Mansoura Engineering Journal

Volume 18 | Issue 2 Article 7

6-1-2021

CAD for Gears - Part 2: Bevel Gears.

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CAD FOR GEARS

Part 2

Bevel Gears

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إستخدام الحاسب الآلي في تصميم المسننات

الجزء الثاني

المسننات المخروطيه

الخلاصة : الغرض من هذا البحث هو إنشاء برنامج (*Software*) يحتوى على خطوات كامله لتصميم ورسم المسننات المخروطية (العدله-المائله-الحلزونيه المتماثله-الحلزونيه). هذا البرنامج يمكن إستخدامه وتشغيله على أى حاسب آلى شخصى. العناصر الرئيسيه للتصميم والتى يتكون منها هذا البرنامج هى : البيانات الداخلة، نوع المسنن المخروطى المستخدم، شكل ونظام السنة طبقا للمواصفات (الدولية-الأمريكية-البريطانية-الألمانية-شركة جلسون)، الموديول، عدد الأسنان الإفتراضية، أقل عدد من الأسنان لتلاشى التداخل بين الأسنان، تصحيح بروفيل السنة، الزاوية بين المحورين، نسبة التغطية، عرض المسنن، زاوية حلزون السنة، تحليل القوى على الأسنان، السرعة الإنزلاقية والكفاءة، نوع الخام المستخدم، معادلات التصميم طبقا لمقاومة الإنحناء للسنة (لويس المعدلة-باكنجهام-الجمعية الأمريكية لإنتاج المسننات-شركة جلسون)، معادلات التصميم طبقا لإنهيار سطح السنة (إجهاد التماس-حمل التآكل لباكنجهام-الجمعية الأمريكية لإنتاج المسننات-شركة جلسون-النقر-درجة حرارة التماس) وأيضا معادلات سعة التحميل لمقاومة الإنحناء وإنهيار سطح السنة والمسنن (النظام الدولى-الجمعية الأمريكية لإنتاج المسننات-النظام البريطانى-نظام شركة جلسون مع إعتبار الحمل ثابت أو متغير). تم إختيار وإستنتاج مجموعة من المعادلات والعلاقات العملية لإنشاء ورسم المسننات المخروطية (مسنن مخروطى مخلق فى العمود-مصمت-له جدار كامل مصمت-له جدار بة ثقوب-له أعصاب-مسنن يتكون من اجزاء مربوطة مع بعضها).

إن تشغيل هذا البرنامج يعطى النتائج التالية: هندسية ومواصفات المسننات (القائد والمنقاد) كينماتيكية المسننات والأحمال والاجهادات على السنة والمسنن والرسم الانشائى الكامل للمسننات (القائد والمنقاد) وذلك طبقا للبيانات الداخلة. تم عمل أمثلة مختلفة ومتنوعة لبيان مدى قدرة واستجابة هذا البرنامج لاتمام عملية التصميم بالكامل والحصول على بيانات ورسومات كاملة صحيحة. ولقد أظهرت هذة النتائج الدقة الكبيرة فى أبعاد ومواصفات المسنن وكذلك الوفر الكبير فى الوقت والتكاليف.

ABSTRACT

The aim of this paper is to construct a software containing a complete design procedure and detailed drawing for bevel gears (straight, skew, zerol and spiral). This software can be executed on all the IBM personal computers XT/AT or compatibles. The main elements of the design in this program are: input data, type of bevel gear, type and shape of gear tooth system (ISO, US, BS, DIN and Gleason), module, virtual number of teeth, minimum number of teeth to avoid interference, tooth profile modification, shaft angle, contact ratio, face width, spiral angle, force analysis, sliding velocity and efficiency, material, equations for bending strength (Modified Lewis, Buckingham, AGMA and Gleason) and surface durability, also load carrying capacity for bending strength and surface durability using ISO, AGMA, Gleason and BS equations with constant or variable tooth load. Many equations and practical formulae are selected for making the gear construction (integar gear, solid gear, gear with web, gear with web and holes, gear with arms and composite gear).

With running the software, full specifications, geometry, kinematics, loads, stresses and detailed drawing of the gears are obtained according to the input data. Different examples are selected to show the variety of the output data and the strength of the software. Also these runs show the save in time and accuracy of the results

NOMENCLATURE

English letters

A	mean cone distance,	mm
CR	contact ratio	
C_F	surface condition factor	
C_H	hardness-ratio factor	
C_L, K_L	life factor	
C_R, K_R	reliability factor	
C_T, K_T	temperature factor	
d, $d_{1,2}$	pitch diameter,	mm
$E_{1,2}$	modulus of elasticity,	N/mm^2
F	face width,	mm
J	geometry factor	
K_p	pitch factor $= P^{0.8}$	
m	module, mm	
$m_{1,2}$	effective mass ,	slugs
m_G, u	gear ratio	
P_d	diametral pitch	
r, $r_{1,2}$	pitch circle radius,	mm
T_f	flash temperature,	^oF
v	tangential velocity,	mt/sec
W_a	acceleration load,	N
W_1	average force required to accelerate the masses	
X_b	speed factor for strength	
X_c	speed factor for wear	
$Z_{1,2}$	number of teeth	
Z'	zone factor	
A_o	outer cone distance,	mm
C_p	coefficient for elastic properties of the materials used	
C_m, K_m	load distribution factor	
C_o, K_o	overload factor	
C_s, K_s	size factor	
C_v, K_v	dynamic factor	
$d_{v_{1,2}}$	virtual pitch diameter,	mm
e	measured error in action	
I	durability geometry factor	
K_a	application factor	
K_x	cutter radius factor	
m_p	profile contact ratio	
m_F	face contact ratio	
p	circular pitch,	mm
P_n	normal diametral pitch	
r_o	edge radius,	mm
T_b	blank temperature,	^oF
v_s	sliding velocity,	mt/sec
W_d	dynamic load,	N
W_2	force required to deform the teeth through amount of effective error	
x, $x_{1,2}$	tooth correction factor	
Y	form factor	
$Z_{v1,2}$	virtual number of teeth	
$Z_{min1,2}$	minimum number of teeth	

Greek letters

α	pressure angle,	deg
$\theta_{1,2}$	pitch angle,	deg
λ	factor	
μ	coefficient of friction	
σ_w	working stress,	N/mm^2 or psi
σ_c	contact stress,	N/mm^2 or psi
β	spiral angle,	deg
Σ	shaft angle,	deg
σ_{ad}	allowable design stress,	N/mm^2
σ_b, σ_t	bending stress,	N/mm^2 or psi
$\rho_{1,2}$	tooth radius of curvature,	mm

Subscripts 1,2 = pinion and wheel respectively

INTRODUCTION

Bevel gears are the most efficient means of transmitting rotation between the intersecting shafts. Power requirements may be in the thousands of horsepower and in aircraft they have been successfully operated at very high pitchline speeds (about 25000 fpm). According to the power transmition, speed of rotation, speed ratio, available space, material to be used and the angle between the two shafts, there are different types of bevel gears are to be used as; straight, skew, zerol and spiral teeth bevel gears. Also Palloid, Oerlikon and kurvex tooth system of bevel gears are to be used.

