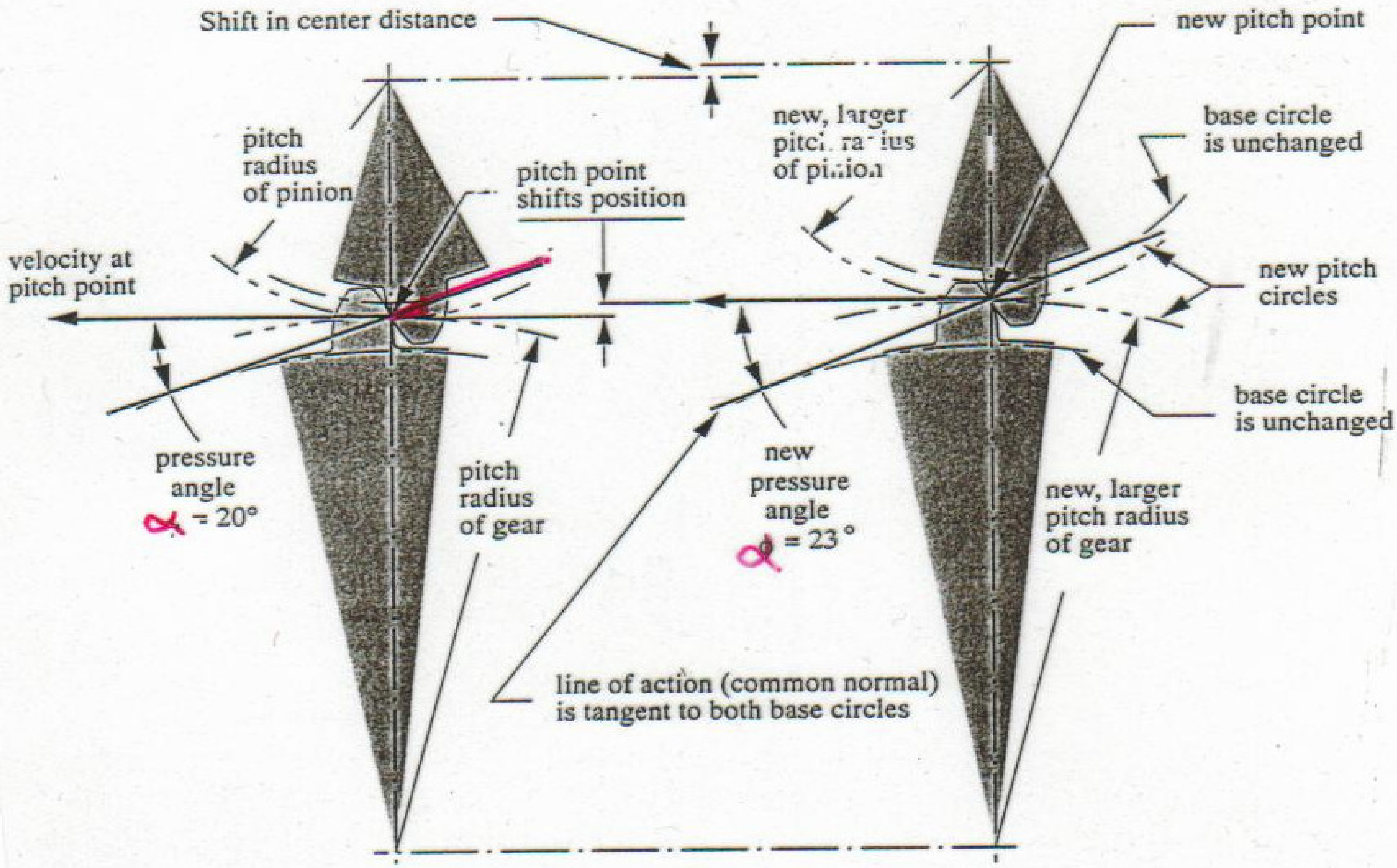
Frezeyle dış açma



a b c



Construction of (a) involute and (b) approximate-involute curves.



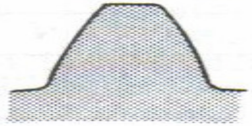
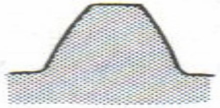
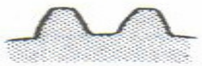
(a) Correct center distance (b) Increased center distance

The 14.5° pressure angle has been used for many years and remains useful for duplication or replacement of gearing. Standard angles of 20° and 25° have become the standard for new gearing because of the smoother and quieter-running characteristics, greater load-carrying ability, and the fewer number of teeth affected by undercutting.

Standard spur gears having a 14.5° pressure angle should have a minimum of 16 teeth with at least 40 teeth in mating pair. Gears with 20° pressure angle should have a minimum of 13 teeth with at least 26 teeth in a mating pair.

The formulas for the 14.5°-, 20°-, and the 25°-full-depth teeth are identical.

Gear-teeth sizes.

Module for Metric Size Gears	Diametral Pitch for Inch Size Gears	Pressure Angle 14.5°	Pressure Angle 20°
6.35	4		
5.08	5		
4.23	6		
3.18	8		
2.54	10		
2.17	12		
1.59	16		
1.27	20		
1.06	24		
0.79	32		

Basic Formulae for Involute Gears :

module , $m = \frac{\text{pitch-circle diameter}}{\text{number of teeth}}$, $\boxed{m = \frac{d_p}{z}}$

pitch , $p = \frac{\text{circumference of pitch circle}}{\text{number of teeth}}$, $p = \frac{\pi d_p}{z} = \boxed{\pi m}$

Tooth thickness , $\boxed{s = \frac{p}{2}}$, space width , $\boxed{e = \frac{p}{2}}$

Addendum , $\boxed{h_a = m}$, clearence , $\boxed{c = 0.167 * h_a}$
(or 0.157)

Dedendum , $h_d = h_a + c = \boxed{1.167\ m}$

Tooth depth $h = h_a + h_d = \boxed{2.167\ m}$

Tooth width , $\boxed{b = (2.5 \sim 3.5)\, p}$ Base Circle Diameter , $\boxed{d_b = d_p \cos\alpha}$

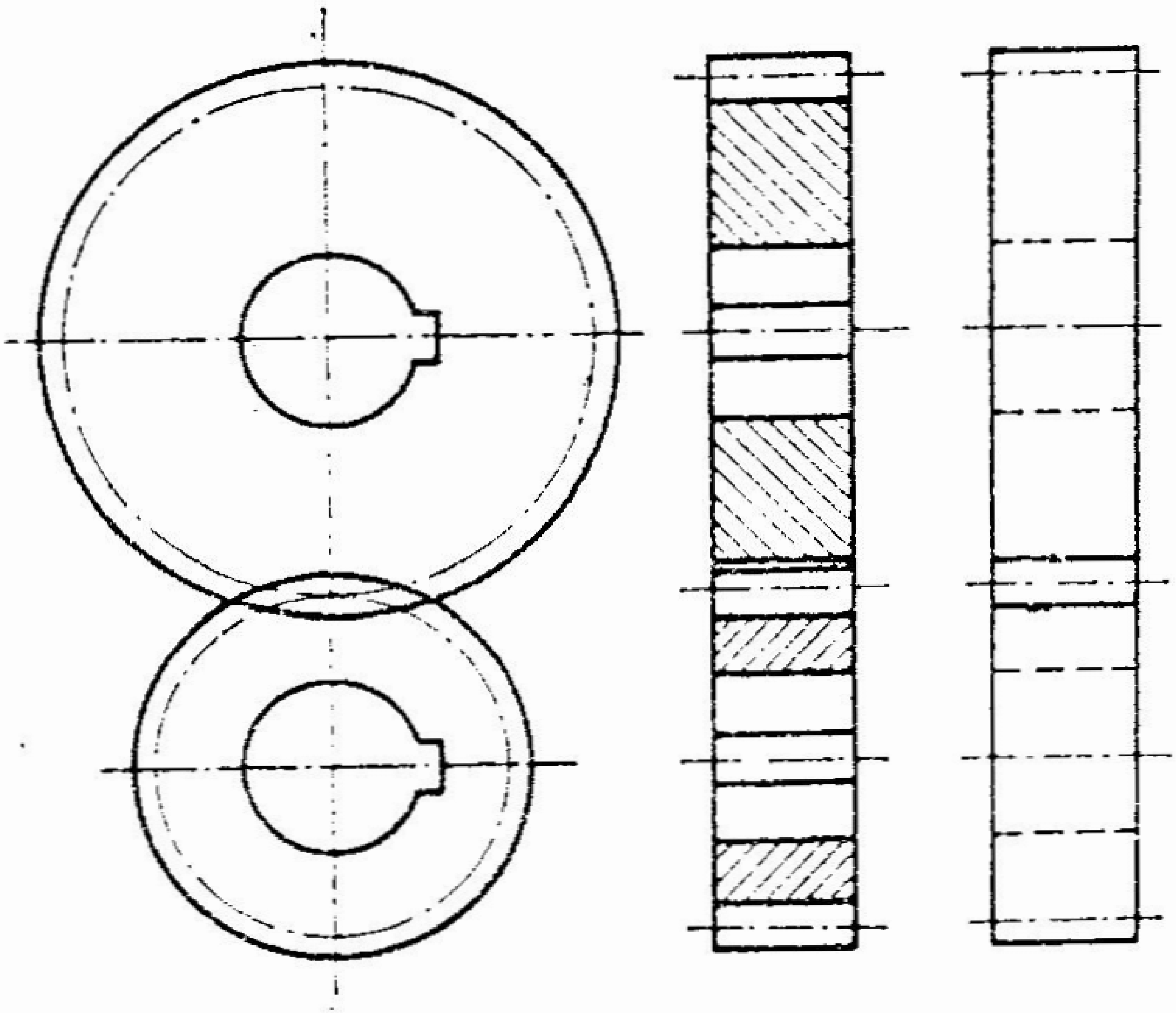
$\alpha = 14.5^\circ$
(or 20°)

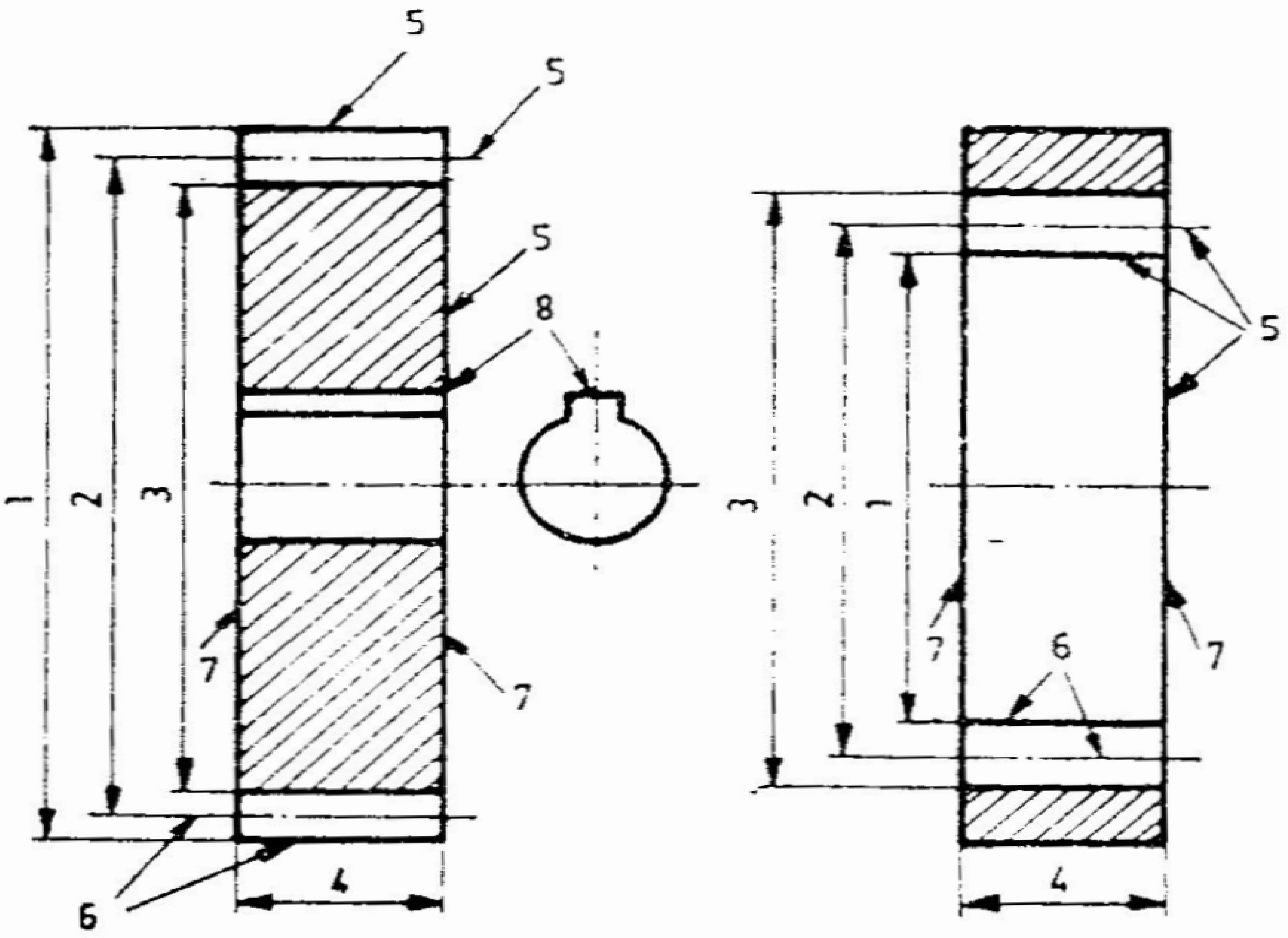
Fillet Radius $= \boxed{0.4\ m}$ (or 0.2 m)

center distance $\boxed{a = \frac{d_{p_1} + d_{p_2}}{2}}$ Gear Ratio $\boxed{U = \frac{z_2}{z_1} = \frac{d_{p_2}}{d_{p_1}}}$ wheel / pinion Angular pitch , $\boxed{\tau = \frac{360^\circ}{z}}$

Modules for spur and bevel gears [mm] (TS 429, DIN 780)

Series I	0,05	0,06	0,08	0,1	0,12	1,16	0,20	0,25	0,3	0,4	0,5	0,6	0,7	0,8
	0,9	1	1,25	1,5	2	2,5	3	4	5	6	8	10	12	16
	20	25	32	40	50	60								
Series II (not preffered)	0,055	0,07	0,09	0,11	0,14	0,18	0,22	0,28	0,35	0,45	0,55	0,65	0,75	0,85
	0,95	1,125	1,375	1,75	2,25	2,75	3,5	4,5	5,5	7	9	11	14	18
	22	28	36	45	55	70								



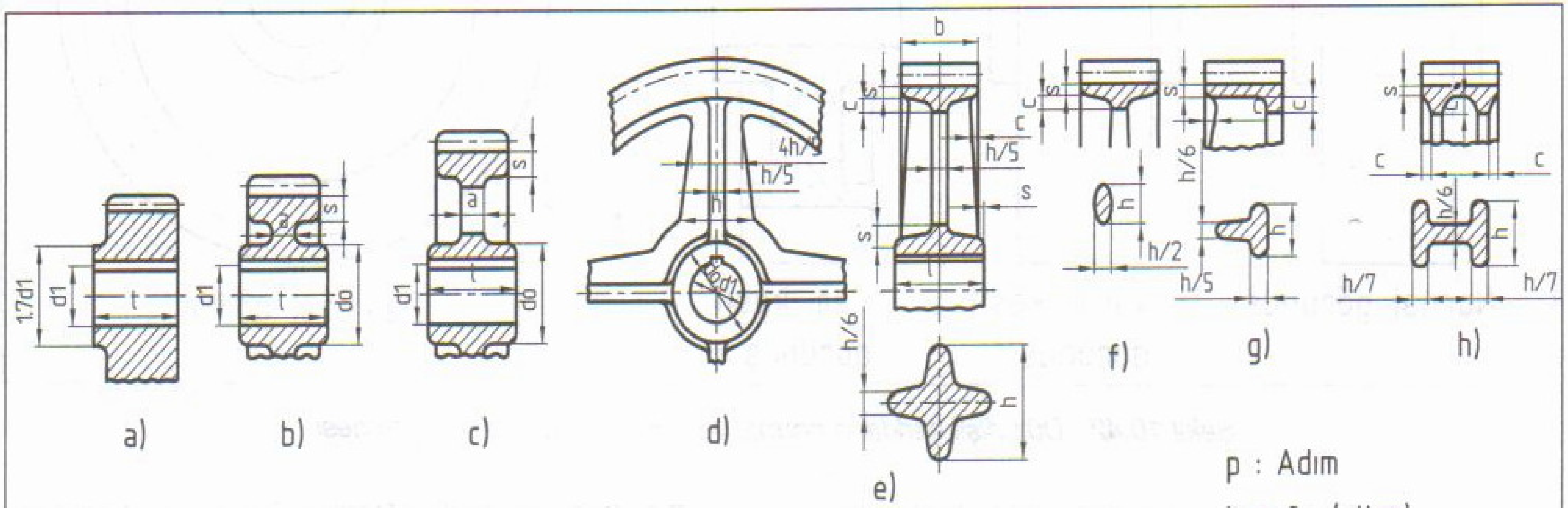


5
5
5
8
1
2
3
7
7
6
4
3
2
1
5
7
6
7
4

$$\# \text{ of arms} = \frac{\sqrt{d_p}}{7} \quad \text{if } d_p \geq 200 \cdot \text{(Gears should be armed)}$$

$200 \leq d_p \leq 600 \Rightarrow 4 \sim 5$ arms | $1500 < d_p \leq 2400 \Rightarrow 8$ arms

$600 < d_p \leq 1500 \Rightarrow 6$ arms | for greater values $\Rightarrow 10 \sim 12$ arms.