The most Important stresses which should be considered for the bevel gear design are:

a- Stresses due to the bending of the tooth.

b- Stresses created by contact pressure, generally known as Hertz stresses. Besides, gear failure by wear, pitting and scoring are also considered.

Bevel gears are more difficult to design, drawing, manufacturing and assembly than that of spur and helical gears. They require special tools and machines to cut the teeth. Also cone apices must be brought into coincidence very carefully. Since the shafts intersect, one of the mating gears has to be mounted on the overhanging part of the shaft. Because of this the load is nonuniformly distributed over the face width, and the axial forces developed as the gears slide into mesh call for elaborate bearing assemblies.

Due to the above problems, the precise computation of the bevel gear capacity is an extremely difficult process, i.e. gear design is a complex and time-consuming task which must satisfy numerous design constraints. It is desirable to use CAD techniques to accomplish this task. Many investigators [1-6] have attempts for the gear tooth design using the computer. El-Bahloul [7] construct a software containing a complete design procedure and detailed drawing for spur, helical and double helical gears, this software can be executed on all the IBM personal computers XT/AT or compatibles To the author's knowledge there is no complete work that has been done on the bevel gear design using CAD technique.

The aim of this work is to construct a software containing complete specifications, geometry, kinematics, loads, efficiency, stresses and detailed drawing of the different types of bevel gears. The future steps (under preparation) are to construct softwares for worm and worm wheel, hypoid gears, and gears of circular-arc tooth-profile.

DESIGN APPROACH

1- Classifications of Bevel Gears :

Bevel gears can be classified according to types, shape of cone and pitch cone angle as follow :-

a- Types of bevel gears; There are four basic types of bevel gears, straight, skew, spiral and zerol bevels as shown in Fig.(1). Straight bevels are the oldest, the simplest, and still the most widely used. Teeth are straight and tapered and if extended in ward, would intersect the gear axis. Skew bevel gears "helical bevel gears" have teeth that are straight and oblique or in other words, that are tangent to an imaginary circle and make an angle with the cone element. The localized-tooth design tolerates small amounts of misalignment in the assembly of the gears and some displacement of the gears under load without concentrating the tooth contact at the ends of the teeth. As a result these gears are capable of transmitting heavier loads than the old-style straight bevel gears under the same conditions.

Spiral bevel gears have curved oblique teeth which contact each other gradually and smoothly from one end to the other. Well- designed spiral bevels have two or more teeth in contact at all times. Spiral bevel gears are used for high speeds (up to 11 ms^{-1} with the teeth unground and up to 35 ms^{-1} with their teeth ground or > 1000 rpm), also have the following advantages; greater contact ratio, meshing action is gradual and progressive over the whole length of the gear teeth, noise level is considerably small, the flank and the root strengths of teeth are greater, the minimum number of teeth to avoid undercutting is reduced, higher transmission ratio is achievable, spiral bevel gears have greater load carrying capacity.
Also other different types of spiral bevel gears are used as follow; palloid type with involute tooth trace developed by a German manufacturer, Klingelnberg. The height of tooth remains nearly constant along the tooth width.
Elloid gears have epicycloids or hypocycloids as spirals developed by Swiss manufacturer, Oerlikon, the height of tooth is constant
Kurvex toothed gears have teeth which are curved in the form of circular arc and the tooth height remains constant along the length of the teeth.

Zerol bevel gears, have curved teeth similar to those of the spiral bevels but with zero spiral angle at the middle of the face width and little end thrust. Zerol bevels are widely employed in the aircraft industry, where ground-tooth precision gears are generally required. Also used mostly in high-precision instruments where it is often necessary to have almost zero backlash.

b- Shape of Cone; Shapes of bevel gear cones are divided into three types, shown in Fig.(2). Type (1), the apices of the pitch and dedendum cones coincide, and the dedendum is proportional to the cone distance. This is the main tooth form in straight and skew bevel gears. It is also used in spiral bevel gears when $Z_\Sigma = 20$ to 100. Type (2), the apices of the pitch and dedendum cones do not coincide. The width of the bottom land is constant and the circular thickness of the teeth increases in proportion to the cone distance and used for spiral bevel gears. Type (3), the generators of the pitch, dedendum, and addendum cones are parallel. This kind of tooth finds application in spiral bevel gears when $Z_\Sigma \geq 40$.

c- Pitch cone angle; Fig.(3) shows the bevel gears arrangements, according to the angle between the driving and driven shafts and gear ratio.

2- Types of Gear Tooth systems :

The reference profiles of the tooth (basic racks) of ISO, US st., Gleason system, BS and DIN [8-19] are shown in table (1). 20° pressure angle is the most used in bevel gears. This alleviates the interference and under cutting problem and gives a stronger root section. Also 14.5°, 15°, 16°, 22.5°, and 25° pressure angle is used in some cases. Minimum pressure angle give a biger minimum number of teeth to avoid interference. In all cases, full depth teeth are used. Stub teeth are avoided because of the reduction in contact ratio, which may increase noise and the reduction in wear resistance. Table (2) shows the amount of gear addendum recommended for bevel gears [16].

3- Virtual Number of Teeth :

The virtual pitch diameters are the diameters of the pitch circles on the developed back cones as,

$$d_{v_{1,2}} = d_{1,2} \sec\theta_{1,2} \tag{1}$$

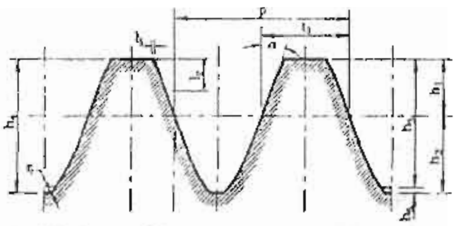
The circular pitch round the developed pitch circle is the same as that round the actual pitch circle, and the numbers of teeth round the developed back cones, when completed are the "virtual numbers of teeth" given by;

$$Z_{v_{1,2}} = Z_{1,2} \sec\theta_{1,2} \quad \text{For straight, and zerol bevel gears, and,} \tag{2}$$

$$Z_{v_{1,2}} = Z_{1,2} \sec\theta_{1,2} \sec^3\beta \quad \text{For spiral bevel gears.} \tag{3}$$

Des.	Standards	Gear type	$\alpha°$	h_1/m	h_2/m	h_3/m	h_4/m	h_5/m	r_f/m max	l_1/m max	l_2/m max	l_3
ISO	IS:5037-1976	Straight	20	1.00	1.20	2.00	2.20	0.20	0.3	0.02	0.60	P/2
US	AGMA 208.02	Straight	20	1.00	1.188 + a	2.00	2.188 + c	0.188 + a				
	AGMA 209.02	Spiral	20	0.70	1.00	1.70	1.888	0.188				
	AGMA 202.02	Zerol	20 22.5 25	1.00	1.188 + a	2.00	2.188 + c	0.188 + a				
Gleason Syst.		Straight	20	1.00	1.188 + b	2.00	2.188 + b	0.188 + b				
		Spiral	20 16	0.70	1.00	1.70	1.888	0.188				
		Zerol	20 22.5 25	1.00	1.188 + b	2.00	2.188 + b	0.188 + b				
BS	BSS 545/1949	Straight	20	1.00	1.25	2.00	2.25	0.25	0.257 max 0.191 min	0.019	0.628	
DIN	DIN 867	Straight	20	1.00	1.1:1.3	2.00	2.1:2.3	0.1:0.3				

Table (1) Reference profiles (basic racks) of IS, USA, British, DIN standards and Gleason syst. a,b & c constants



Gear type	Metric	English
Straight or Zerol	$0.540m_t + \frac{0.460m_t}{u^2}$	$\frac{0.540}{P_d} + \frac{0.460}{P_d m_G^2}$
Spiral	$0.540m_t + \frac{0.390m_t}{u^2}$	$\frac{0.540}{P_d} + \frac{0.390}{P_d m_G^2}$

Table (2) Gear addendum for bevel gears

4- Minimum Number of Teeth to Avoid Interference :

Minimum number of teeth required of the pinion to avoid interference "under cutting" is a function of the pressure angle, pitch cone angle, spiral angle, tooth profile modification and gear reduction ratio according to the following equations

$$Z_{min_1} = Z_{min} \cos\theta_1 = (2/\sin^2\alpha)\cos\theta_1 \quad \text{For straight, and zerol bevel gears, and,} \tag{4}$$

$$Z_{min_1} = Z_{min} \cos\theta_1 \cos^3\beta = (2/\sin^2\alpha)\cos\theta_1 \cos^3\beta \quad \text{For spiral bevel gears} \tag{5}$$

By trigonometrical transposition and allowing a marginal amount of undercutting as in the case of spur gears,

$$Z_{min_1} = 14\cos\theta_1 \quad \text{For straight and zerol bevel gears, and,} \tag{6}$$

$$Z_{min_1} = 14\cos\theta_1 \cos^3\beta \quad \text{For spiral bevel gears} \tag{7}$$

Table (3) gives the minimum numbers of teeth in the pinion and wheel for different types of bevel gears at different gear ratios and pressure angles according to [8-19].

5- Tooth Profile Modification :

Tooth profile modification is carried out to avoid undercutting, changing pitch cone angles (shaft angle), increasing the strength at the root and flank of the tooth, also betterment of sliding and contact relations. The amount of correction is,

$x_1=(14-Z_1\sec\theta_1)/17$ and $x_2=(14-Z_2\sec\theta_2)/17$ For straight, and zerol bevel gears, and (8)

$x_1=(14-Z_1\sec\theta_1\sec^3\beta)/17$ and $x_2=(14-Z_2\sec\theta_2\sec^3\beta)/17$ For spiral bevel gears (9)

$$Z_1\sec\theta_1 + Z_2\sec\theta_2 \geq 2Z_{min}$$

Tooth thickness at the tip circle $= 0.25m$

Another technique [9 and 10] for determining the tooth profile modification for straight, zerol and spiral bevel gears is given as:

$$x=0.5[1-Z_{v_1}/Z_{v_2}] \qquad (10)$$

x is not less than $0.025(30-Z_{v_1})$ or $x_{min}=(1-Z_v/20)$ and

$$x_1=-x_2 \qquad (11)$$

The condition $Z_{v_1}+Z_{v_2}< 60$ is unlikely to occur.

α^o	Type of bevel gear						α^o	Type of bevel gear					
	Straight tooth		Spiral tooth		Zerol tooth			Straight tooth		Spiral tooth		Zerol tooth	
	Pinion	Wheel	Pinion	Wheel	Pinion	Wheel		Pinion	Wheel	Pinion	Wheel	Pinion	Wheel
14.5	29	29	28	28			20	16	16	17	17	17	17
	28	29	27	29				15	17	16	18	16	20
	27	31	26	30				14	20	15	19	15	25
	26	35	25	32				13	30	14	20		
	25	40	24	33	Not used					13	22		
	24	57	23	36						12	26		
			22	40									
			21	42									
			20	50									
			19	70									
16			24	24			22.5	13	13	16	16	14	14
			23	25						16	19	13	15
			22	26						15	15		
			21	27						15	24		
	Not used		20	29	Not used					14	14		
			19	31						13	15		
			18	36			25	12	12	13	13	13	13
			17	45						13	14		
			16	59						12	12		

Table (3) Minimum number of teeth in pinion and gear

6- Contact Ratio :

Contact ratio for straight and zerol bevel gears is a profile contact ratio and equal to the length of action in normal section divided by the normal base pitch, this value must be greater than one.

$$CR=Z_n/p_{b_n} \qquad (12)$$

For spiral bevel gears, due to the spiral angle contact ratio is divided into two components, face contact ratio and profile (transverse) contact ratio as follow;

$$CR=\sqrt{m_F^2+m_p^2} \qquad (13)$$

$m_F=[(K_2\tan\beta-\frac{K_2^3}{3}\tan^3\beta)A_oP_d]/\pi$, $K_2=\frac{F}{A_o}[(2-F/A_o)/2(1-F/A_o)]$, and

$m_p=Z'/p'$, $Z' = \Delta\rho_1 + \Delta\rho_2$, $\Delta\rho = \sqrt{r_{a_n}^2-r_{b_n}^2}-r_n\sin\alpha$