p : Adım

b = (2,5 ~3,5)p

do= 2d1

s = 0,6p

l = (1,5~2)d1

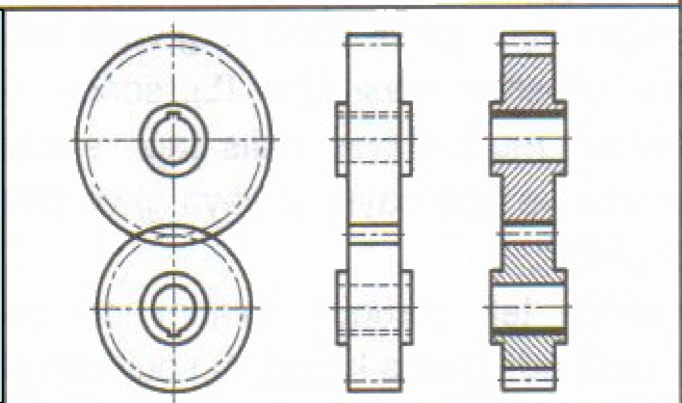
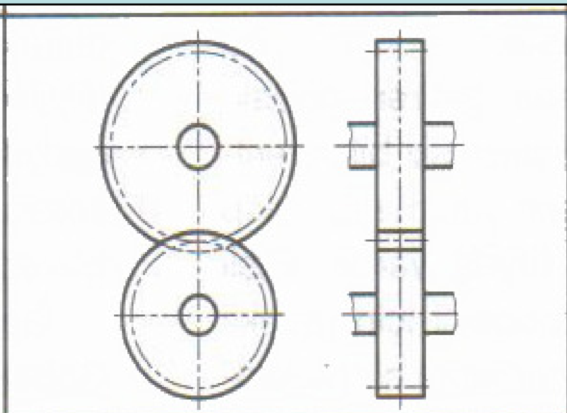
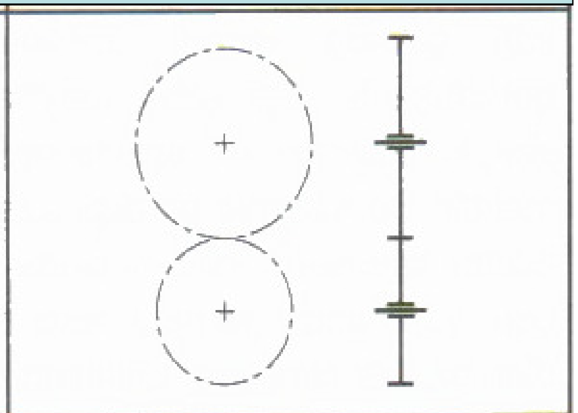
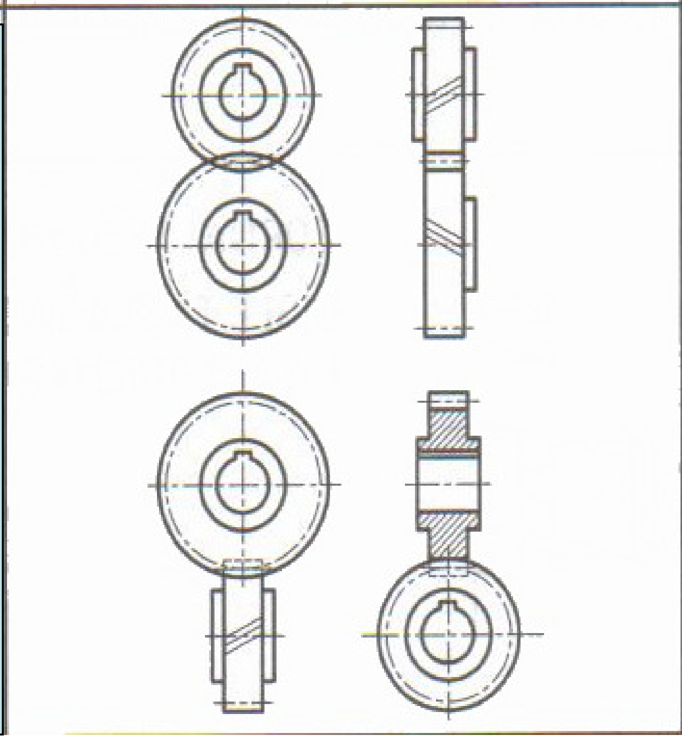
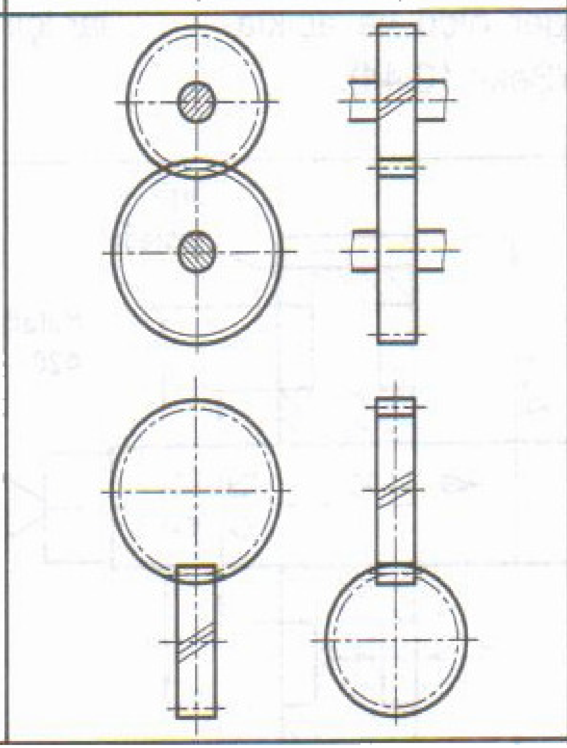
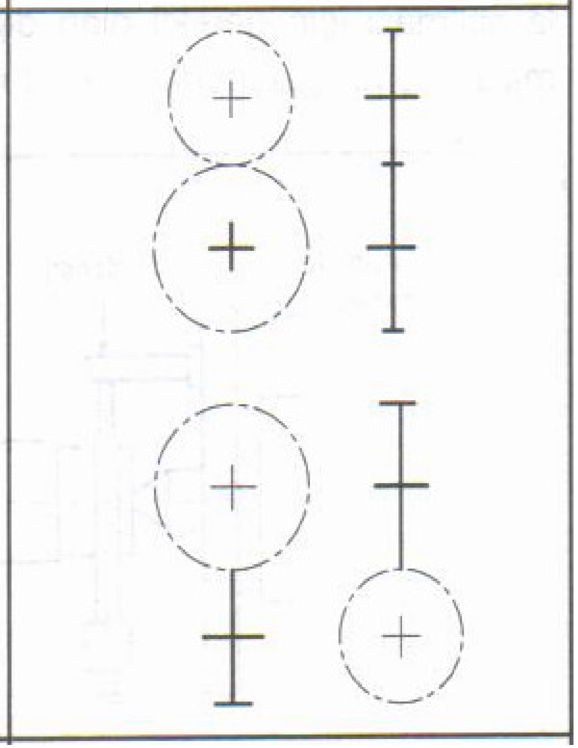
a = 0,7p

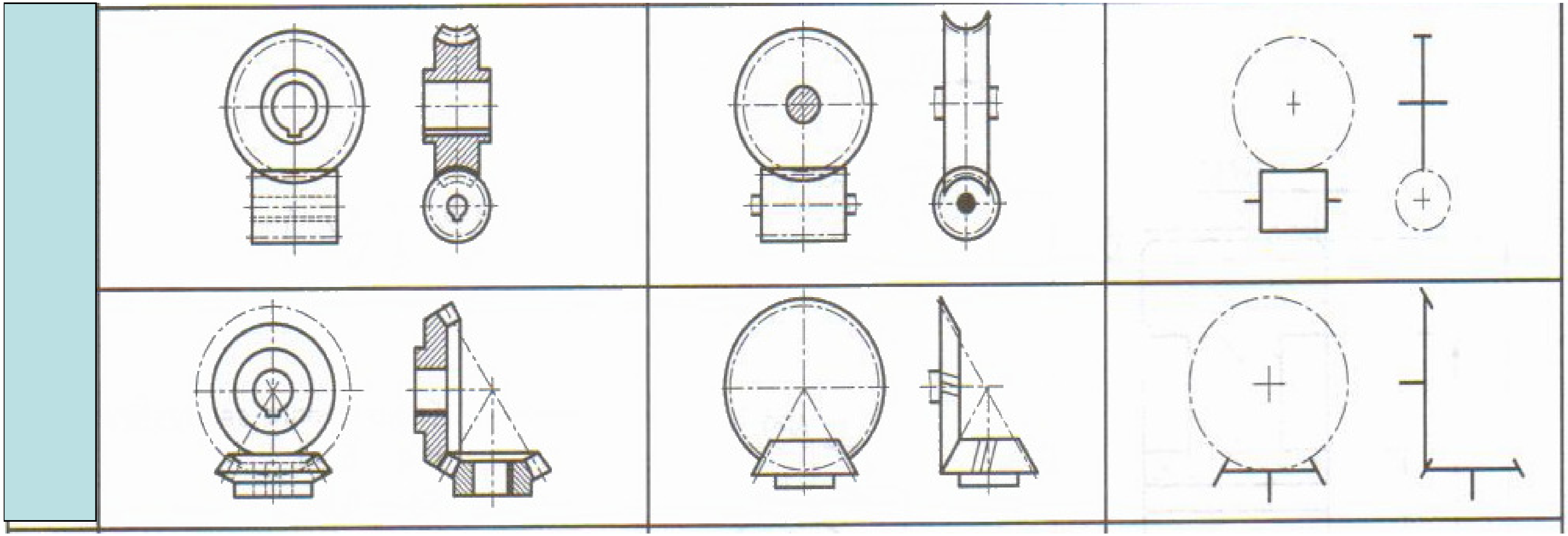
c= 0,6p

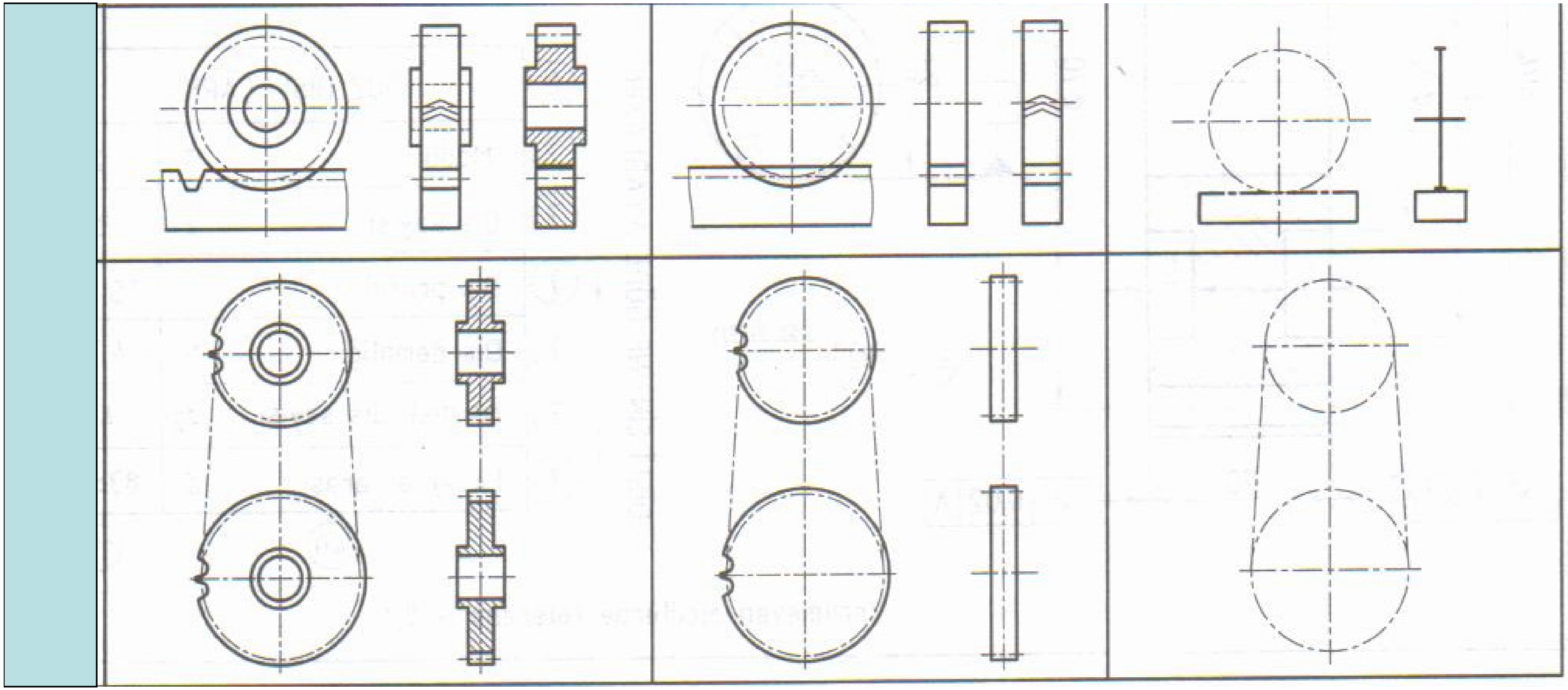
h = 2p (elips)

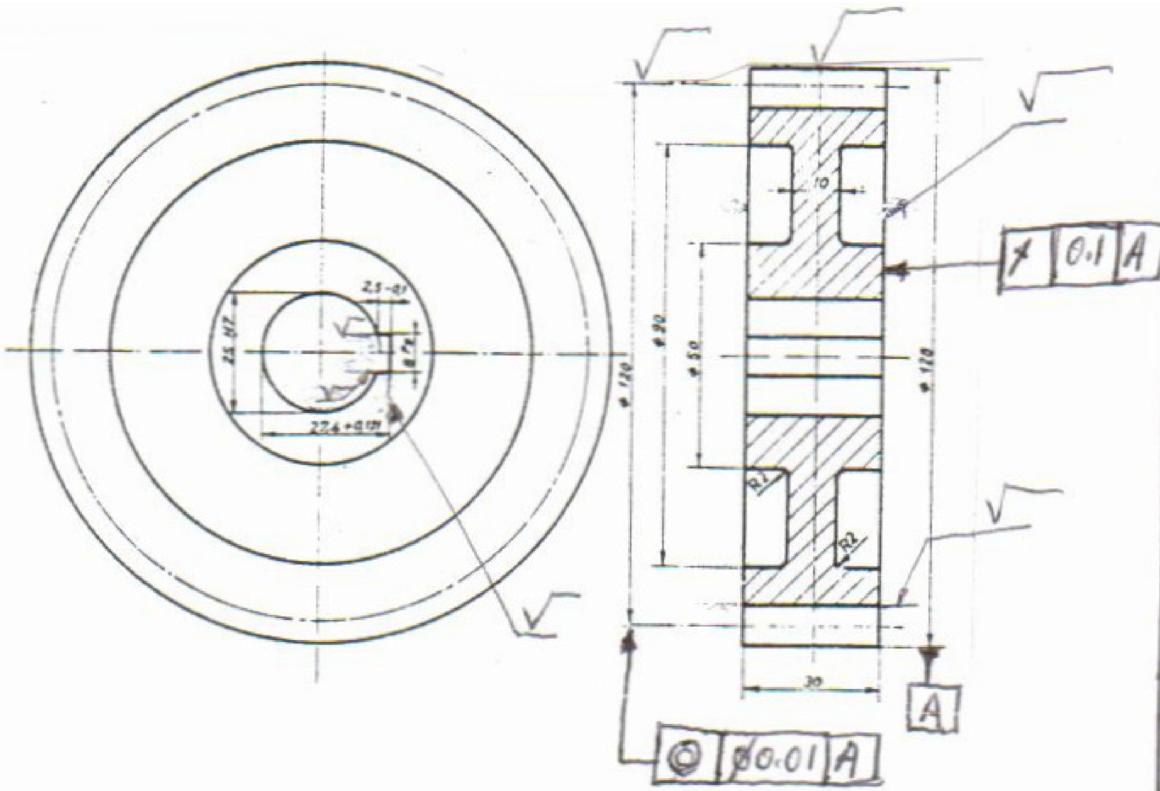
h = 2,5p (T, H tipi)

Dişli çarkların göbek, kol ve jant ölçüleri

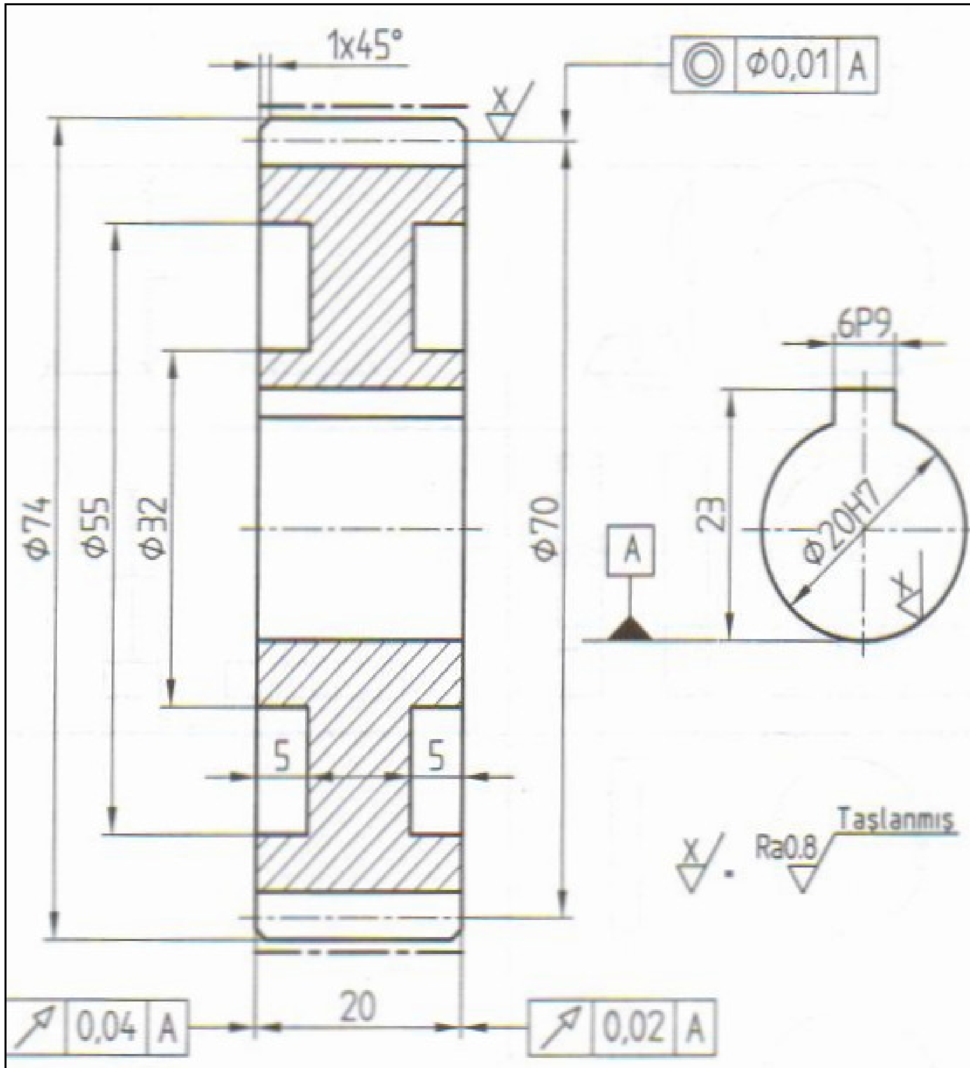






Spur Gear Data		
Number of Teeth	z_1	30
module	m	4
Tooth Form		14.5° Inv.
whole depth	h	8.68
mating gear part no:		6852
Number of teeth of mating gear	z_2	42
Center distance		144 ± 0.025

A detail drawing of a spur gear.



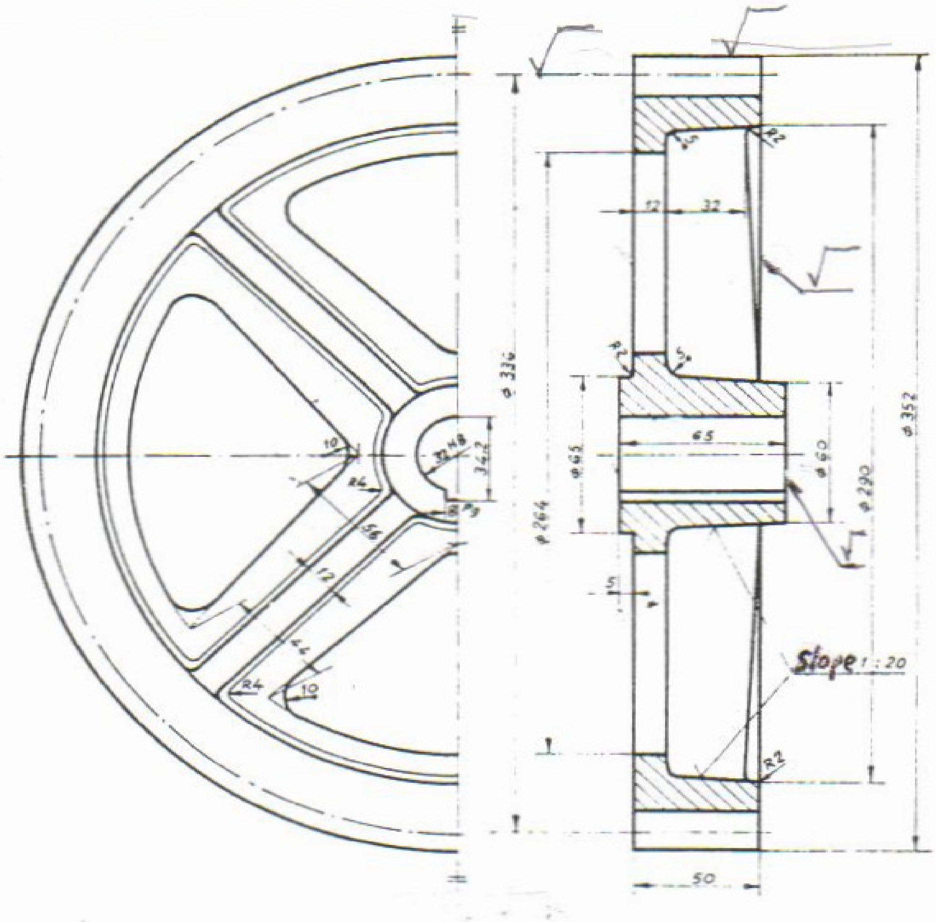
Ra1.6 (X)

Endüksiyonla sertleştirilmiş
HRc : 52+1
Sd : 0,8

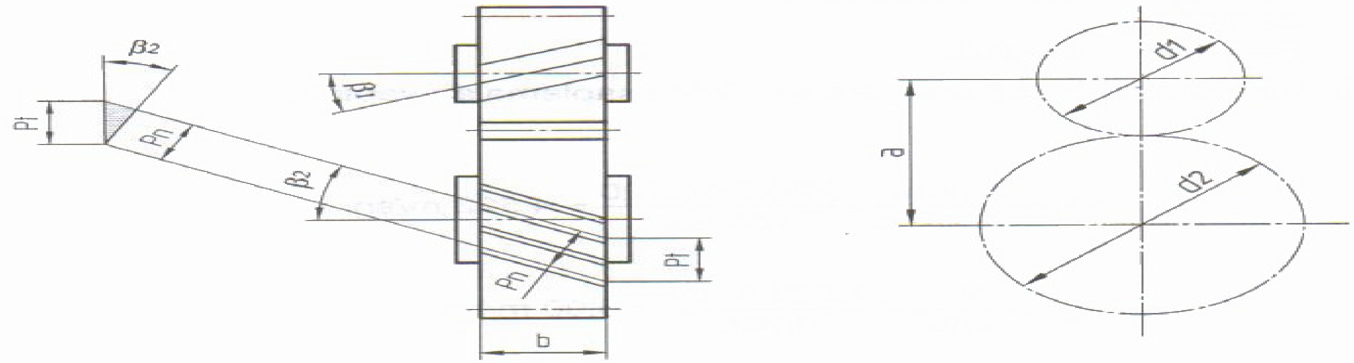
DİŞLİ CİNSİNE GÖRE AYARLANIR

8	DÜZ DİŞLİ ÇARK		
7	Modül	m	2
7	Diş sayısı	z	35
7	Diş profili		TS 3601
7	Diş derinliği	h	4,332
7	Eş dişli diş sayısı	z_2	48
7	Eksenler arası	a	83±0,05
	40		15