$r_{a_n}=r_n+a$, $r_{b_n}=r_n\cos\alpha$, $r_n = r / \cos^2\beta = [d/(2\cos\theta\cos^2\beta)]\ \frac{A}{A_o}$

$a=a_o-0.5F\tan\alpha$, $A=A_o-0.5F$

$p'=p_n/[\cos\alpha(\cos^2\beta + \tan^2\alpha)]$, $p_n=(\pi A/A_oP_d)\cos\beta$

7- Face width :
The face width of bevel gears is limited by considerations of both tooth strength and tooling. The greater the face width, the smaller is the pitch of the small end of the tooth. Since deflection of the shafts, mountings, or bearings will occur and will allow the tooth contact to shift to the small ends of the teeth, the minimum size of the teeth at the small end must be sufficient to withstand the loads imposed. In addition, the space width at the roots of the teeth at the small end and the large end must be such that the cutter width required to clear the small end is not so narrow as to leave a "flange" at the large end of tooth space. Due to these difficulties the following empirical values are to be used:
For straight bevel gears $F \le A_o/3$ or $10/P_d$ using the smallest value.
For spiral bevel gears $F \le 0.3A_o$ or $\le 10/P_d$ using the smallest value.
For zerol bevel gears $F \le 0.25A_o$ or $\le 10/P_d$ maximum use whichever value is smaller.
Well-proportioned bevel gears have a face width from $6/P_d$ to $10/P_d$ but never exceeding $A_o/3$. On duplex zerol bevel gears 1" is the maximum face width in all cases.

8- Spiral angle :
The spiral angle β is usually specified at a pitch point P located at the middle of the face at a mean cone distance A. The spiral angle is different at different cone distances; at a general cone distance A' the spiral angle β' is given by the following formula:

$$\sin\beta' = \frac{A}{A'}\left(\sin\beta + \frac{A'^2 - A^2}{2Ar_c}\right) \quad , \quad 2r_c = (2A_o - F)/\sin\beta \qquad (14)$$

Though the spiral angle varies according to design considerations, its usual value is 35°. When other design factors permit, the spiral angle should be so selected that a face contact ratio of at least 1.25 is assured. Maximum smoothness of drive, however, is attained when the face contact ratio is between 1.5 and 2. If smaller spiral angles are used undercut may occur and the contact ratio may be less.

9- Force analysis :
The mean normal tooth force F_n acts on the pitch point P at the middle of the tooth width is resolved into three mutually perpendicular components as follows :
- *For straight or zerol bevel gears:*

$$\text{Tangential force} \quad F_{t_1} = F_{t_2} = F_t = F_n\cos\alpha = M_1/r_{m_1} \quad , \quad M_1 = 9550HP/n_1 \qquad (15)$$

$$\text{Radial force} \quad F_{r_1} = F_t\tan\alpha\cos\theta_1 \quad , \quad F_{r_2} = F_t\tan\alpha\cos\theta_2 \qquad (16)$$

$$\text{Axial force} \quad F_{a_1} = F_t\tan\alpha\sin\theta_1 \quad , \quad F_{a_2} = F_t\tan\alpha\sin\theta_2 \qquad (17)$$

- *For spiral bevel gears:*

$$\text{Tangential force} \quad F_{t_1} = F_{t_2} = F_t = M_1/r_{m_1} \quad , \quad M_1 = (9550HP/n_1)K \qquad (18)$$

The axial force F_a and the radial force F_r are given in table (4)

Pinion		Force on	Axial force	Radial force
Hand of spiral	Dir. of rotation			
Right	Clockwise	Pinion	$F_a = (F_t/\cos\beta)(\tan\alpha\sin\theta_1 - \sin\beta\cos\theta_1)$	$F_r = (F_t/\cos\beta)(\tan\alpha\cos\theta_1 + \sin\beta\sin\theta_1)$
Right	Counterclockwise		$F_a = (F_t/\cos\beta)(\tan\alpha\sin\theta_1 + \sin\beta\cos\theta_1)$	$F_r = (F_t/\cos\beta)(\tan\alpha\cos\theta_1 - \sin\beta\sin\theta_1)$
Left	Counterclockwise	Wheel	$F_a = (F_t/\cos\beta)(\tan\alpha\sin\theta_2 + \sin\beta\cos\theta_2)$	$F_r = (F_t/\cos\beta)(\tan\alpha\cos\theta_2 - \sin\beta\sin\theta_2)$
Left	clockwise		$F_a = (F_t/\cos\beta)(\tan\alpha\sin\theta_2 - \sin\beta\cos\theta_2)$	$F_r = (F_t/\cos\beta)(\tan\alpha\cos\theta_2 + \sin\beta\sin\theta_2)$

Table (4) Axial and radial forces acting on spiral bevel gears.

10- Sliding velocity and efficiency :
The sliding velocity of a bevel gears at a distance S from the pitch point is approximately:

$$v_s = S(\omega_1^2 + \omega_2^2 + 2\omega_1\omega_2\cos\Sigma)(\sin^2\alpha + \cos^2\beta\cos^2\alpha)^{0.5} \quad \text{For spiral bevel gear} \qquad (19)$$

In the case of right-angle bevel gear drives, $\Sigma = 90°$ and

$$v_s = S(\omega_1\cos\theta_1 + \omega_2\cos\theta_2)(\sin^2\alpha + \cos^2\beta\cos^2\alpha)^{0.5} \qquad (20)$$

For straight and zerol bevel gears $\beta = 0$, in equations (19 and 20).
Efficiency of the bevel gear drives is very important in the applications where large amounts of power are being transmitted. Efficiency is calculated from the following equations [8]

$$\eta = 100 - 50\mu[(\cos\theta_1 + \cos\theta_2)/\cos\alpha]\left(\frac{H_s^2 + H_t^2}{H_s + H_t}\right) \quad \text{For straight and zerol bevel gears, and} \qquad (21)$$

$$\eta = 100 - 50\mu(\cos\theta_1 + \cos\theta_2)\left(\frac{H_s^2 + H_t^2}{H_s + H_t}\right)(\cos^2\beta/\cos\alpha) \quad \text{For spiral bevel gears} \quad (22)$$

$$H_s = (m_G + 1)\left[\sqrt{\left(\frac{r_{o_2}}{r_2}\right)^2 - \cos^2\alpha} - \sin\alpha\right], \quad \text{and} \quad H_t = \left(\frac{m_G + 1}{m_G}\right)\left[\sqrt{\left(\frac{r_{o_1}}{r_1}\right)^2 - \cos^2\alpha} - \sin\alpha\right]$$

Another equations are used for calculating the efficiency [12]

$$\eta = 1 - F\mu[(\cos\theta_1/Z_1) + (\cos\theta_2/Z_2)] \quad \text{For straight and zerol bevel gears, and} \quad (23)$$

$$\eta = 1 - (0.8F\mu\cos\beta)[(\cos\theta_1\cos^3\beta/Z_1) + (\cos\theta_2\cos^3\beta/Z_2)] \quad \text{For spiral bevel gears} \quad (24)$$

11- Design of Bevel Gear Tooth According to Bending Strength Failure :

Bevel gear tooth design according to bending strength is divided into two items, first, static failure due to bending stress and the second, fatigue failure due to bending stress. To cover these items, many techniques are used as follows, modified Lewis equation, Gleason technique, AGMA equation and Buckingham equation.

- Modified Lewis Equation:

The load acting on a bevel gear tooth varies linearly along the face of the tooth. Tooth thickness also varies linearly along the face of the gear. Thus,

$$F_t = \sigma_b FYm(1 - F/A_o) \quad \text{or} \quad \sigma_b = F_t/FYm(1 - F/A_o) \quad (25)$$

- The basic equation for the bending stress in a bevel gear is given as follow according to Gleason [14]

$$\sigma_t = F_t K_o P_d K_s K_m/(K_v FJK_x), \quad \sigma_w = \sigma_{at} K_L/K_T K_R \quad (26)$$

- AGMA equation for bending stress:

$$\sigma_t = F_t K_o P_d K_s K_m/(K_v FJ), \quad \sigma_{ad} = \sigma_{at} K_L/K_T K_R \quad (27)$$

- Buckingham Equation, $W_d = F_t + \sqrt{W_a(2.W_2 - W_a)}$ (28)

$$W_a = (W_1.W_2)/(W_1 + W_2)$$

$$W_1 = \frac{c_1 m_1 m_2}{(m_1 + m_2)}[(1/r_{v_1}) + (1/r_{v_2})]V^2, \quad W_2 = \frac{F.e}{c_2(1/E_1 + 1/E_2)} + F_t \quad \text{For straight and zerol bevel gears}$$

$$W_1 = \frac{c_1 m_1 m_2}{(m_1 + m_2)}[(1/r_{v_1}) + (1/r_{v_2})]V^2\cos^2\beta, \quad W_2 = \frac{F.e}{c_2(1/E_1 + 1/E_2)}\cos^2\beta + F_t \quad \text{For spiral bevel gear}$$

$C_1 = 0.00086$ For 14.5° gears , 0.0012 For 20° gears

$C2 = 9.345$ For 14.5° gears , 9.000 For 20° Full-depth gears , 8.7 For 20° stub gears

12- Design of Bevel Gear Tooth According to Surface Failure :

Design of bevel gear tooth according to surface failure is done using the following equations:-

- Contact stress, The Hertzian contact stress for bevel gears is given by the equation

$$\sigma_H = C_p\sqrt{F_t/(C_v F d_1 I)} \quad (29)$$

- Limiting load for wear :

$$F_w = 0.75 d_{v_1} FKQ \quad \text{For straight and zerol bevel gears} \quad (30)$$

$$F_w = 0.75 d_{v_1} FKQ/\cos^2\beta \quad \text{For spiral bevel gear} \quad (31)$$

$$Q = 2(Z_2/\cos\theta_2)/[(Z_1/\cos\theta_1) + (Z_2/\cos\theta_2)]$$

$$d_{v_1} = (d_1 - F\sin\theta_1)/\cos\theta_1, \quad K = \sigma^2\sin\alpha[(1/E_1) + (1/E_2)]/1.4$$

- AGMA wear equation :

$$\sigma_c = C_p[F_t C_o C_s C_m C_F/(C_v d_1 F I)]^{0.5}, \quad C_p = [1.5/(\pi(1 - \upsilon_1^2)/E_1 + (1 - \upsilon_2^2)/E_2)]^{0.5} \quad (32)$$

$$\sigma_c \leq \sigma_{ac}[C_L C_H/C_T C_R] \quad (33)$$

- Gleason (Pitting Formula) :

$$\sigma_c = C_p\left(\frac{2M_1 C_o}{C_v} \cdot \frac{1}{Fd_1^2} \cdot \frac{C_s C_m C_f}{I}\right)^{0.5}\left(\frac{M_{1c}}{M_1}\right)^{v_x}, \quad I = SR_t\cos\alpha\cos\theta_1/(Fd_1 C_I m_n) \quad (34)$$

- Flash Temperature :

$$T_k = T_B + \Delta T_o = T_B + (\pi/4)^{1/2}\sigma_c\,\mu\upsilon_s/[C_1(v_1/l_1)^{1/2} + C_2(v_2/l_2)^{1/2}] \quad (35)$$

$$\mu = K_1\log K_2(v_s V_T^j \eta_o^m F_t^n)$$

- *Scoring Index :*

$$SI = (F_t/F)^{0.75}.(n_1)^{0.5}.(m)^{0.25} \tag{36}$$

13- Design of Bevel Gear According to Load Carrying Capacity :

After the bevel gear-tooth data have been calculated, it is necessary to calculate the capacity of the gearset. Since the design started from an estimate, it may be that the first design which is worked out in detail is too small or too large. Once all the gear-tooth data have been calculated, it is possible to use design formulae to determine a rated capacity of the gearset. This rated capacity should be larger than the actual load which will be applied to the gearset. Complete form of AGMA, ISO [16], BS [9] and Gleason [14] bevel gear rating formulae for bending strength, surface durability and power are used and given as follows :

- *Strength formula :*

$$\sigma_t = K_t.U_t.K_d \tag{37}$$

$$U_t = (F_t/F\cos\beta).\frac{Z_1}{d_1}.(\frac{A_o}{A_o-0.5F})^2 \quad , \quad K_t = \cos\beta \;/\; J \quad , \quad K_d = (K_a.K_m.K_s)/K_v$$

- *Durability formula:*

$$\sigma_c = C_k(K.C_d)^{0.5} \tag{38}$$

$$C_k = C_p[m_G/I(1+m_G)]^{0.5}, \quad K = F_t(m_G+1)/(Fdm_G) \quad , \quad C_d = C_a.C_m.C_s/C_v$$