Verilmeyen ölçülerde tolerans : ± 0,1

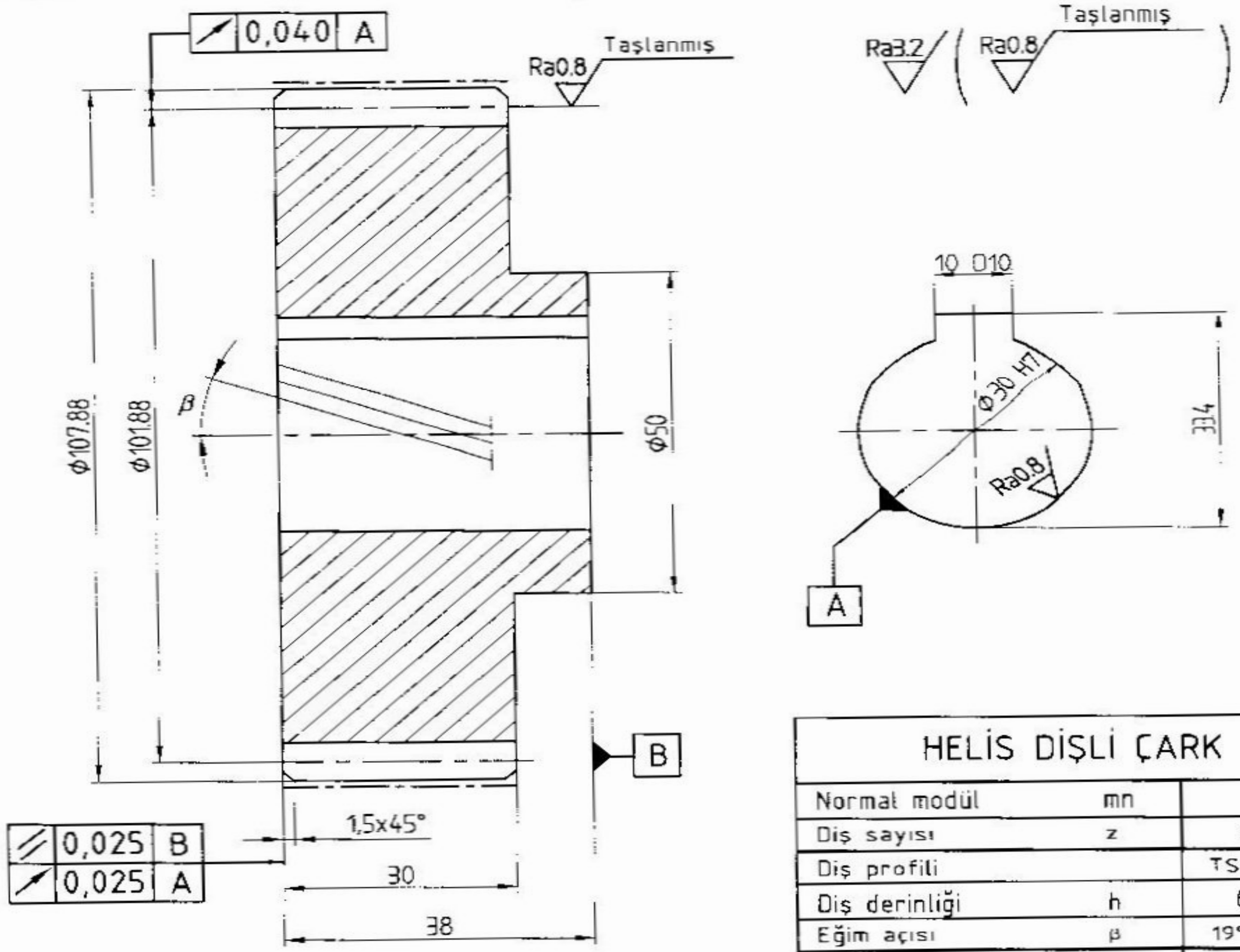


Spur Gear Data



HELİS DİŞLİ ÇARKLAR

Eğim açısı	β	Eksenler paralel ise ; $\beta_1=\beta_2$ ve en fazla 20° Eksenler kesişiyorsa ve $\Sigma=90° \Rightarrow \beta_1=\beta_2$ Helis yönleri aynı $\beta_1<\beta_2 \Rightarrow \beta_2=\Sigma-\beta_1$ Helis yönleri farklı $\beta_1>\beta_2 \Rightarrow \beta_2=\Sigma+\beta_1$
Eksenler açısı	Σ	$\beta_1+\beta_2$
Normal adım	p_n	$m_n\alpha=p_t.\cos\beta=\dfrac{z.m_n}{\cos\beta}$
Alın adımı	p_t	$\dfrac{p_n}{\cos\beta}=\dfrac{m_n.\pi}{\cos\beta}$
Normal modül	m_n	$\dfrac{p_n}{\pi}=m_t.\cos\beta$
Alın modülü	m_t	$\dfrac{p_t}{\pi}=\dfrac{m_n}{\cos\beta}$
Bölüm dairesi çapı	d	$z.m_t=\dfrac{z.m_n}{\cos\beta}=\dfrac{z.p_n}{\pi.\cos\beta}$
Diş üstü çapı	d_a	$d+2m_n$
Helis adımı	p_z	$d.\pi.\cot\beta=\dfrac{\pi.d}{\tan\beta}$
Helis açısı	α	$\tan\alpha=\dfrac{p_z}{\alpha.d}$
Diş sayısı	z	$\dfrac{d}{m_t}=\dfrac{\pi.d}{p_t}=\dfrac{d.\cos\beta}{m_n}$
İdeal diş sayısı	z_i	$\dfrac{z}{\cos 3\beta}$
Diş boyu	B	$\dfrac{z}{\cos\beta}$
Diş genişliği	b	3.p veya $\approx 10.m_n$
Diş yüksekliği	h	$\frac{13}{6}\,m_n=2.166.m_n$
Diş üstü yüksekliği	h_a	$1.m_n$
Diş dibi yüksekliği	h_f	$\frac{7}{6}\,m_n=1.166.m_n$
Eksenler arası	a	$\dfrac{d_1+d_2}{2}$



- - - - - Endüksiyonla sertleştirilecek
HRC 50+5
Sd : 1 – 0,2

Genel tolerans:
TS 1980'e göre m

Ölçü	Sembol	Tolerans
φ30	H7	+ 0,021 / 0
10	D10	+ 0,098 / + 0,040

HELİS DİŞLİ ÇARK

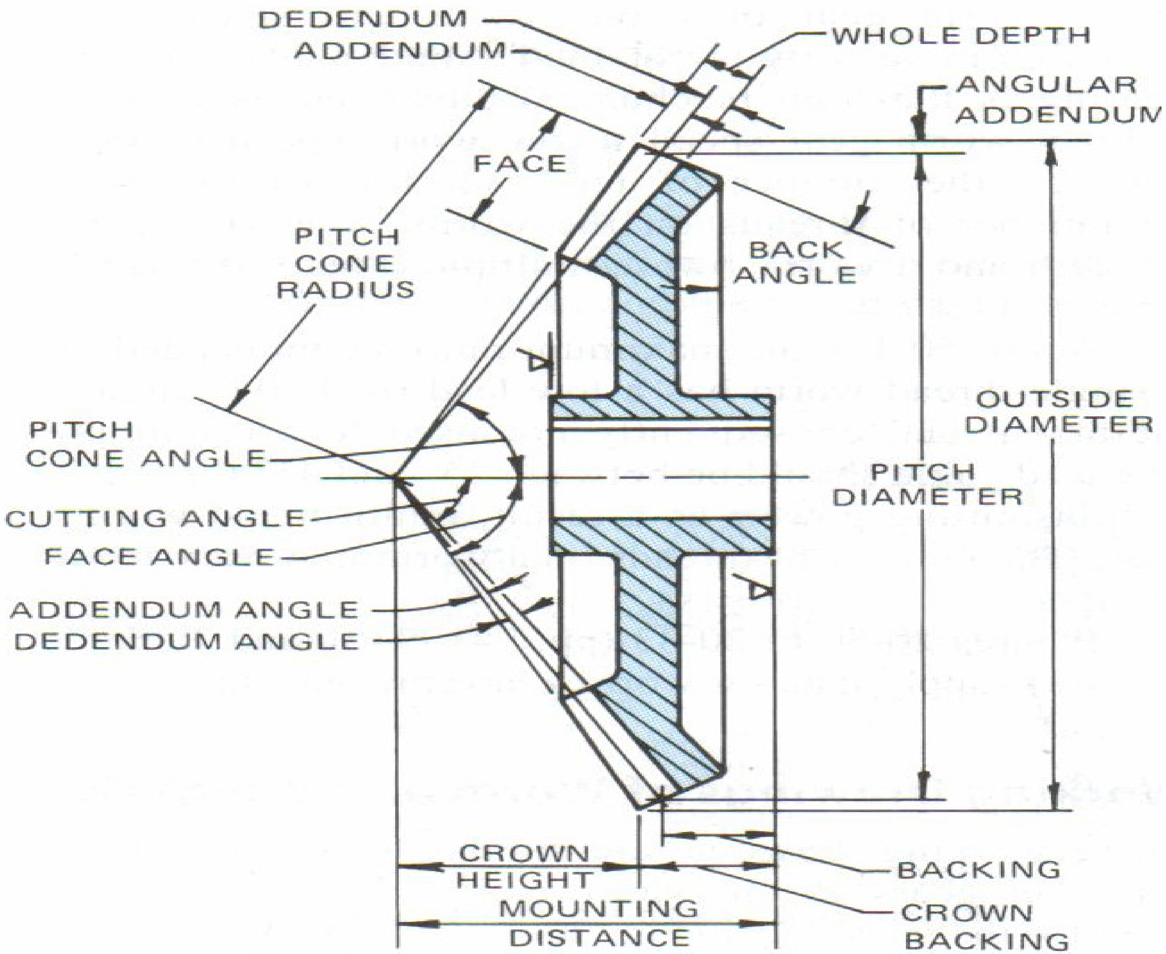
Normal modül	mn	3
Diş sayısı	z	32
Diş profili		TS 3601
Diş derinliği	h	6,5
Eğim açısı	β	19° 30'
Helis açısı	α	70° 30'
Helis yönü		Sağ
Helis adımı	pz	903,5
Eş dişli diş sayısı	z2	56
Eş dişli Resim Nr.		DK. 5
Eksenler arası	a	140,03 ±0,1
İdeal diş sayısı	z.	38

	Tarih	Adı	İmza	Sayı	Gereç
Çizen					
Kontrol					

Ölçek		Resim Nr.
1:1	HELİS DİŞLİ ÇARK	DK . 4

BEVEL GEARS

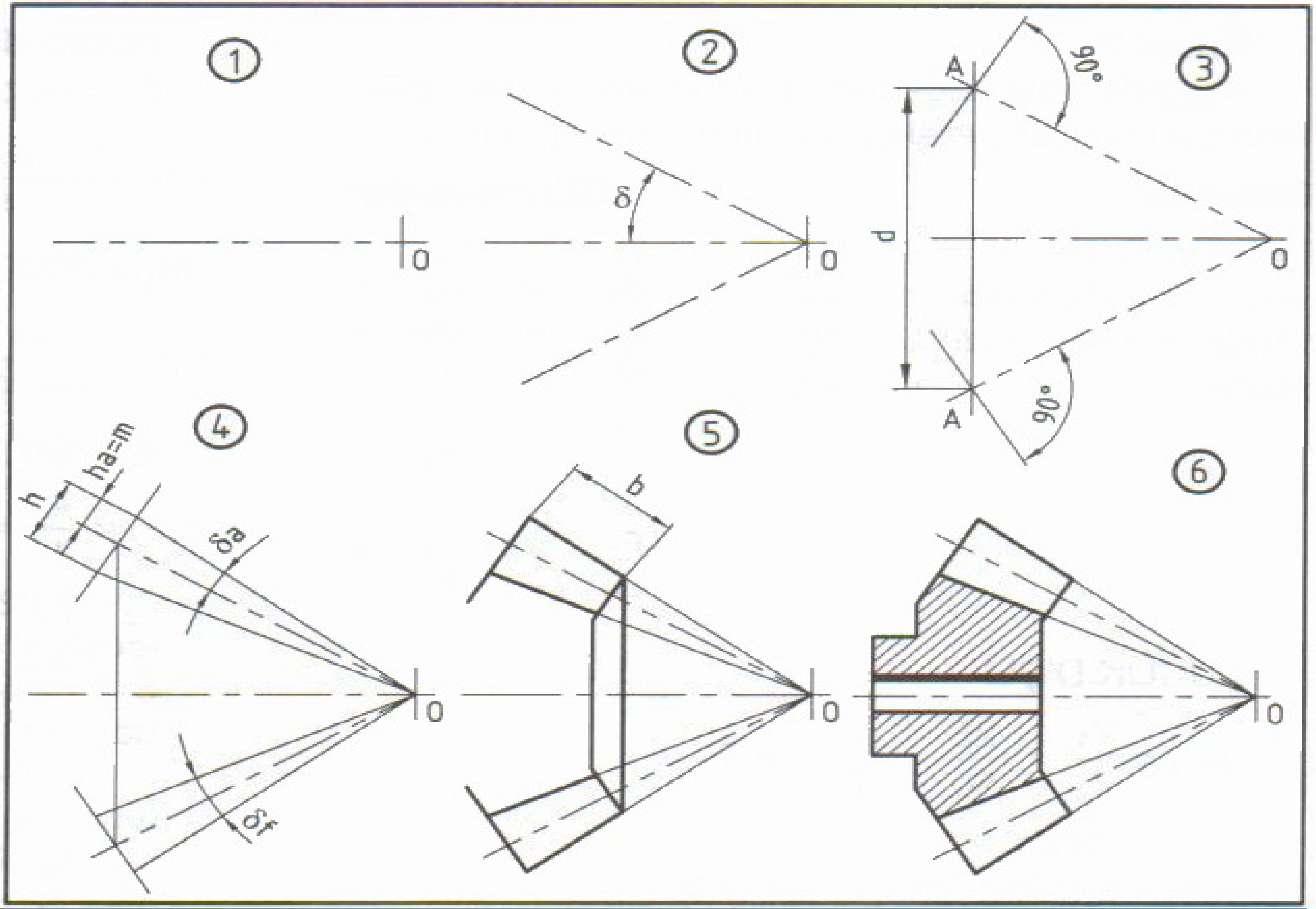
Bevel gears are used to transmit power between two shafts whose axes intersect. The axes may intersect at any angle, but the most common is 90°. They are similar to rolling cones having the same apex. The teeth are the same shape as spur gear teeth but taper toward the cone apex. Therefore, many spur gear terms may apply to bevel gears. **Miter gears** are bevel gears having the same diametral pitch or module, pressure angle, and number of teeth.



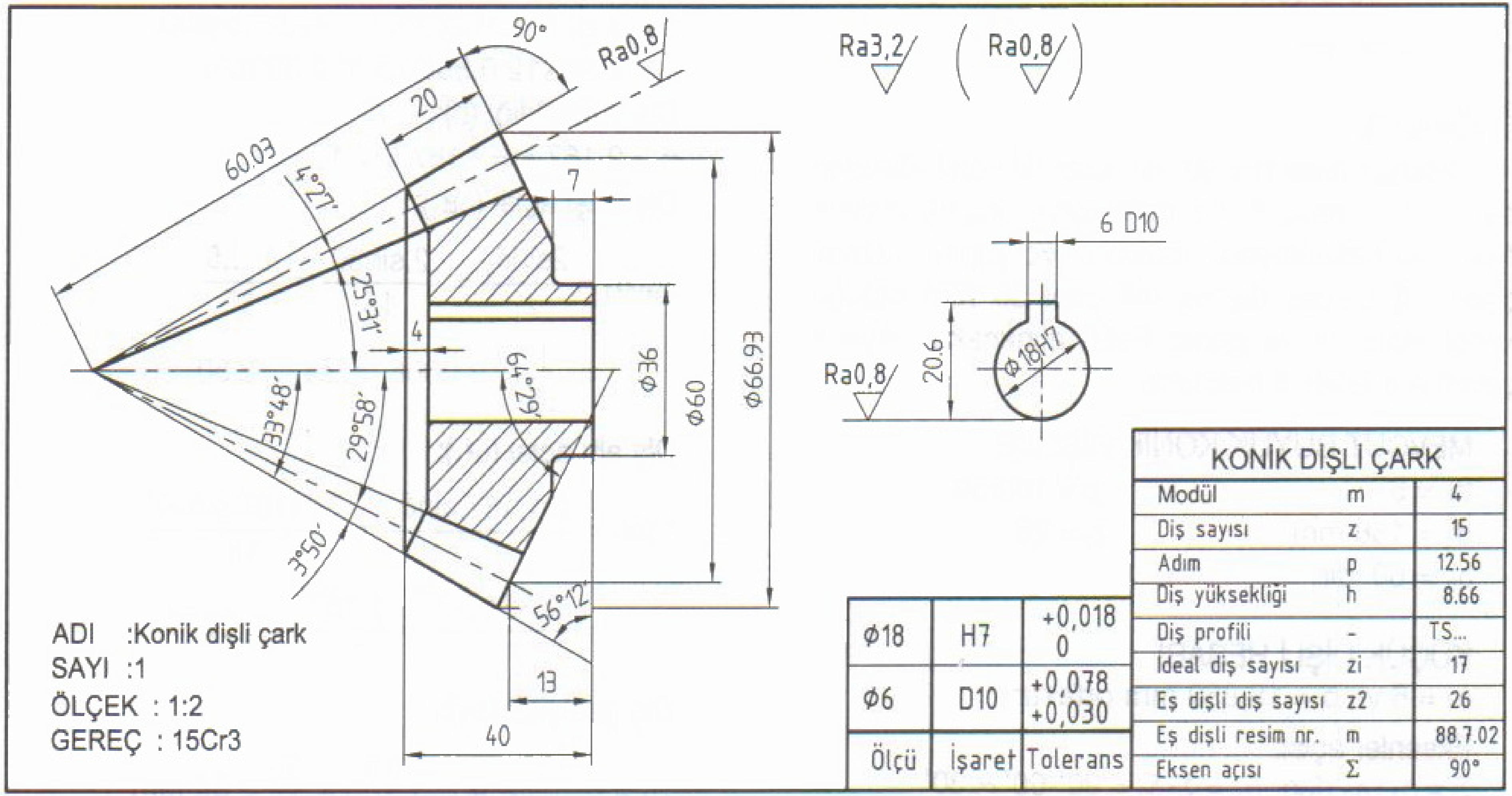
Bevel gear nomenclature.

Bevel gear formulas.