The horsepower rating according to AGMA for bending strength at peak load of straight, zerol and spiral bevel gears is given, by

$$HP = \sigma_{at}[(78nd_1FY(A_o-0.5F)/126000P_dA_o(78+\sqrt{V})] \tag{39}$$

The rated power of a bevel gears according to Gleason is given by:

$$HP = \sigma_{at}[(n_1d_1FJK_x)/(126.050P_dK_s)] \quad , \quad HP' = HP[K_LK_v/(K_oK_TK_RK_m)] \tag{40}$$

The maximum allowable transmitted horsepower based on wear according to AGMA is given as follow [18]:

$$P_{ac} = \frac{n_1FIc_v}{126000C_sC_mC_fC_o}.(\sigma_{ao}.\frac{d_1}{C_p}.\frac{C_LC_H}{C_TC_R})^2 \tag{41}$$

Horsepower rating according to AGMA for surface durability is given by the equations

$$HP = 0.8C_mC_8F \qquad \text{For straight and zerol bevel gears} \tag{42}$$

$$HP = C_mC_8F \quad , \quad C_m = \frac{d_1^{1.5}n_1}{233}(1.4-\frac{V}{4400}) \qquad \text{For spiral bavel gears} \tag{43}$$

- *British standard rating formulae:*

The permissible tangential load, calculated as acting at the pitch circle, at normal rating is given by:

-*Straight and zerol bevel gears :*

$$F_t' = \sigma_{bo}\frac{X_b.Y.F}{1.1P}(\frac{A_o-F}{A_o}) \qquad \text{For strength} \tag{44}$$

$$F_t' = \sigma_{co}\frac{X_c.Z.F}{1.47.K_p}(\frac{A_o-F}{A_o}) \qquad \text{For wear} \tag{45}$$

-*Spiral bevel gears :*

$$F_t' = \sigma_{bo}\frac{X_b.Y.F}{1.1P(1-\frac{m_F}{4})}(\frac{A_o-F}{A_o})(1/\sec^2\beta) \qquad \text{For strength} \tag{46}$$

$$F_t' = \sigma_{co}\frac{X_c.Z.F}{1.1K_p(\frac{4-m_F}{3})}(\frac{A_o-F}{A_o}) \qquad \text{For wear} \tag{47}$$

The corresponding horse-power in each case is obtained by multiplying the above equations by $Z.n/126000P$.

- *Variable Loading:*

If duration time other than 12 hours per day, torque and speed are constant, but the gear runs for U hours per day, the normal rating of the gears is adjusted to the value

$$\text{Normal rating} = \text{actual load} \times K_u \tag{48}$$

According to B.S. 545 - 1949 , $K_u = \sqrt[3]{U/12}$ (49)

If the torque and/or speed are variable according to a known daily load-cycle which can be expressed as

U_1 hours at the maximum torque M_1 and speed n_1. U_2 hours at a torque M_2 and speed n_2. U_3 hours at a torque M_3 and speed n_3, etc., the equivalent running time U_e is given by .

$$U_e = U_1 + U_2(n_2/n_1)(M_2/M_1)^3 + U_3(n_3/n_1)(M_3/M_1)^3 + \dots\dots\dots \tag{50}$$

This equation is applied in wear. For strength, the index 3 is replaced by 7. In order to determine the equivalent normal rating, the value of Ue is substituted in the appropriate expression (49) above and equivalent running time factor is then applied in expression (48). Also another equation is used according to Gleason [14]

$$U_{e_1} = 60L_H[K_1 n_{1_1} + K_2 n_{1_2}(M_2/M_1)^{5.68} + K_3 n_{1_3}(M_3/M_1)^{5.68} + \dots\dots\dots + K_n n_{1_n}(M_n/M_1)^{5.68}] \tag{51}$$

$$U_{e_2} = U_{e_1}.(n_1/n_2) \tag{52}$$

14- Gear Construction

Gear construction depends upon the size, material, stress analysis, method and accuracy of manufacturing, type of application, operational parameters, type and technique of heat treatment and cost. Different types of bevel gear constructions are shown in Fig(5) [20-25]. More informations about the design calculations of the gear blanks are given in [7].

FLOW CHART AND COMPUTER PROGRAM

Construction of the software containing design and drawing of straight, skew, zerol and spiral bevel gears covers all requirements of the designers and users of the gears. These requirements are divided and specified into the following:

1- Experience and expertise of the user is not efficient, and the available data are not enough. For example, available data are transmitted power, input and output speeds. Complete calculations, selections and checks are done for running the program automatically without interface with the software to obtain the safe design and drawing of the gears according to the flow chart shown in Fig(6).
2- Experience and expertise of the user is efficient, some items are selected or assumed such as module and/or material, type of gear and spiral angle. This facility gives minimum running time of the program and minimum cost for the design.
3- Improving the design and performance of the bevel gear in service or old design by feeding the software with some information. The program calculates the new required dimensions, specifications and new drawings.
4- Obtaining specifications dimensions and drawings to manufacture a new bevel gear instead of an old one broken in service.

The program is written in Turbo Basic Language [26]. The compiler of the program with Turbo Basic gives an executable file to run on the Dos prompt. The program has created and constructed automatically two files which include all information on designing and drawing the gear. Types of these files are the DXF file and SCR file. The DXF and SCR files format are familiar with AutoCAD program [27,28]. By the (DXFin) command, the drawing can be generated on the screen.

This software can be executed on all the IBM personal computers XT/AT or compatibles with 640 KB RAM, Math. Co-processor and 10 MB hard disk

Fig(7) shows the different menus of the software. For example input power = 25 Kw, Input speed 750 rpm, gear ratio 2 and shaft angle 90°.

COMPUTATIONAL RESULTS AND DISCUSSION:

Complete output of any run is divided into three items:

1. Specifications and geometry,
2. Kinematics, loads and stresses,
3. Pinion and wheel constructions with the required partial views.

The specifications, geometry, kinematics, loads, stresses, pinion and wheel constructions of 25Kw and 750 rpm, 2 gear ratio and 90° shaft angle input data are shown in table (5,6) and Fig.(8), those of 30 Kw, 800 rpm, 3 gear ratio and 90° shaft angle input data are shown in table (7,8) and Fig.(9), and those of 100 Kw, 3000 rpm, 4 gear ratio and 90° shaft angle input data are shown in table (9,10) and Fig.(10). Table (11,12) and Fig.(11) show the output results of 100 Kw, 3000 rpm, 3 gear ratio and 120° shaft angle input data. From these tables and figures, it is clear that the variety of the output results according to the input data. Straight bevel gears are given for the smallest power. For increasing power and speed skew bevel gears are shown. For more increasing power and speed spiral bevel gears are shown with the same module and materials for all cases.