Term	Formula
Addendum, dedendum, whole depth, pitch diameter, module, diametral pitch, number of teeth, circular pitch, chordal thickness, circular thickness	Same as for spur gears
Pitch cone radius	$\frac{\text{PD}}{2 \times \text{sin of pitch angle}}$
Pitch cone angle	$\text{Tan pitch angle} = \frac{\text{PD of gear}}{\text{PD of pinion}} = \frac{\text{N of gear}}{\text{N of pinion}}$
Addendum angle	$\text{Tan addendum angle} = \frac{\text{Addendum}}{\text{Pitch cone radius}}$
Dedendum angle	$\text{Tan dedendum angle} = \frac{\text{Dedendum}}{\text{Pitch cone radius}}$
Face angle	Pitch cone angle plus addendum angle
Cutting angle	Pitch cone angle minus dedendum angle
Back angle	Same as pitch cone angle
Angular addendum	Cos of pitch cone angle × addendum
Outside diameter	Pitch diameter plus two angular addendums
Crown height	Divide ½ the outside diameter by the tangent of the face angle
Face width	1½ to 2½ times the circular pitch
Chordal addendum	$\text{Addendum} + \frac{\text{circular thickness}^2 \times \text{cos pitch cone angle}}{4\text{PD}}$



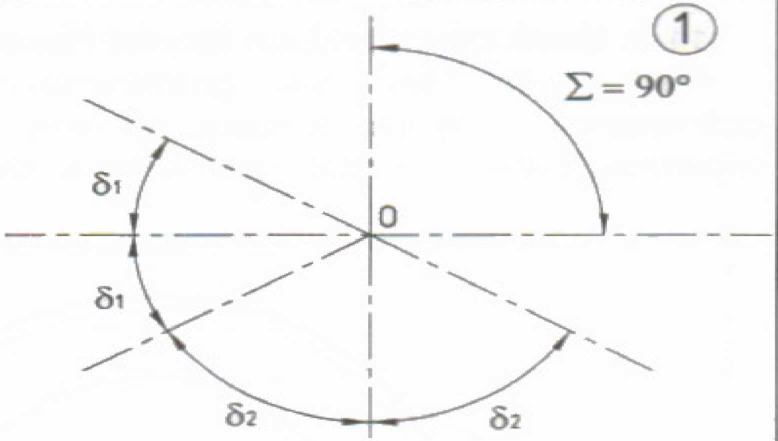
1
0
2
δ
0
3
A
90°
d
0
A
90°
4
h
ha=m
δa
0
δf
5
b
0
6
0

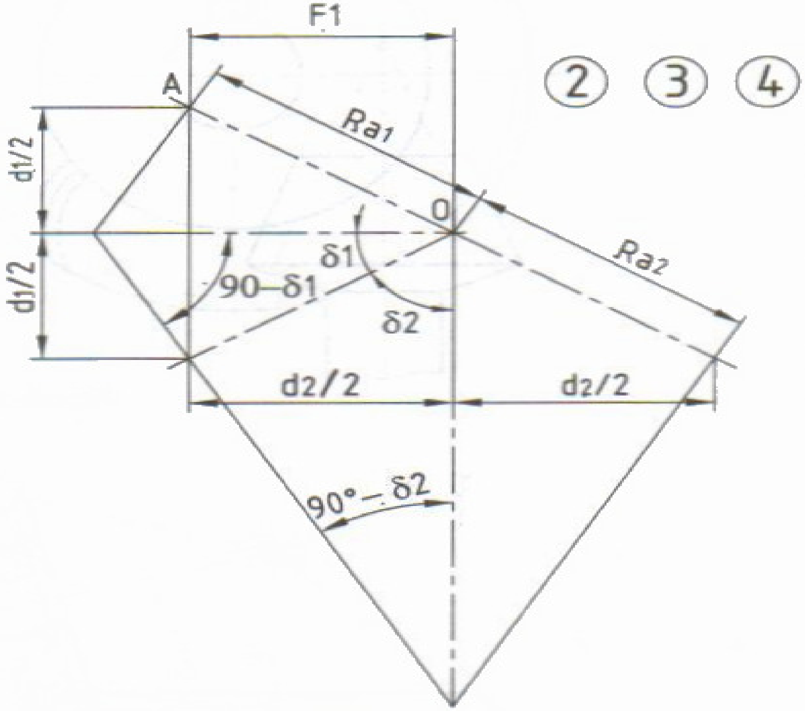


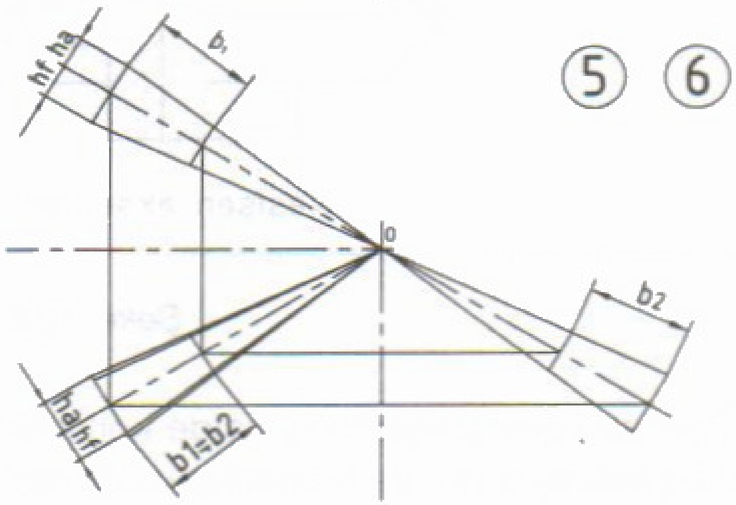
KONİK DİŞLİ ÇARK		
Modül	m	4
Diş sayısı	z	15
Adım	p	12.56
Diş yüksekliği	h	8.66
Diş profili	-	TS...
İdeal diş sayısı	zi	17
Eş dişli diş sayısı	z2	26
Eş dişli resim nr.	m	88.7.02
Eksen açısı	Σ	90°

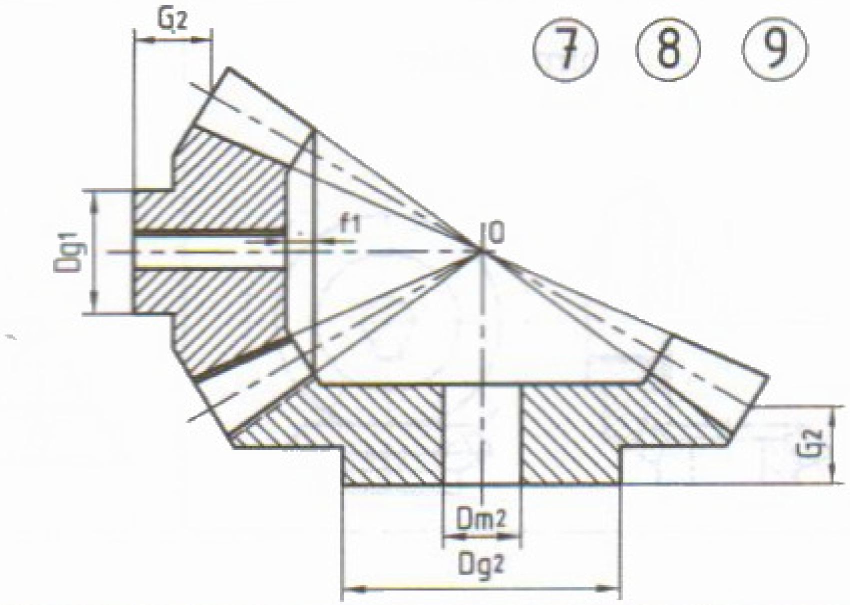
Ölçü	İşaret	Tolerans
Ø18	H7	+0,018 0
Ø6	D10	+0,078 +0,030

Detailed Drawing of a Bevel Gear


1
Σ = 90°
δ1
δ1
O
δ2
δ2


F1
2
3
4
A
Ra1
d1/2
O
δ1
90−δ1
δ2
Ra2
d1/2
d2/2
d2/2
90°−δ2


hf ha
b1
5
6
o
b2
ha hf
b1=b2

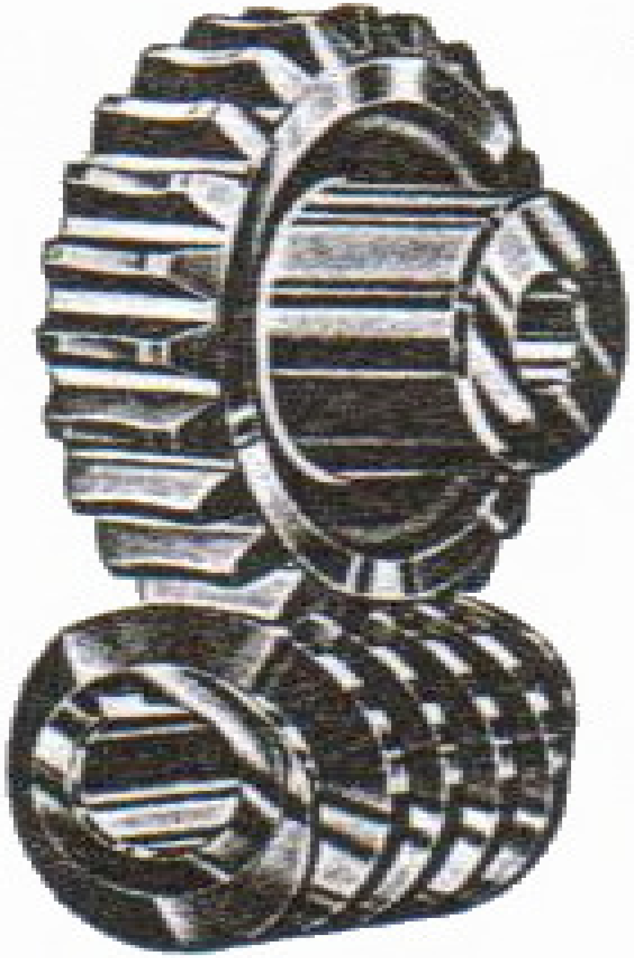

G2
7
8
9
f1
O
Dg1
G2
Dm2
Dg2

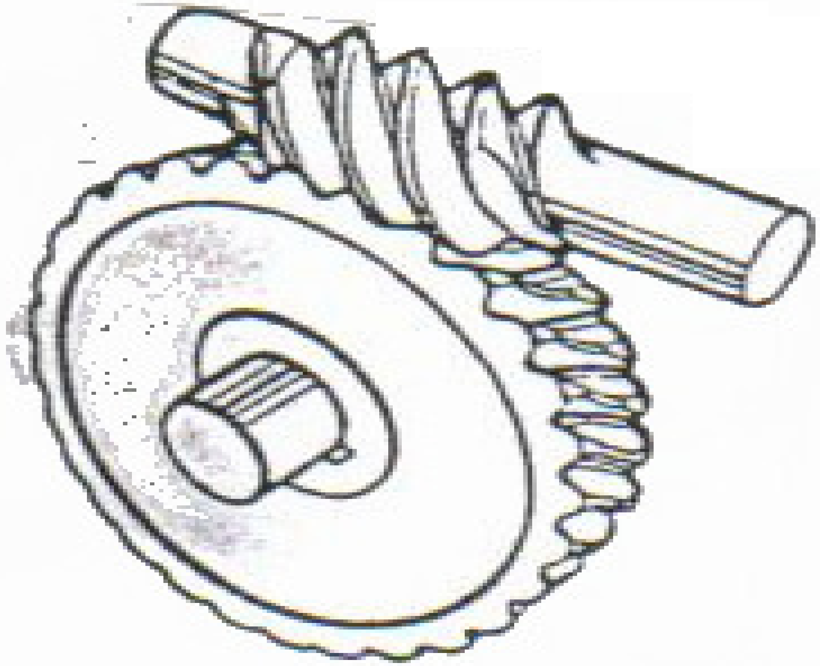
WORMS AND WORM GEARS

Worm gears are used to transmit power between two shafts that are at right angles to each other and are nonintersecting. The teeth on the worm are similar to the teeth on the rack. and the teeth on the worm gear are curved to conform with the teeth on the worm. Thread terms such as *pitch* and *lead* are used on the worm.

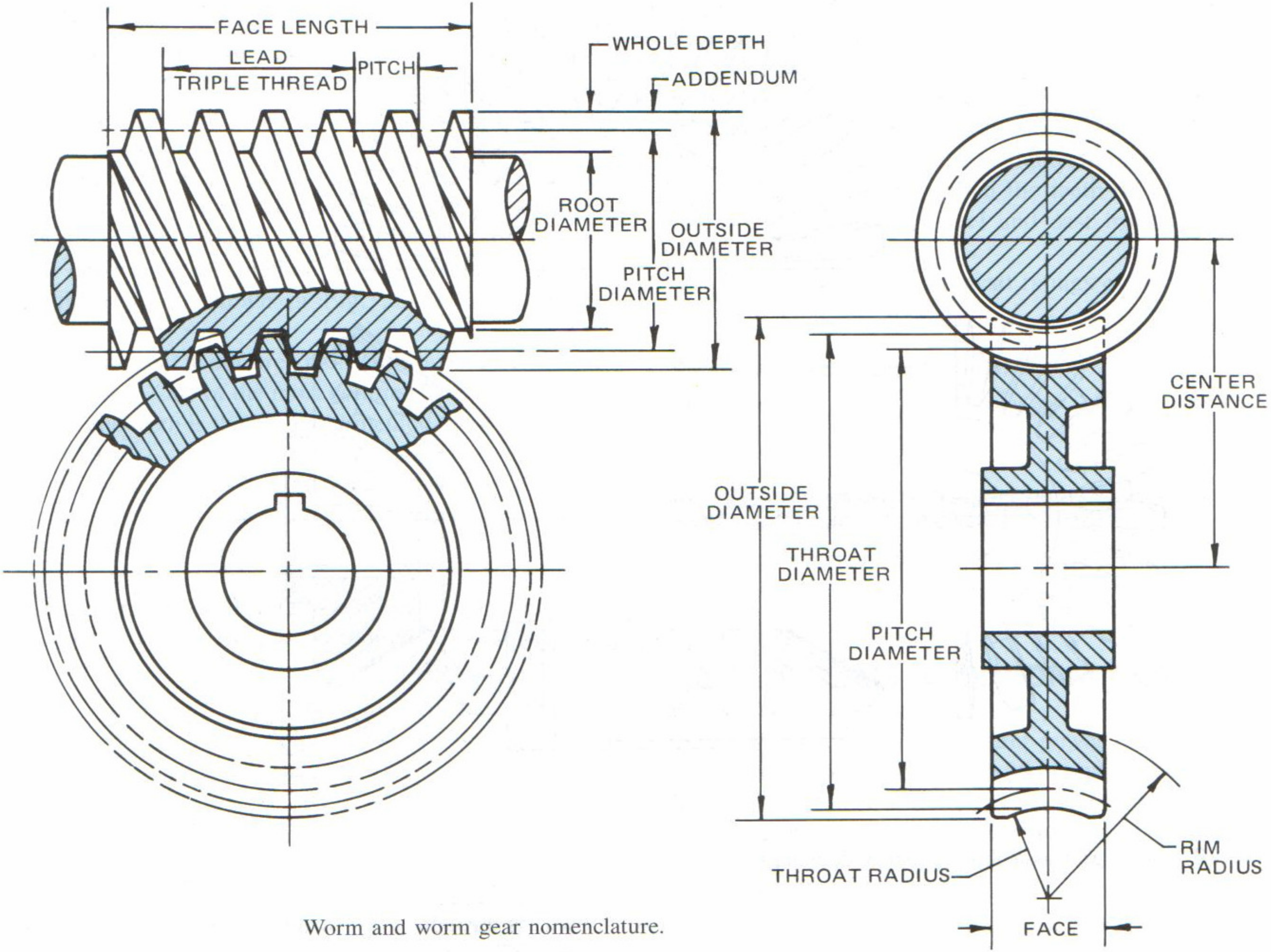
Since a single-thread worm in one revolution advances the worm gear only one tooth and space, a large reduction in velocity is obtained. Another feature of worm gearing is the high mechanical advantage acquired. The ratio of worm gear speed to the worm speed is the ratio between the number of teeth on the worm gear and the number of threads on the worm. A worm gear with 33 teeth and a worm with a multiple thread of three has a ratio of 11:1.

About 50:1 is the maximum ratio recommended. Since a single-thread worm has a low lead (or helix) angle, it is inefficient and consequently not used to transmit power. The lead angle should be between 25° and 45° for efficiency in transmitting power; as a result, multi-thread worms are used. The number of threads on a worm may vary from one to eight.





A Single-Enveloping Wormset, Consisting of a Worm and an Enveloping Worm Gear



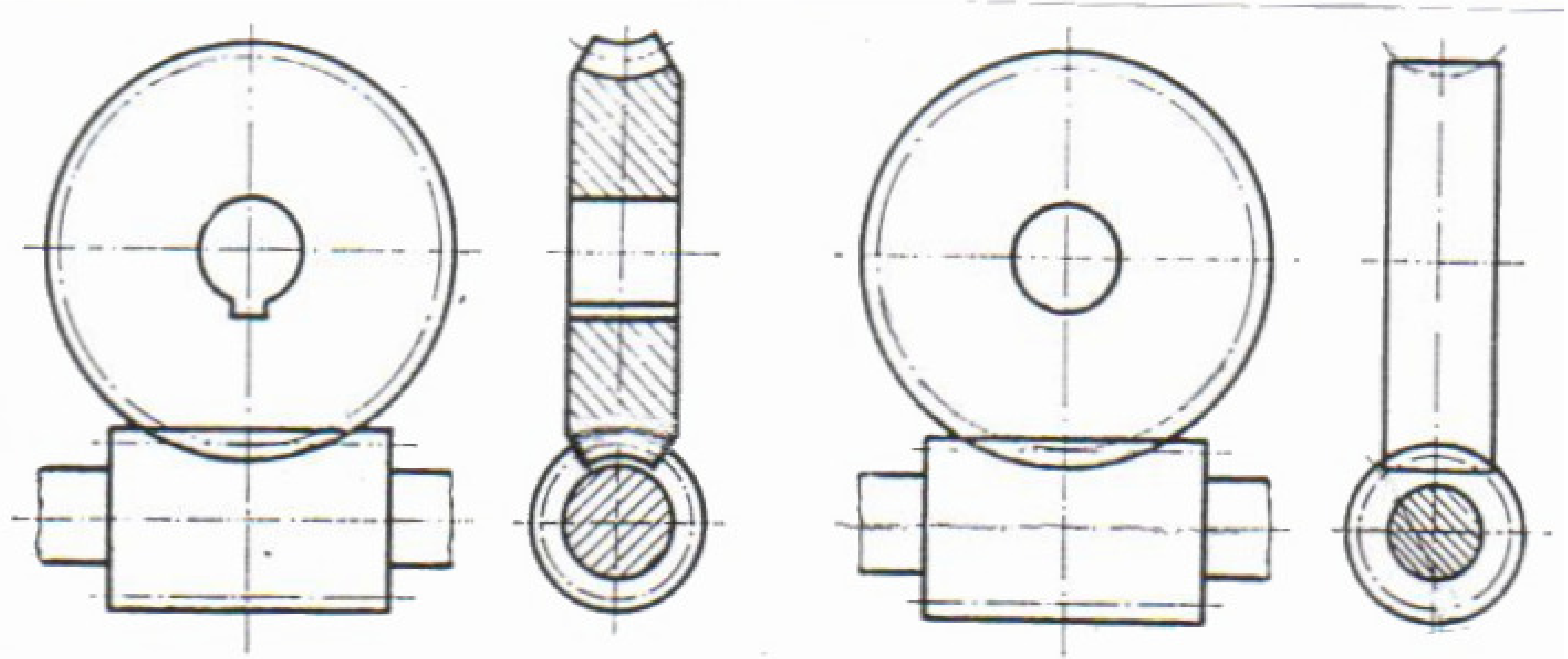
Worm and worm gear nomenclature.

Worm and worm gear formulas.

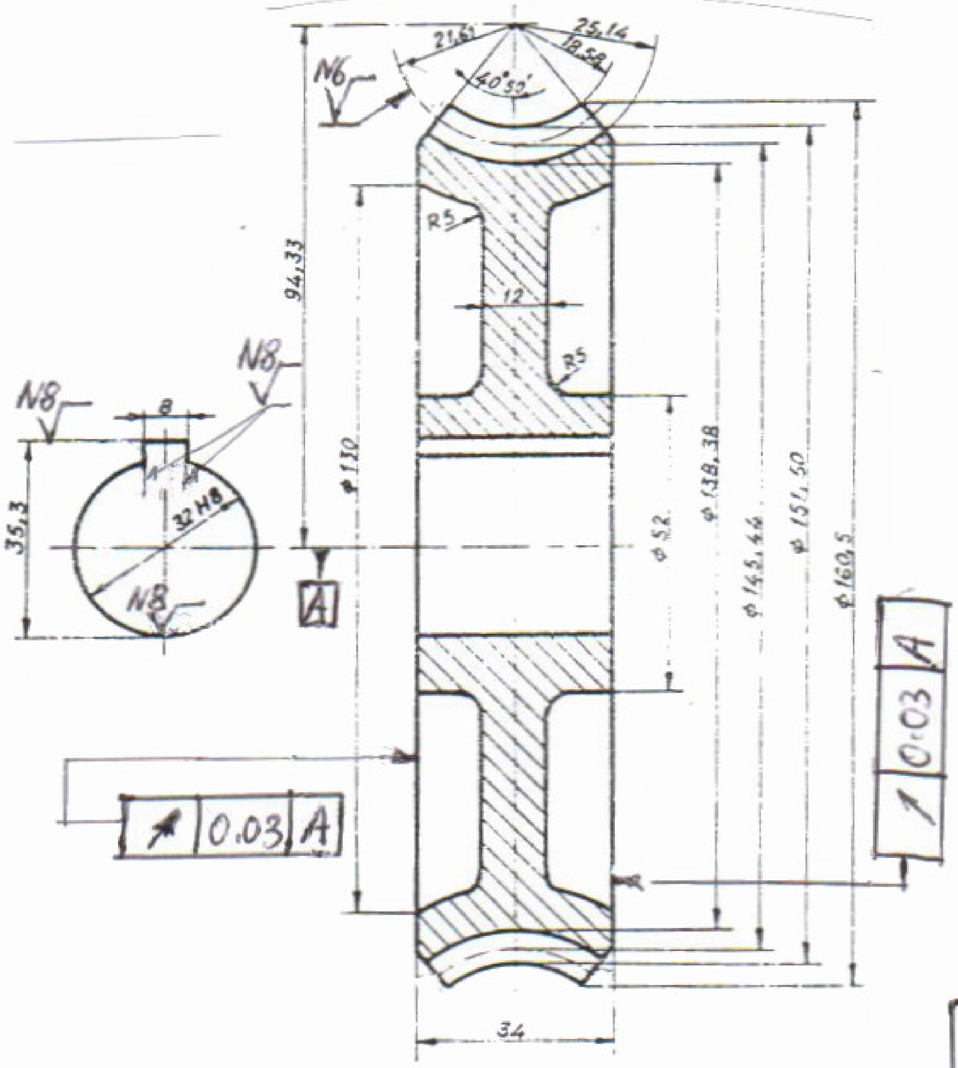
Term	Symbol	Formula	Definition
Pitch diameter of worm	PDw	$PDw = 2C - PDg$	
Pitch diameter of gear	PDg	$PDg = 2C - PDw$ or $\frac{NP}{\pi}$	
Pitch	P	$P = \frac{L}{T}$ $P = \frac{(2C - PDw) \times \pi}{N}$	The distance from one tooth to the corresponding point on the next tooth measured parallel to the worm axis. It is equal to the circular pitch on the worm gear
Lead	L	$L = \pi PDg \div R$ $L = P \times T$ $L = \text{Tan La} \times \pi PDw$	The distance the thread advances axially in one revolution of the worm
Threads	T	$T = \frac{L}{P}$	The number of threads of starts on worm; e.g., 2 for double thread, 3 for triple thread
Gear teeth	N	$N = \frac{\pi PDg}{-P}$	Number of teeth on worm gear
Ratio	R	$R = \frac{N}{T}$	Divide number of gear teeth by number of worm threads
Center distance	C	$C = \frac{PDw + PDg}{2}$	
Addendum	ADD	$ADD = 0.318P$	Single and double threads
		$ADD = 0.286P$	Triple and quadruple threads
Whole depth	WD	$WD = 0.686P$	Single and double threads
		$WD = 0.623P$	Triple and quadruple threads
Outside diameter, worm	ODw	$ODw = PDw + 2ADD$	
Outside diameter, gear	ODg	$ODg = TD + 0.4775P$	Single and double threads
		$ODg = TD + 0.3183P$	Triple and quadruple threads
Throat diameter	TD	$TD = PDg + 2ADD$	
Face width, gear	F	$F = 2.38P + .25$ (inch) $F = 2.38P + 6$ (metric)	Single and double threads
		$F = 2.15P + .20$ (inch) $F = 2.15P + 5$ (metric)	Triple and quadruple threads
Face length, worm	FL	$FL = 6 \times P$	
Lead angle	La	$\text{Tan La} = \frac{L}{PDw \times 3.1416}$	Divided by circumference of pitch diameter of worm. Quotient is tangent of lead angle
Throat radius	R_t	$R_t = \frac{PDw}{2} - ADD$	Subtract addendum from half of pitch diameter of worm
Rim radius	R_r	$R_r = \frac{PDw}{2 + P}$	

1	1,25	1,6	2	2,5	3,15	4	5	6,3	8	10	12,5	16	2

standard module series for wormsets (m_n)



N10 (N8, N6)

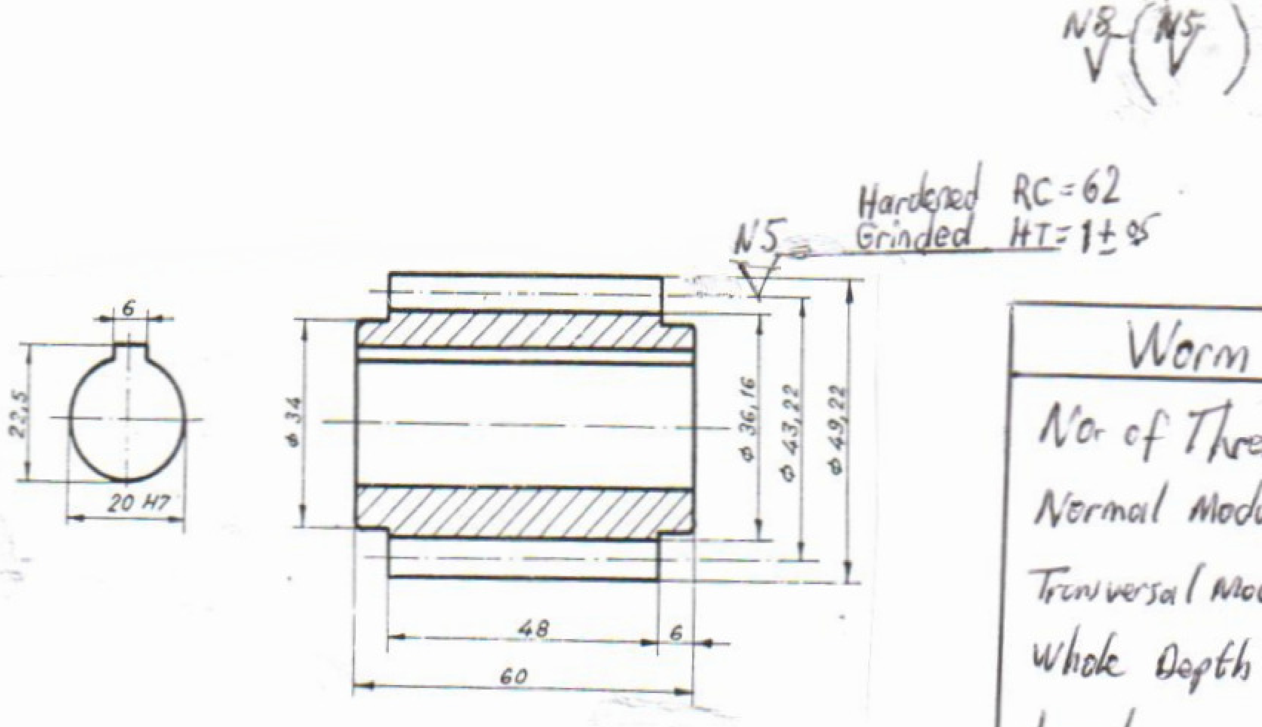


Worm Wheel Data

Number of Teeth z_2	48
No. of Threads	2 Left hand
Normal Module m_n	3.15
Transversal Module m_t	3.5
Whole Depth h	6.5
Lead Angle	8°
Pressure Angle	14.5°
Lead	19.03
Distance betwn. Axes	94.33 ± 0.06
Part No. of Meshing Worm	5215

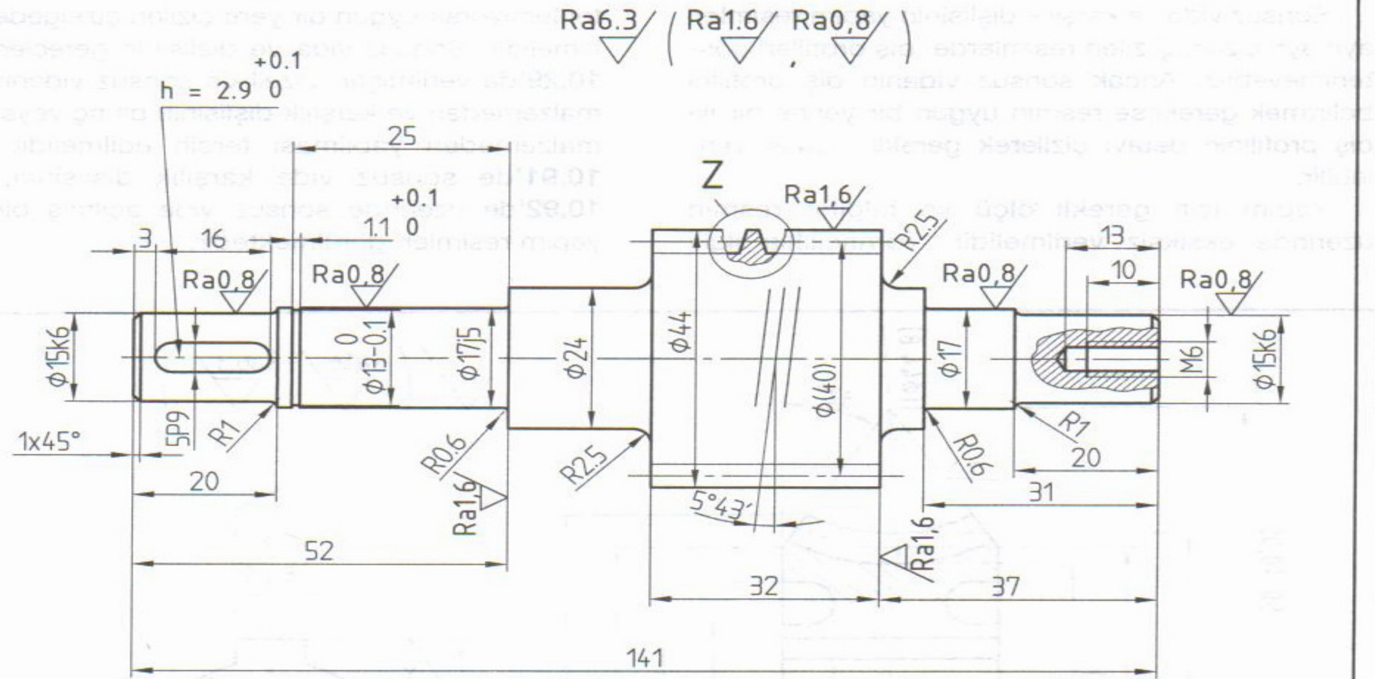
Title Block

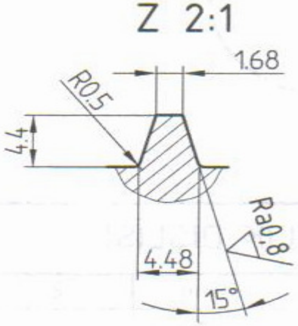
Example Detail Drawing of a Worm Wheel.



Worm Gear Data		
No. of Threads		2 left hand
Normal Module	m_n	3.15
Transversal Module	m_t	3.5
Whole Depth	h	6.5
Lead		19.03
Lead Angle		8°
Pressure Angle		14.5°
No. of Teeth of the meshing wheel	z_2	48
Part No. of the meshing wheel		5216

Detail Drawing of a Worm Gear.





Geneltolerans:
TS 1980'ne göre m

Ölçü	Sembol	Tolerans
⌀17	j5	+ 0,008 – 0,003
⌀15	k6	+ 0,012 + 0,001
5	P9	– 0,012 – 0,042

SONSUZ VİDA		
Modül	mn	4
Ağız sayısı ve yönü	z1	2-sol
Normal modül	mn	2
Diş derinliği	h	4.4
Diş profili açısı	γ	30°
Ağız adımı	pz	12.56
Helis açısı	α	5°43′
Karşılık dişli diş say.	z2	30
Eksenler arası	a	50

	Tarih	Adı	Imza	Sayı	Gereç
Çizen				C45	1
Kontrol					

Ölçek
1:1
2:1

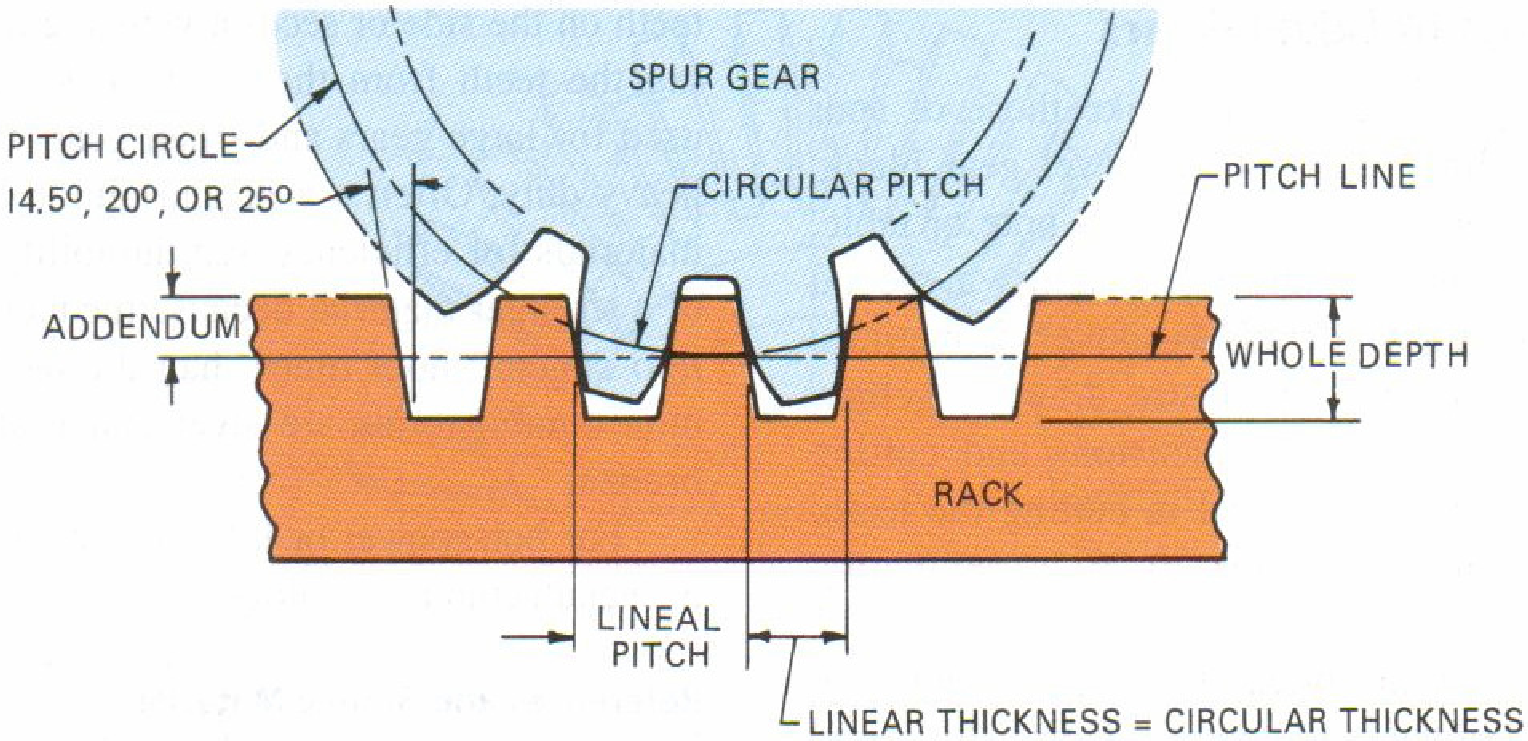
SONSUZ VİDALI MİL

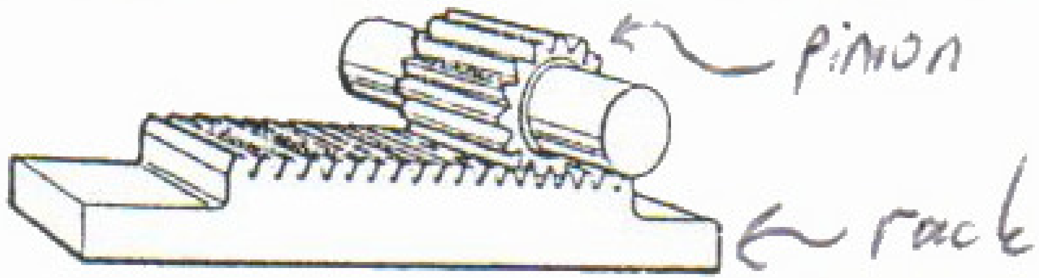
Resim Nr.

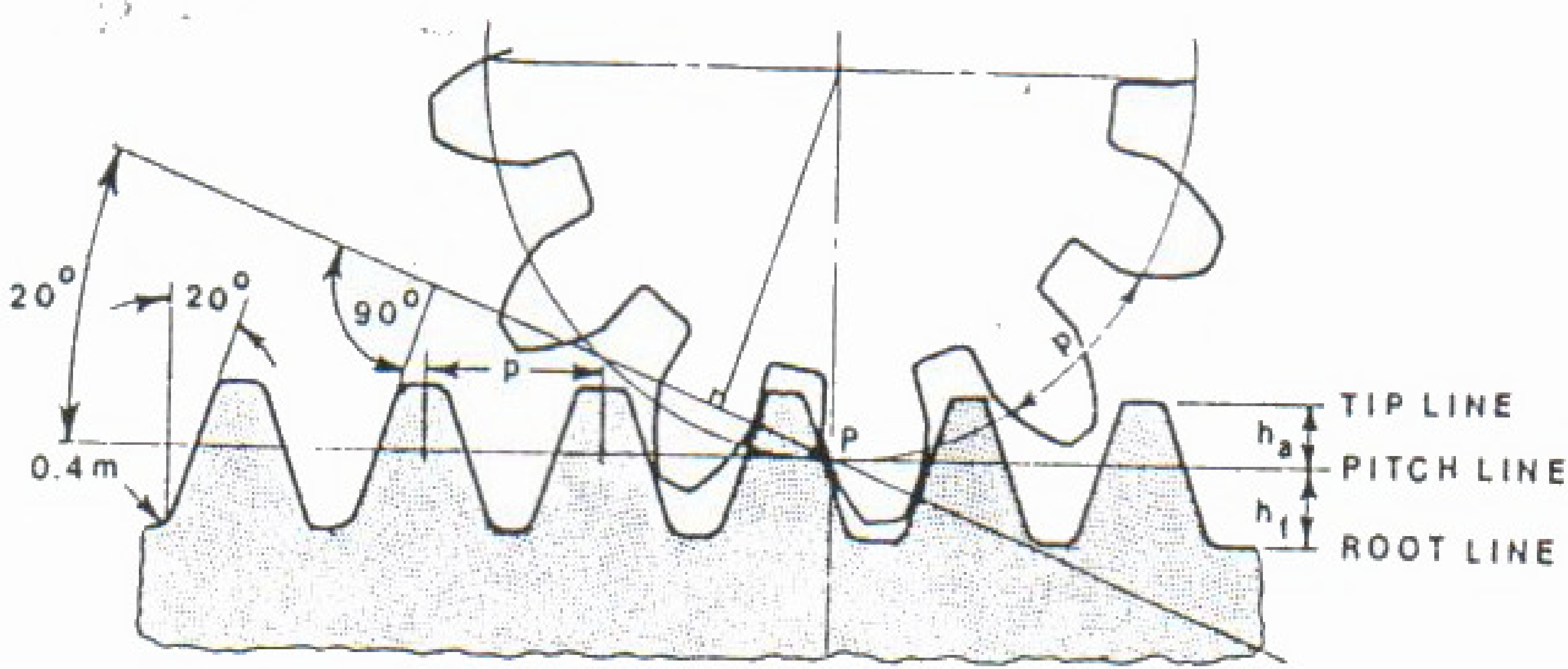
RACK AND PINION

A **rack** is a straight bar having teeth that engage the teeth on a spur gear   In theory, it is a spur gear having an infinite pitch diameter. Therefore, all circular dimensions become linear. The addendum, dedendum, and tooth thickness are the same as those of the mating spur gear. To draw the teeth on a rack, lay out the addendum and dedendum distances from the pitch line. Divide the pitch line into lineal pitch distances equal in size to the circular pitch on the gear. Divide each of these spaces in half to get the lineal thickness. Through these points draw the tooth faces at angles of 14.5°, 20°, or 25° from the vertical lines.