CONCLUSION

It is possible to construct a software containing a complete design and make a detailed drawing of straight, skew, zerol and spiral bevel gears. This software can be executed on all the IBM personal computers XT/AT or compatibles. A complete design and a detailed drawing of the bevel gear can be obtained according to the input power and speed, generally speaking, or entering some information to get a special design of the bevel gear such as dimensions, material, manufacturing process and heat treatment. They can also be used for redesign the gear to improve its performance or life.

By using this software, the results show that:

1- Obtaining a remarkably high degree of accuracy in the calculated dimensions and drawing.
2- Safe design is obtained for all point of view due to using multi equations and different techniques for checking all design procedures.
3- Overcoming the problems of experience and expertise for the designers.
4- Saving time.
5- Minimum cost of the design.

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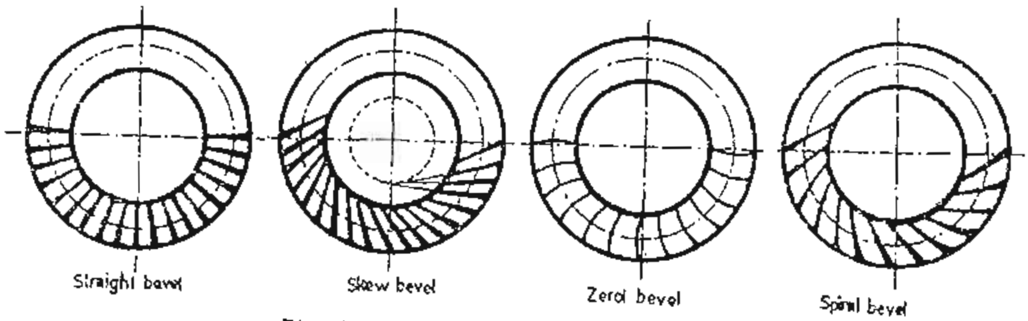
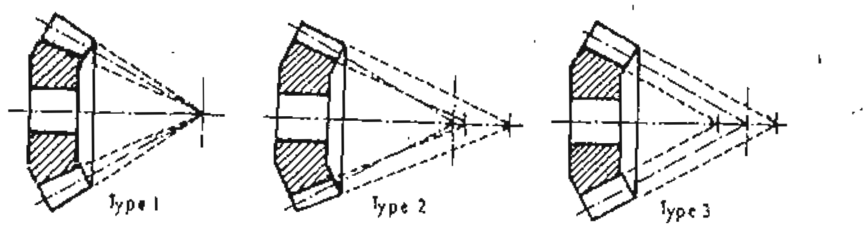
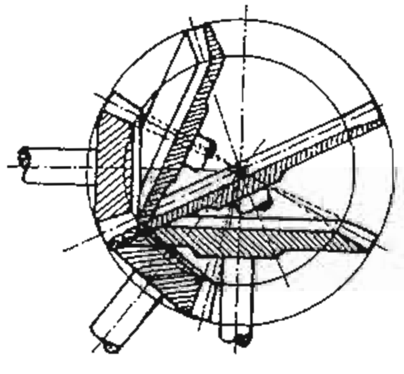