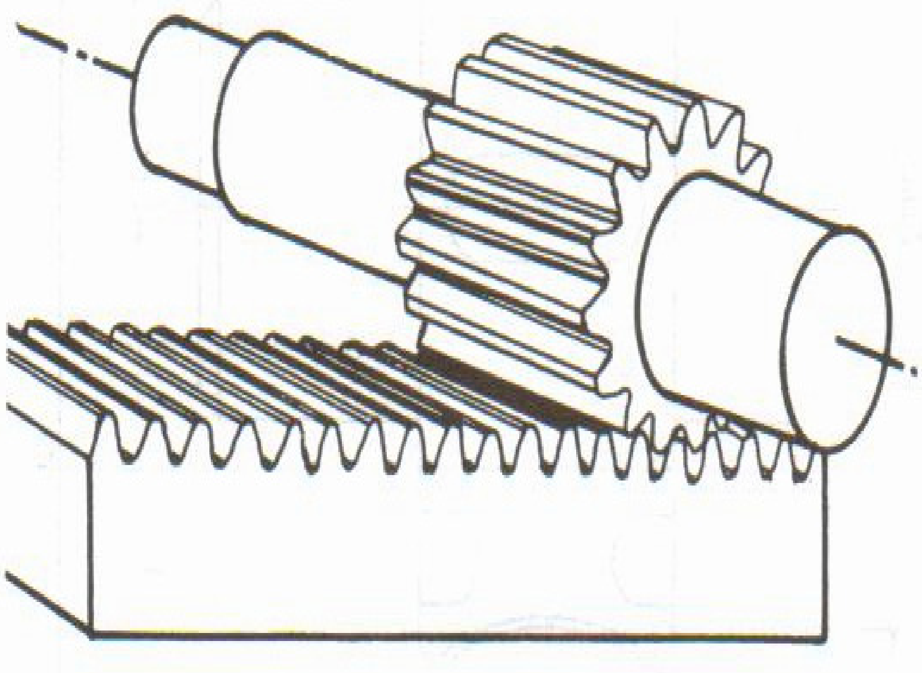
The specifications for the teeth of the rack are given in the same manner as for spur gears



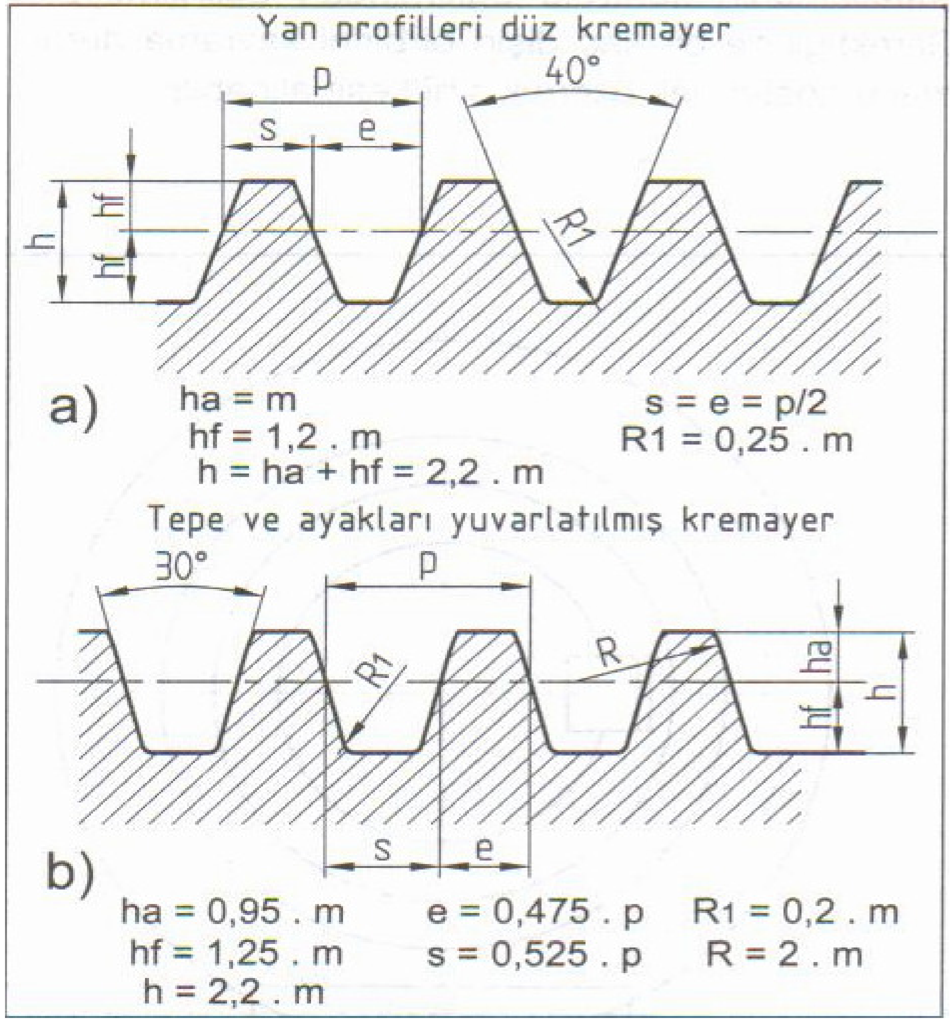


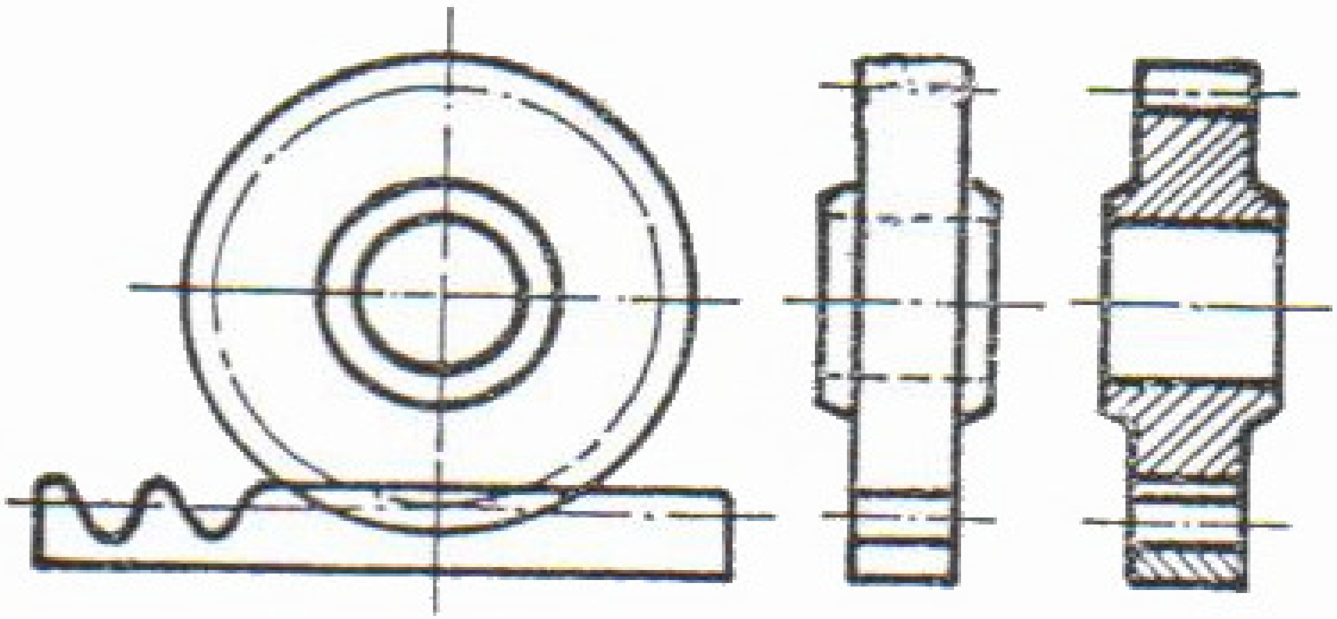


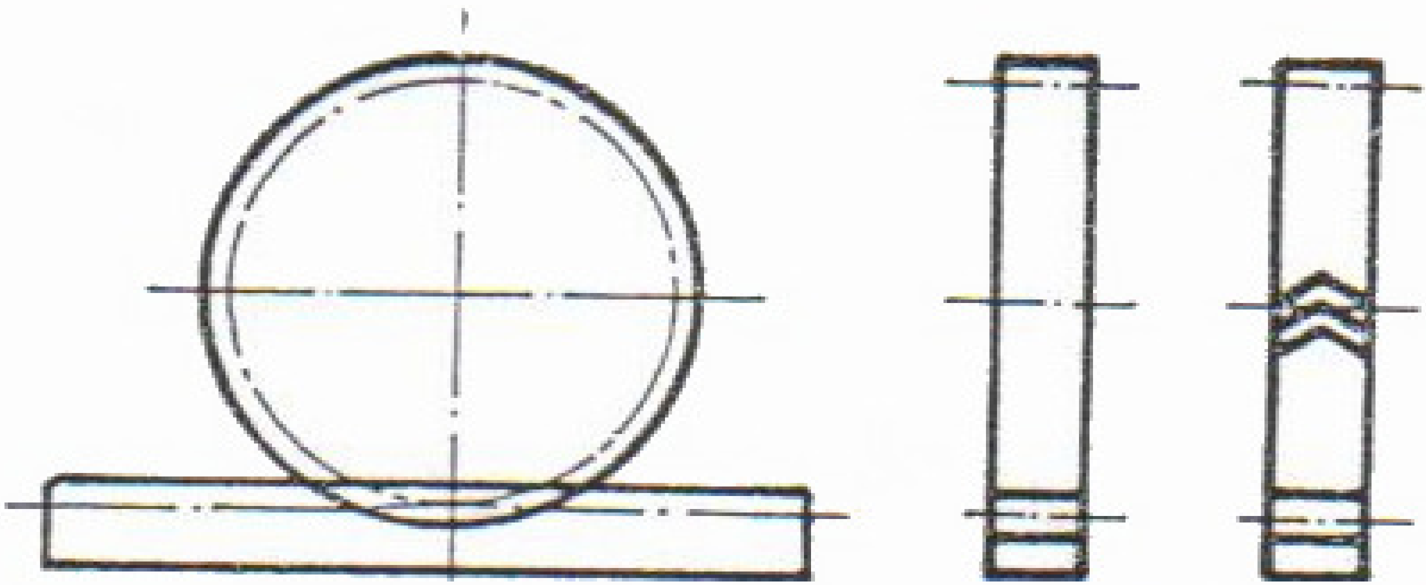
Involute rack

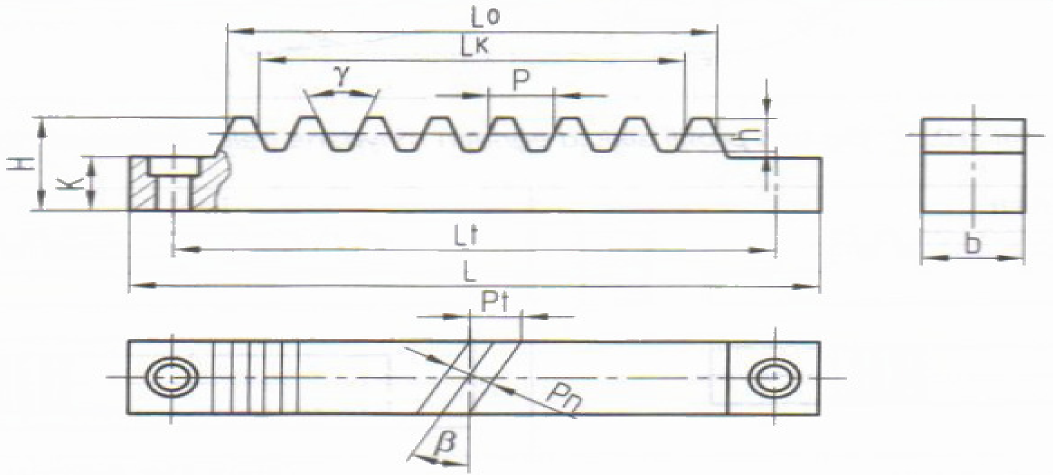


Kremayer dişli



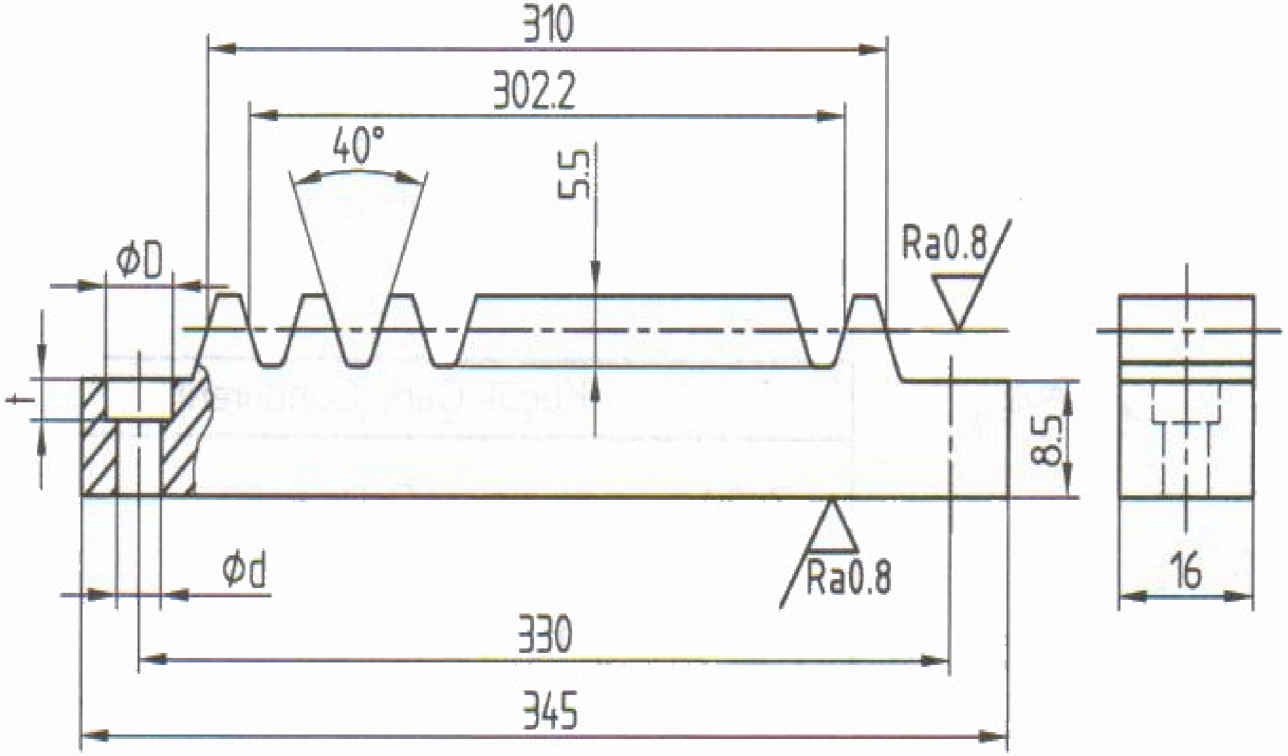






ADI	İŞARET	FORMÜL
Modül , Normal modül	m , m_n	$\frac{p}{\pi}$, $\frac{p_n}{\pi}$
Adım , Normal adım	p , p_n	$m.\pi$, $m_n.\pi$
Alın adımı	p_t	$m_t.\pi = \frac{m_n.\pi}{\cos\beta}$
Alın modülü	m_t	$\frac{m_n}{\cos\beta}$
Diş sayısı	z	$\frac{Lo}{p} + 0{,}5 = \frac{Lk}{p} + 1{,}5$
Diş derinliği	h	$2{,}167.m \approx 2{,}2.m$
Diş profil açısı	γ	30° , 40°
Eğim açısı (Helisel kremayerde)	β	$\cos\beta \; \frac{p_n}{p_t} = \frac{m_n}{m_t}$
Çalışma kursu boyu	Lk	$Lo - p = p(z-1{,}5)$
Kremayer boyu	Lo	$Lk + p = p(z-0{,}5)$
Delik merkezleri arası	Lt	$Lo + 1{,}2.b$
Çubuk boyu	L	$L_t + b$
Dişli genişliği	b	$\approx 2{,}5.p \sim 3.p$
Kremayer dişli yüksekliği	H	$3.h$

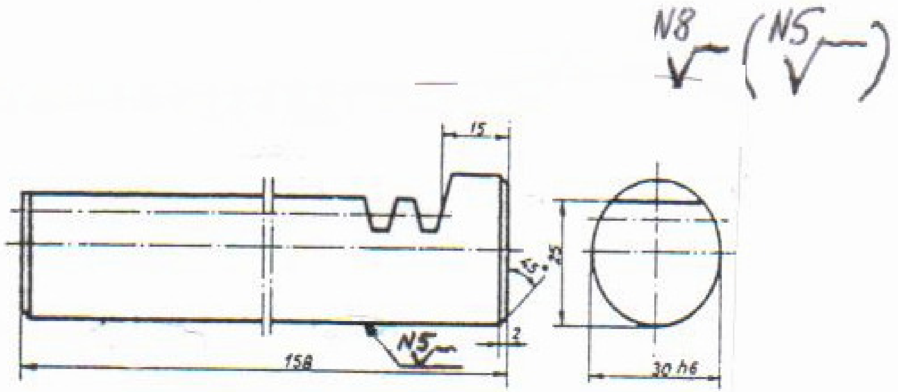
Kremayer dişlisinin elemanları ve formülleri



DÜZ KRAMAYER DİŞLİ		
Modül	m	2.5
Diş sayısı	z	40
Adım	p	7.85
Diş yüksekliği	h	5.416
Diş profili	γ	40° Trapez
Eş dişli diş sayısı	z_2	26

	Tarih	Adı	İmza	Sayı	Gereç
Çizen					Fe60
Kontrol					

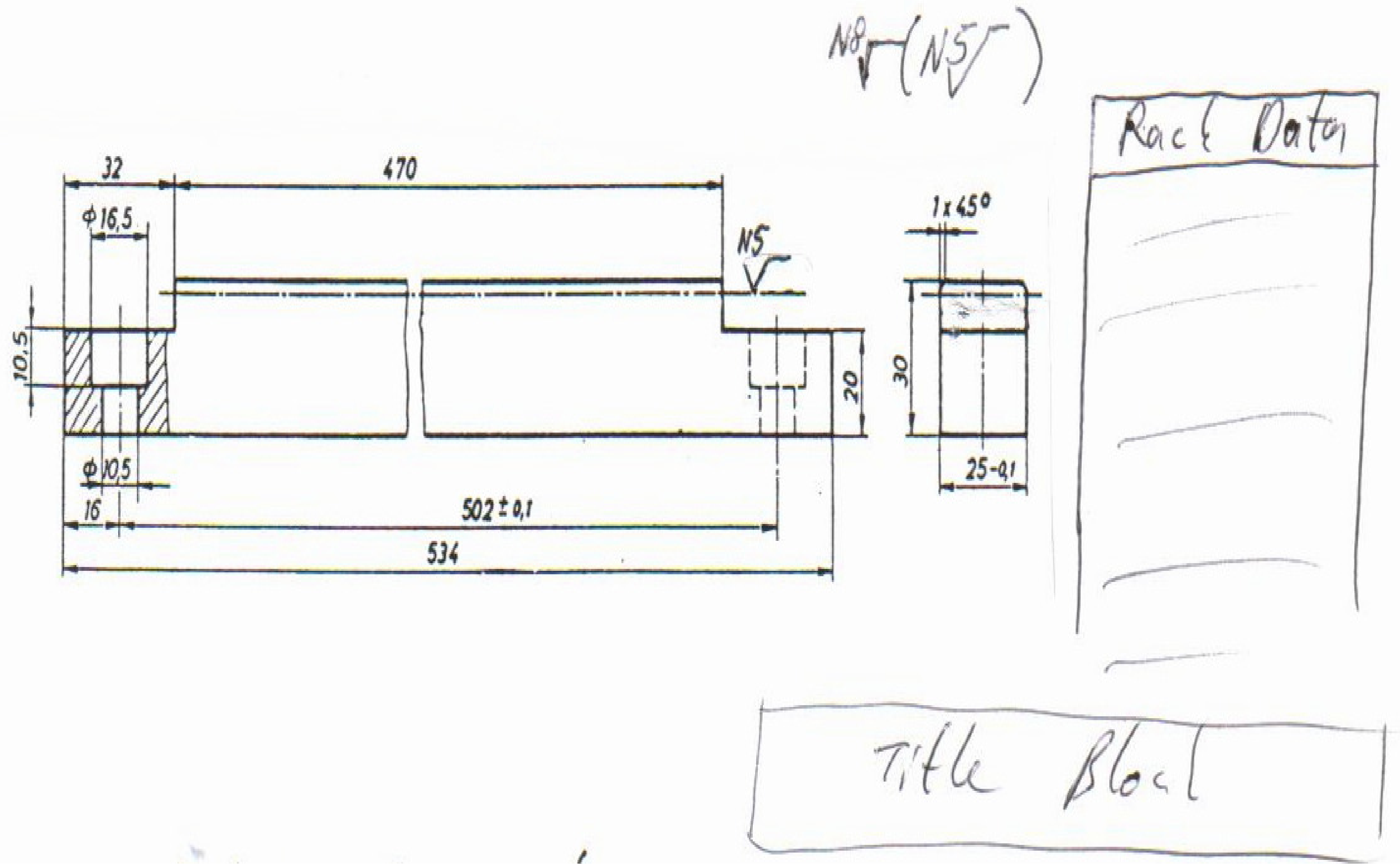
Ölçek		Resim Nr.
1:1	DÜZ KREMAYER DİŞLİ	

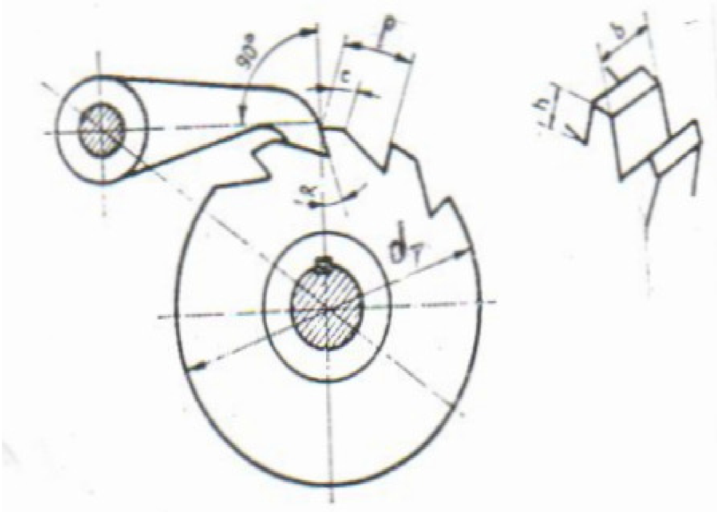


Rack Data

Number of teeth	z_1	15
Module	m	3
Pitch	p	9.42
Tooth Form		14.5° involute
whole depth	h	6.5
mating gear part no:		1504
Number of teeth of mating gear	z_2	26

Title Block


N8 (N5)
Rack Data
32
470
ϕ16,5
N5
10,5
ϕ10,5
16
502 ± 0,1
534
20
1 x 45°
30
25-0,1
Title Block



Pitch, p

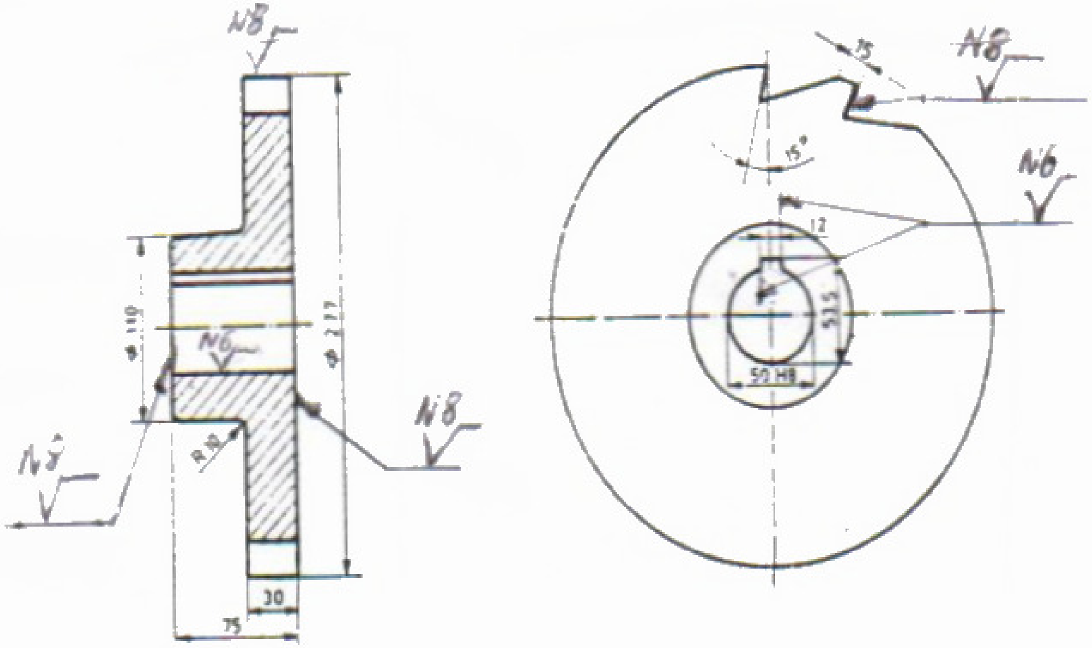
Tip circle diameter, $d_T = \dfrac{pz}{\pi}$

z: number of teeth

$\alpha = 15^\circ$; $c = 0.25 * p$, $h = 0.35\, p$

Face width, $b = (0.3 \sim 0.5)\, p$ (material : steel)

$b = (0.5 \sim 1)\, p$ (material : cast iron)



N10 (N8, N6)

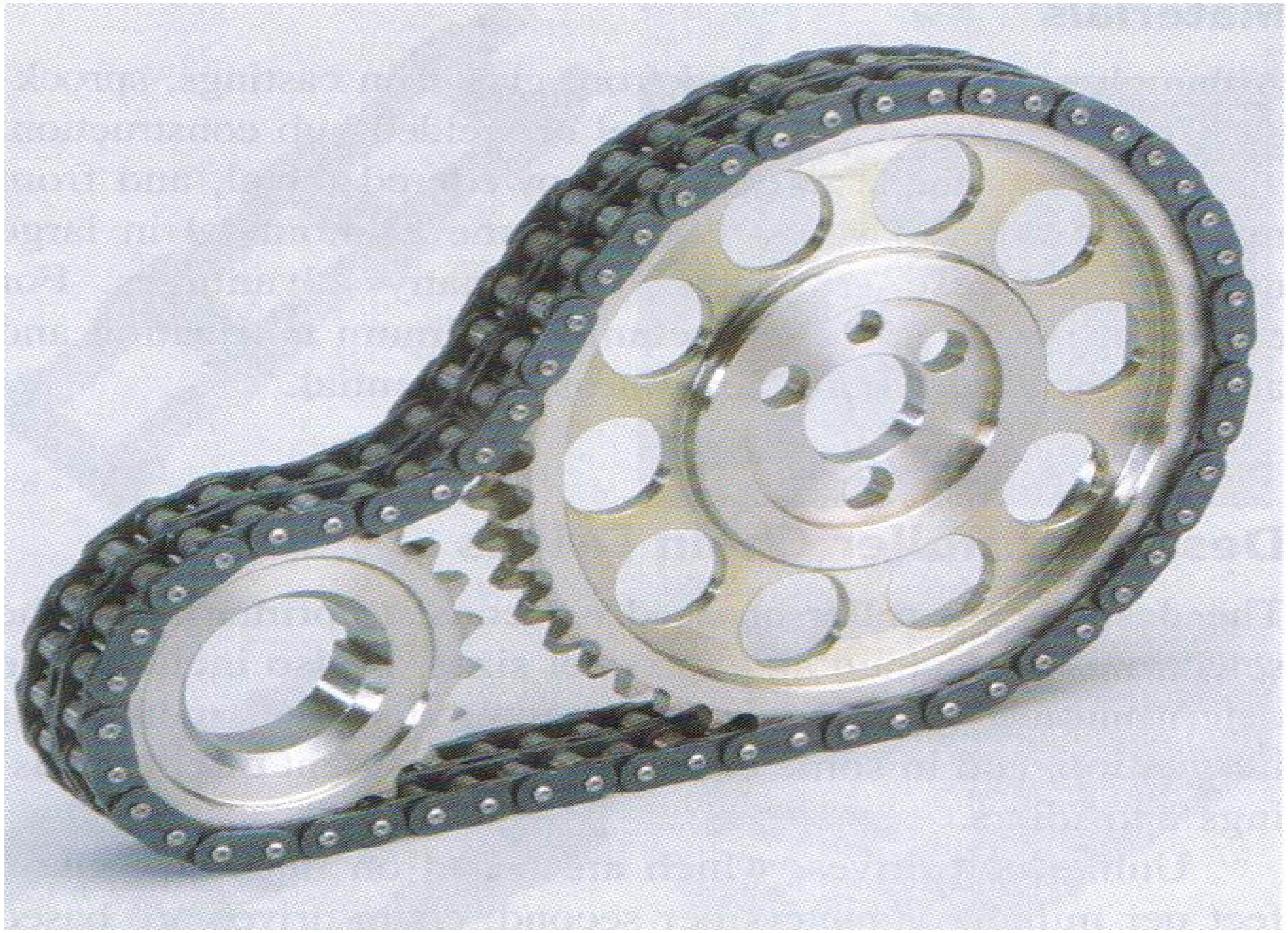
Data		
No. of teeth	z	15
Pitch	p	58
Teeth depth	h	20

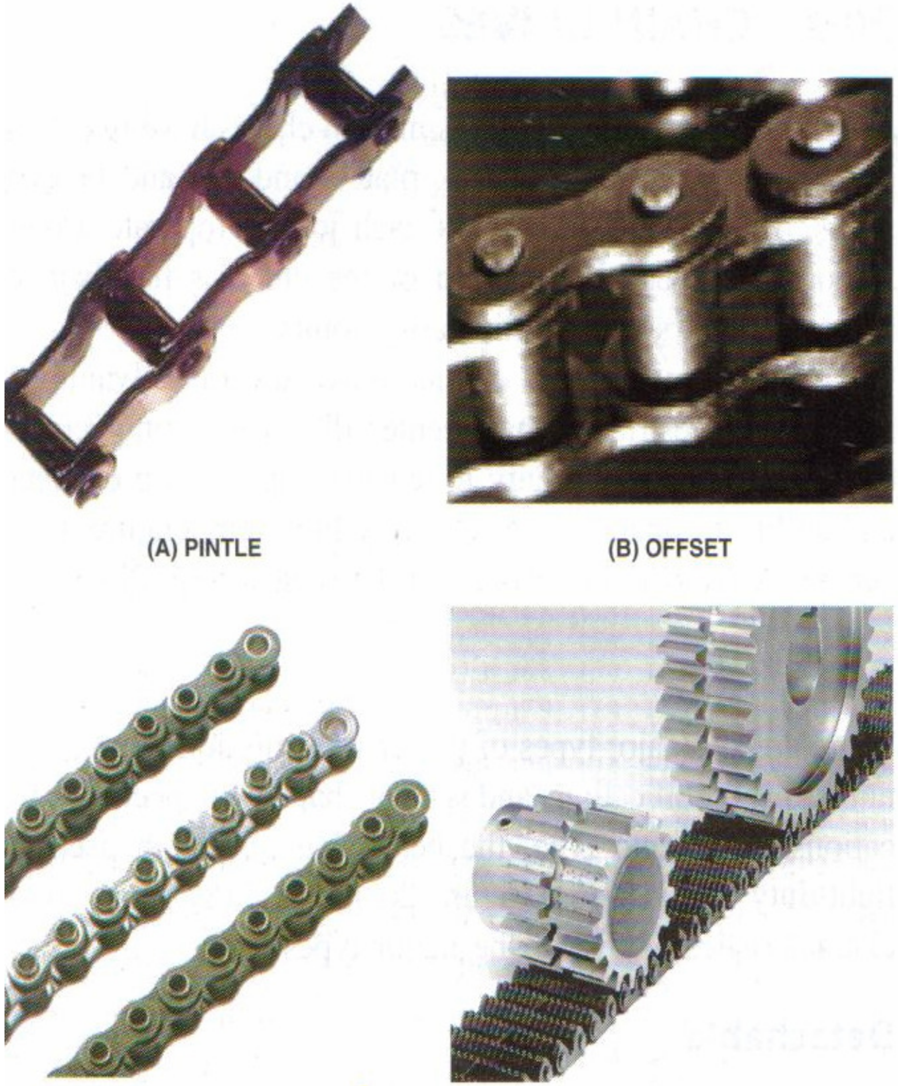
Detail drawing of a locking gear

CHAIN DRIVES

Nearly all types of power-transmission chains have two basic components: side bars or link plates, and pin and bushing joints. The chain articulates at each joint to operate around a toothed sprocket. The **pitch** of the chain is the distance between centers of the articulating joints.

Power-transmission chains have several advantages: relatively unrestricted shaft center distances, compactness, ease of assembly, elasticity in tension with no slip or creep, and ability to operate in a relatively high-temperature atmosphere.



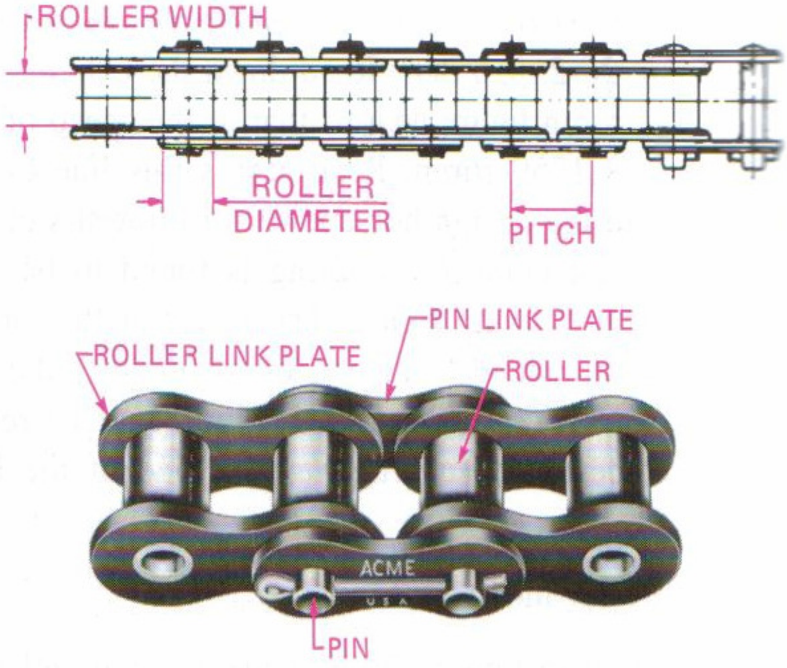


(A) PINTLE

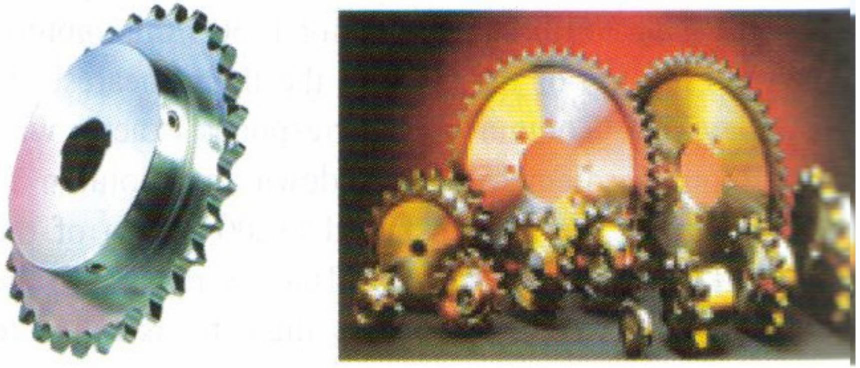
(B) OFFSET

(C) ROLLER

(D) INVERTED TOOTH (SILENT)



(A) CHAIN TERMINOLOGY

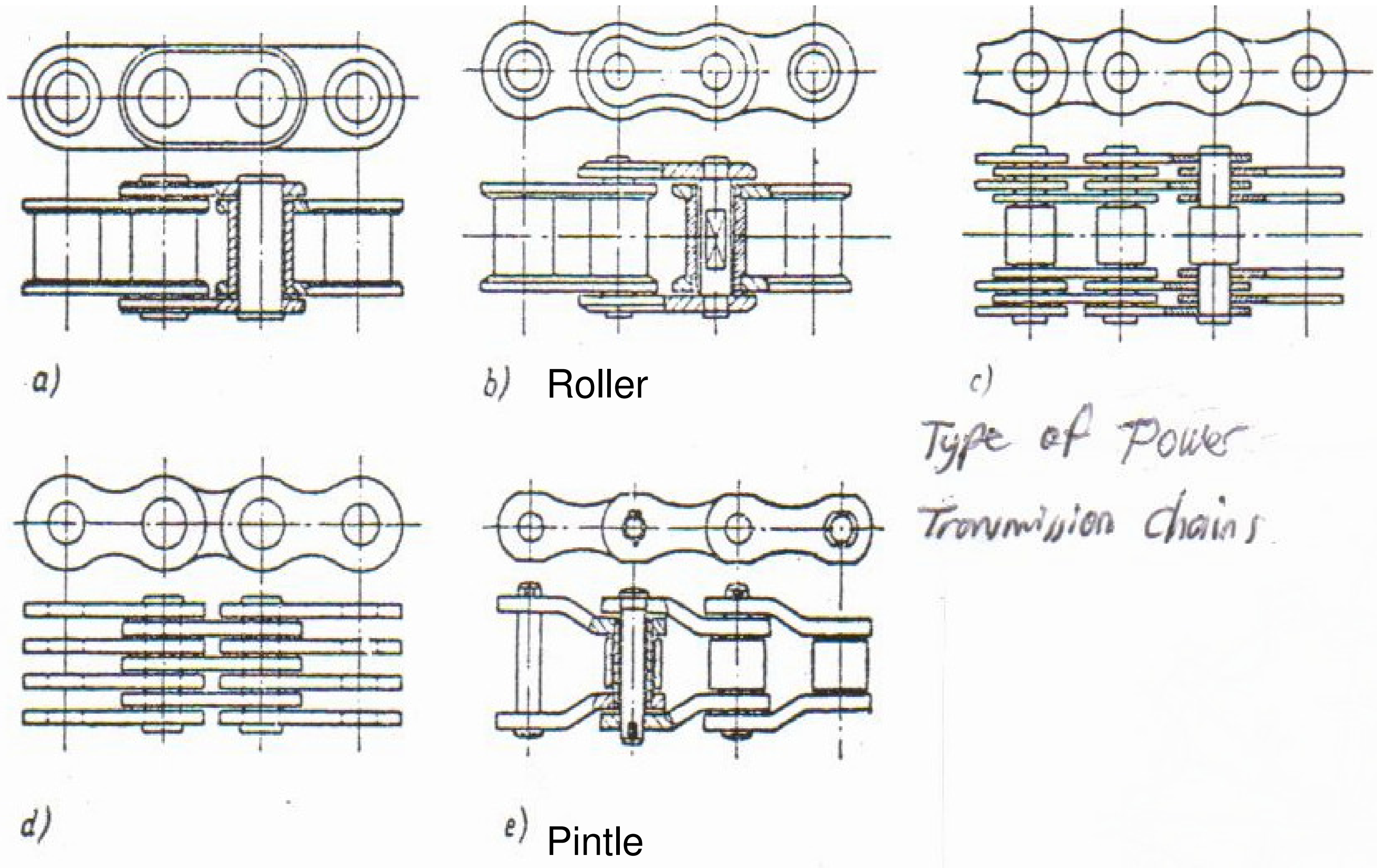


SINGLE STEEL

DOUBLE AND SINGLE STEEL

(B) SPROCKETS

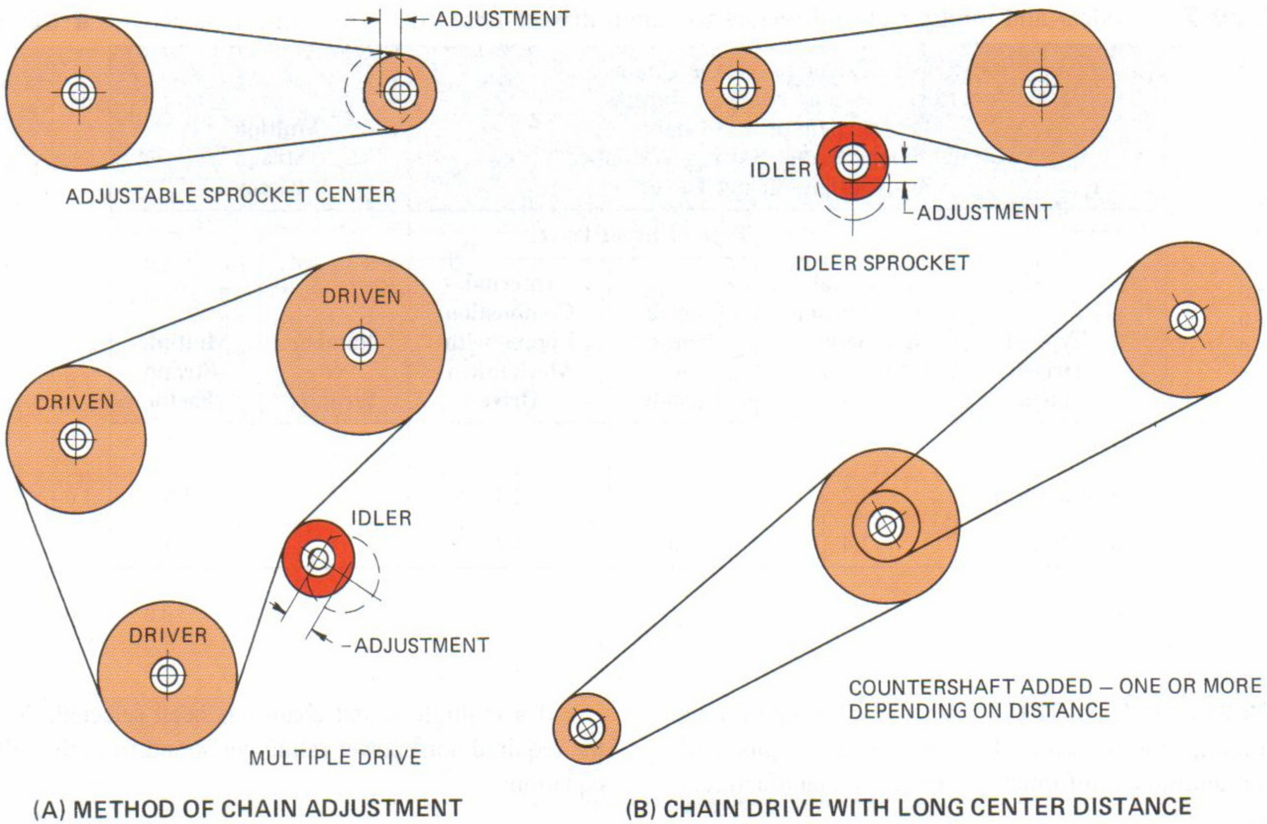
Roller chain terminology and sprockets.



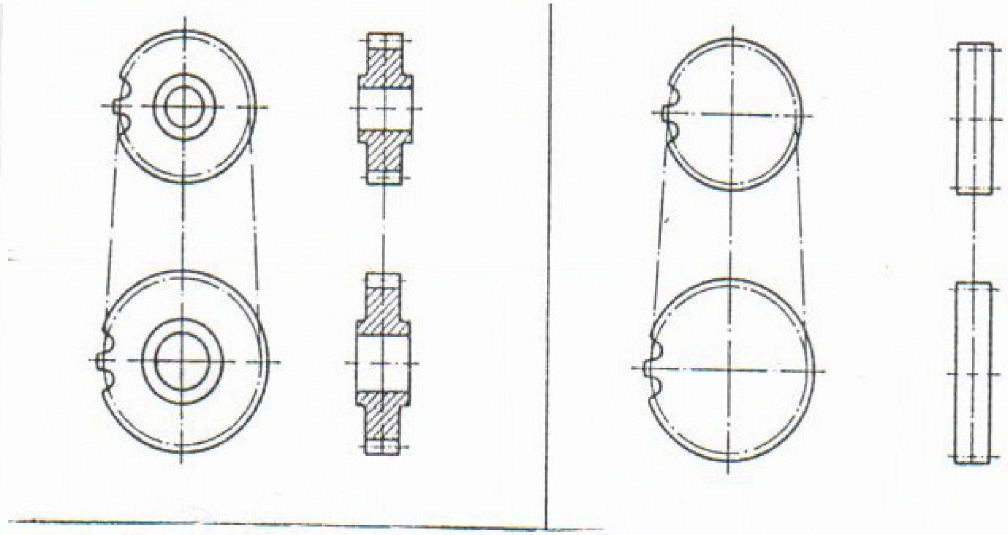
Type of Power Transmission chains

Tentative selection factors for ction factors for chain drives.

Chain fpm	Speed m/s	Power hp	kW	Type of Chain
350	1.0	20	15	Detachable
450	2.2	40	30	Pintle
1000	5	250	190	Offset-sidebar
2500	12.5	1500	1100	Roller
4000	20	2500	1850	Silent

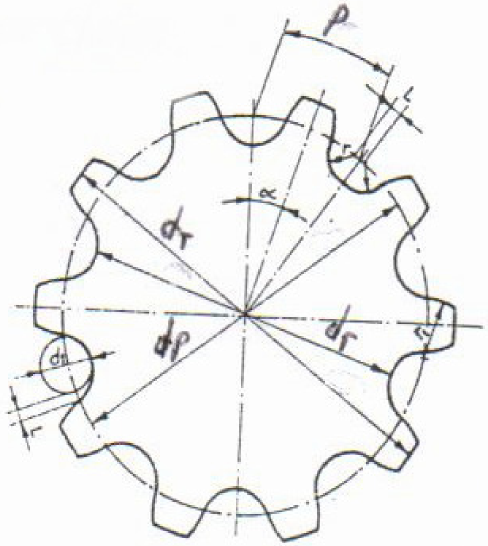


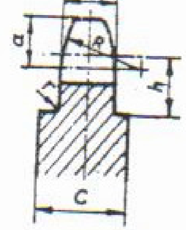
Chain drives.



$R = 1.063\,P$, $a = 0.5\,P$, $r_{max} = 0.04\,P$

For the values of B and H see Table ☆





P	C [mm]
3/8"	8
1/2"	9.5
5/8	10.5
1"	12.7

Chain Gear Calculations (for type a&b)

d_1 : Roller diameter (From Table ☆)

$r_1 = P - d_1/2$, $r = d_1/2$

$L = 0.004\,P + 0.15$ [mm]

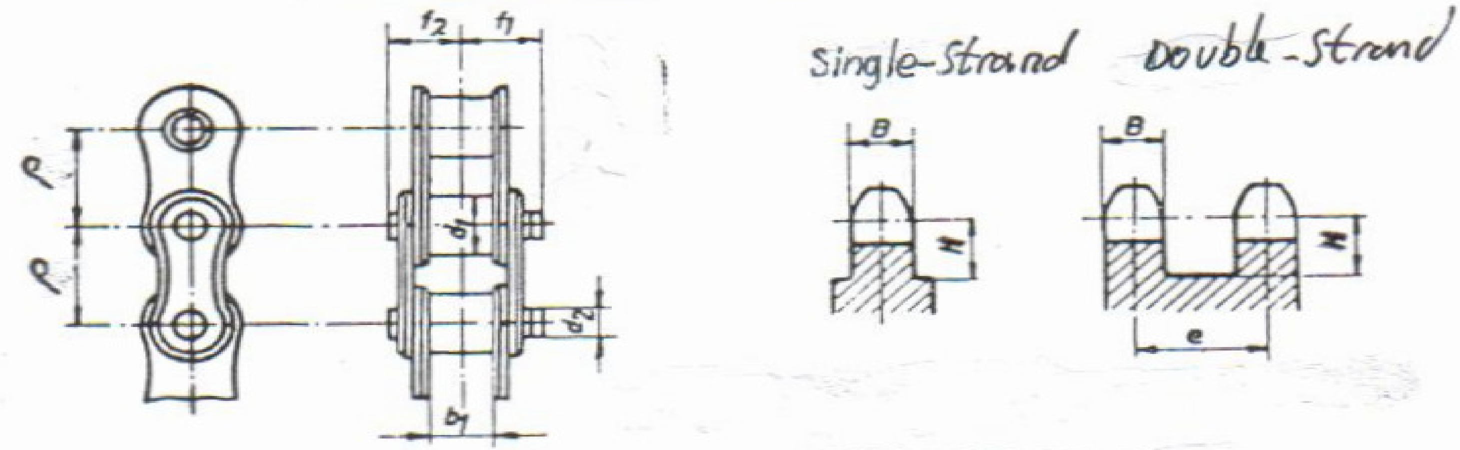
$D_p = P/[\sin(180°/Z)]$

$d_T = d_p + (0.45 \sim 0.55\, d_1)$ (if $Z \leq 16$)

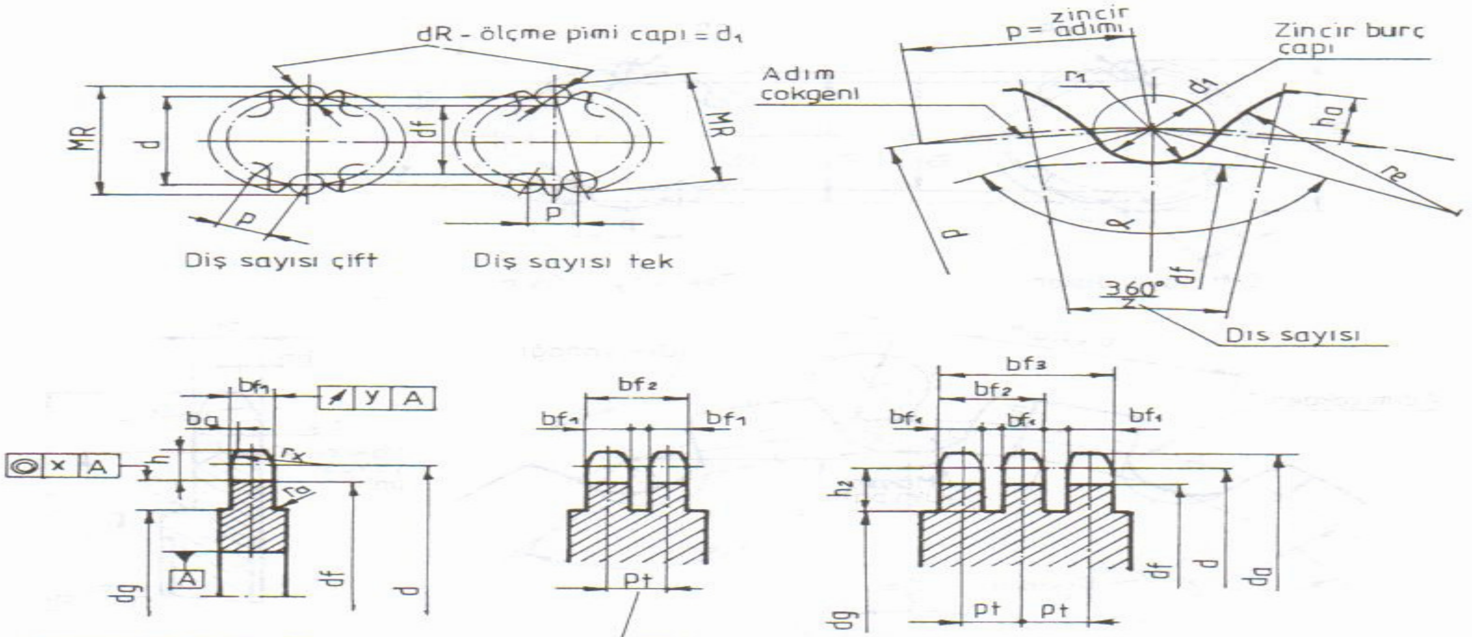
$d_T = d_p + (0.6 \sim 0.8\, d_1)$ (if $Z > 16$)

$d_r = d_p - (d_1 + 0.1)$ [mm]

Table A | Roller-Chain and Sprocket Data.



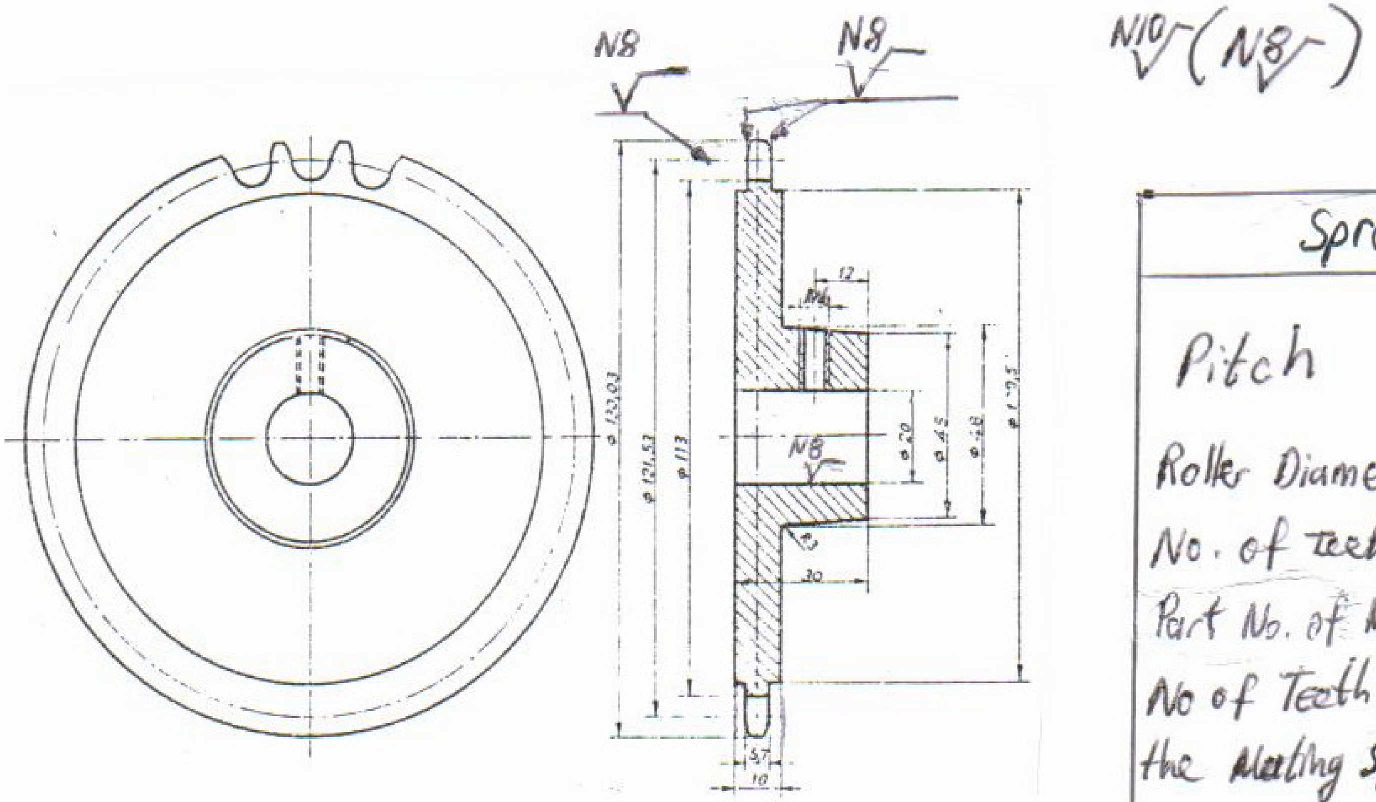
Single-Strand Roller Chain										Tooth Profile				
pitch, P		b_1		d_1	f1 + f2 mm		Tensile strength	Weight per meter	Single		Double			
Inc	mm	Inc	mm	mm	f1	f2	kg_f	kg_f	B	H	B	e	H
–	6	–	2,8	4	4,7	3,7	250	0,18	2,5	3	–	–	–
(1/16)	8	(1/8)	3	5	5,45	4,5	400	0,24	2,7	4	2,7	5,64	4
3/8	9,52	1/8 7/32	3,2 5,72	6 6,35	7,0 7,1	4,8 6,8	600 860	0,25 0,45	2,8 5,2	4,3 4,8	– 5,2	– 10,24	– 4,8
1/2	12,7	– 1/4 5/16	5,2 6,35 7,75	8,5	8,8 9,4 10,1	7,2 7,7 8,4	1600	0,65 0,70 0,80	4,6 5,7 7,0	6,3 6 6,3	– – 7,0	– – 13,92	– – 6,3
5/8	15,88	1/4 3/8	6,35 9,52	10,2	9,8 11,6	8,2 9,7	2000	0,80 0,96	5,7 8,8	7,8 7,8	– 8,8	– 16,59	– 7,8
3/4	19,05	(7/16)	11,7	12,07	13,2	11,3	2600	1,50	10,6	9,2	10,6	19,46	9,2
1	25,4	–	17	15,9	24,5	18	4200	2,90	15,5	11,5	15,5	31,88	11,5



x = 0,0008 df + 0,08 mm
y = 0,0009 df + 0,08 mm

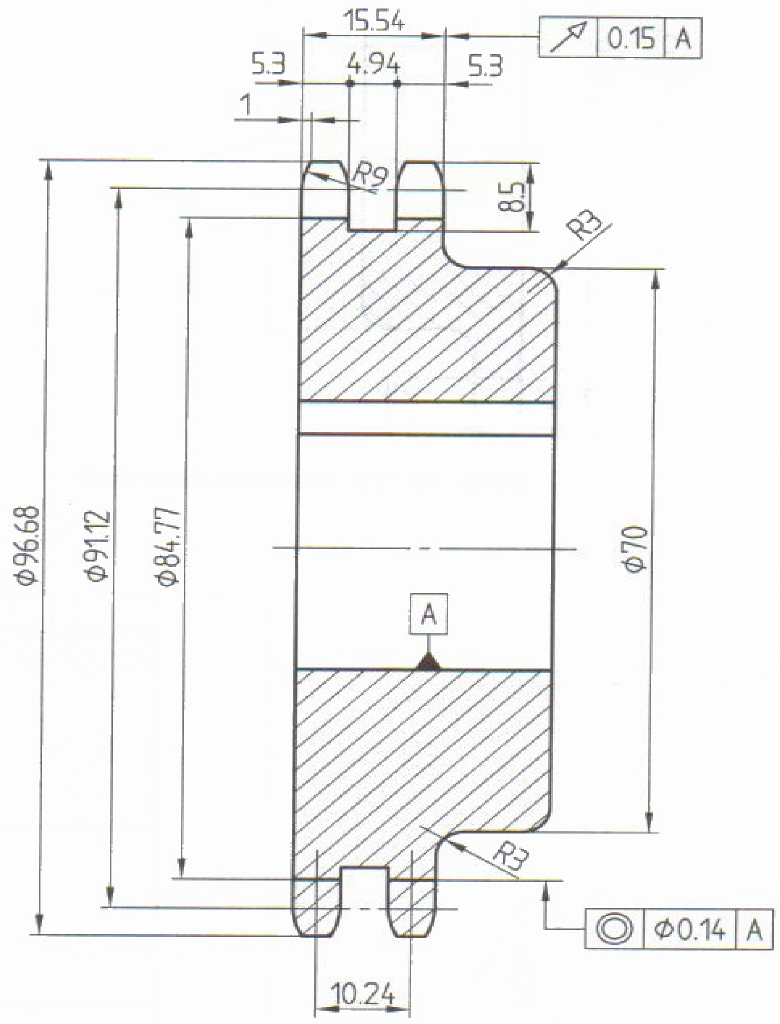
ADI	İşaret	FORMÜLLER	
		Diş boşluğu en küçük profil	Diş boşluğu en büyük profil
Zincir burç oturma yarıçapı	r_1	$= 0,505\, d_1$	$r_2 = (0,505\, d_1 + 0,069 \sqrt[3]{d_1})$
Zincir burç oturma açısı	α_1	$= 140° - \frac{90°}{z}$	$\alpha_2 = 120° - \frac{90°}{z}$
Diş yanak yarıçapı	r_{e1}	$= 0,12\, d_1 (z + 2)$	$r_{e2} = 0,008 . d_1 (z^2 + 180)$
Diş üstü çapı	$da_{max.}$	$d + 1,25\, p - d_1$	
	da_{min}	$d + p\,(1 - \frac{1.6}{z}) - d_1$	
Adım çokgeni üzerinde kalan diş yüksekliği	$ha_{max.}$	$0,625\, p - 0,5\, d_1 + \frac{0,8\, p}{z}$	
	$ha_{min.}$	$0,5 . (p - d_1)$	
Dişli, diş genişliği	bf_1	$0,93 . b_1$ Bir sıralı zincir dişlilerde	
	bf_1	$0,91 . b_1$ İki ve üç sıralı zincir dişlilerde	
	bf_1	$0,88 . b_1$ Üç ve daha fazla sıralı zincir dişlilerde	
Dişler üzerinde ölçülen genişlik	bf_2, bf_3	(Sıra sayısı − 1) pt + bf_1	
Diş yan yarıçapı	r_x	= (en az) ≈ p	
Diş yan boşluğu	b_a	(en az) ≈ 0,1 p en çok ≈ 1,5 p	
Ara yarı çapı	r_a	gerçek ara yarıçapı	
Mutlak max. göbek çapı	dg	$p . \cot \frac{180°}{z} - 1,05\, h_2 - 1 - 2 . r_a$	
Adım dairesi çapı (Bölüm dairesi çapı)	d	$= \frac{p}{\sin \frac{180°}{z}}$	
Pimler üzerinden alınan ölçü	MR	Çift diş sayıları için, $MR = d + dR$, Tek dişlerde, $MR = d . \cos \frac{90°}{z} + dR$	

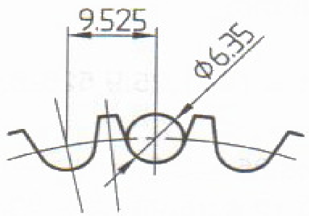
Güç iletiminde kullanılan makaralı ve burçlu zincirler için zincir dişlerin elemanları ve formülleri (TS3578)

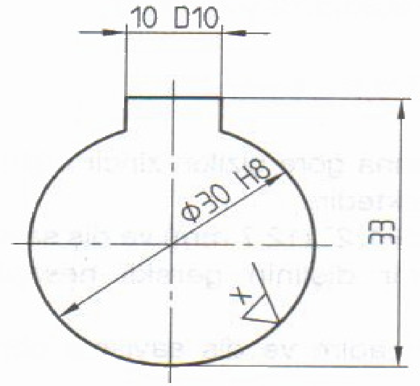


Sprocket		Data
Pitch	p	12.7
Roller Diameter	d_1	8.5
No. of teeth	Z_1	30
Part No. of Mating Sproc.		125
No of Teeth of the Mating Sproc.	Z_2	60

Detail Drawing of a Sprocket







ZİNCİR DİŞLİ		
Adım	p	3/8"=9.525
Diş sayısı	z_1	30
Makara çapı		6.35
Eş dişli diş sayısı	z_2	57
Eş dişli dış Nr.		
Eksenler arası	a	300

	Tarih	Adı	İmza	Sayı	Gereç	
Çizen				1	Fe 60	
Kontrol						
Ölçek						Resim Nr.