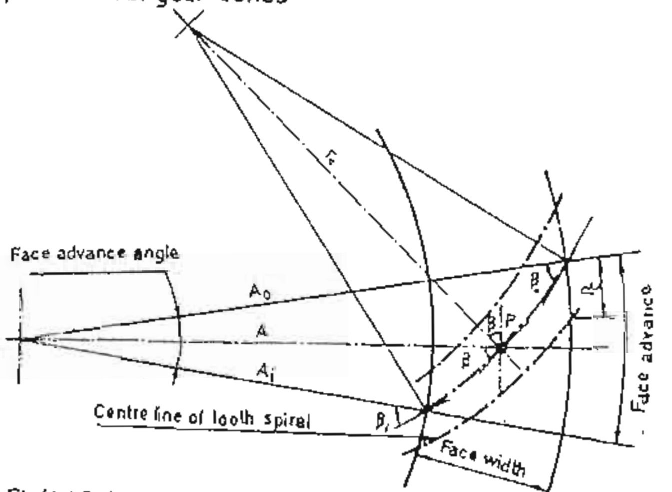
Fig(1) Types of bevel gears



Fig(2) Shapes of bevel gear cones



Fig(3) Bevel gears arrangements



Fig(4) Spiral angle of spiral bevel gear

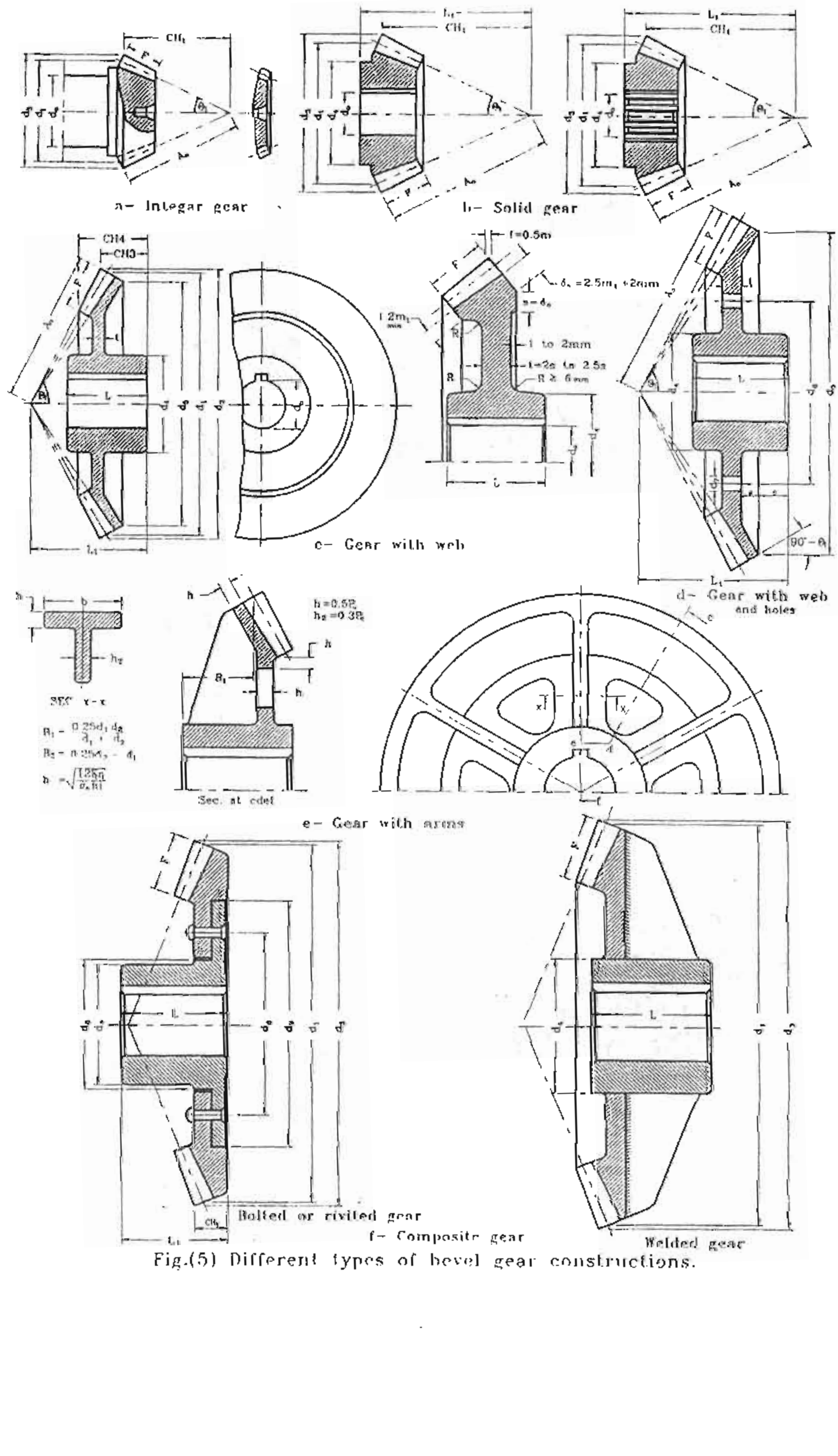


Fig.(5) Different types of bevel gear constructions.

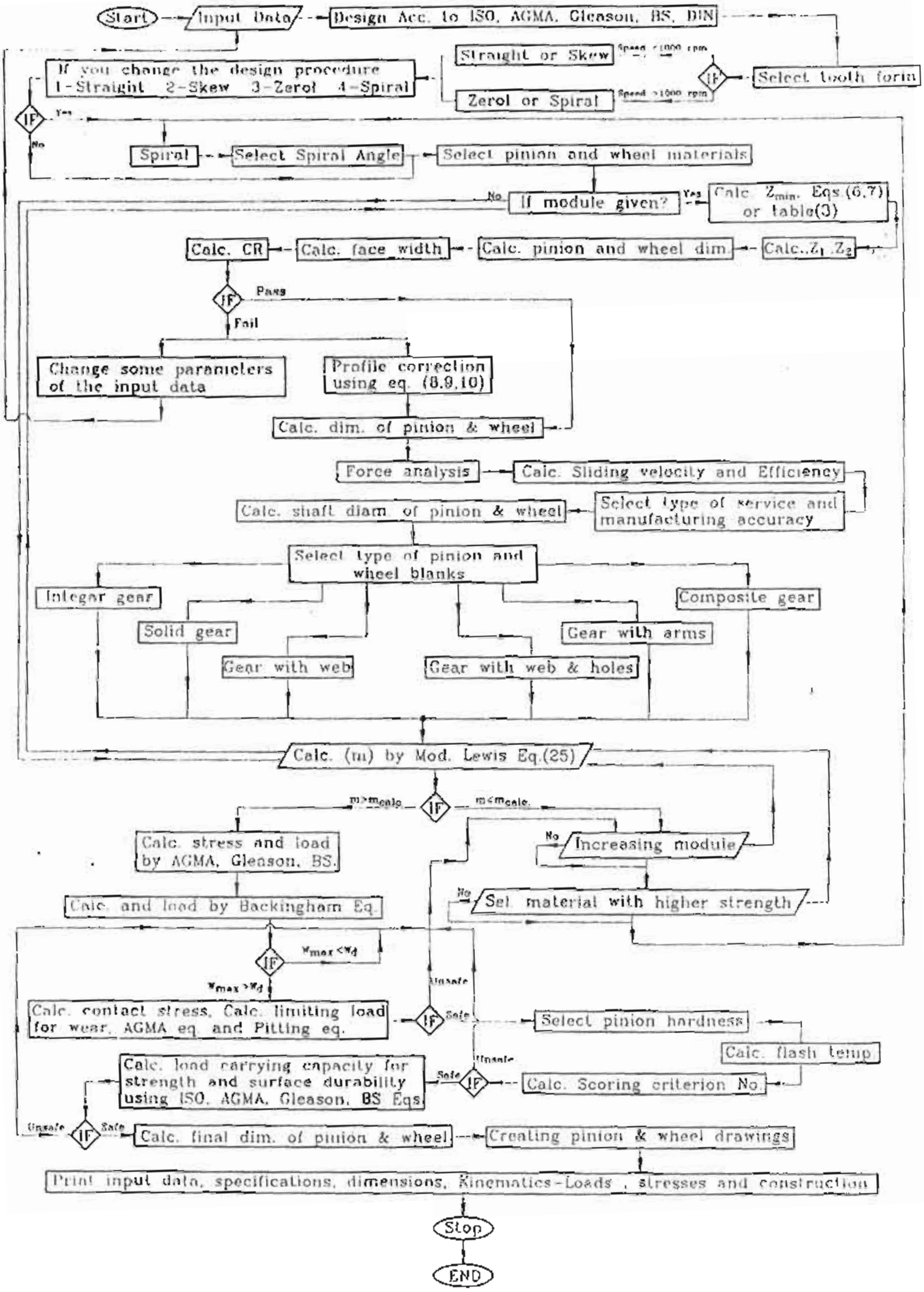
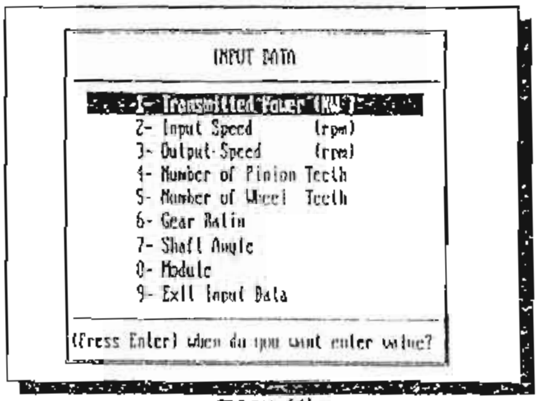


Fig.(6) Flow chart.



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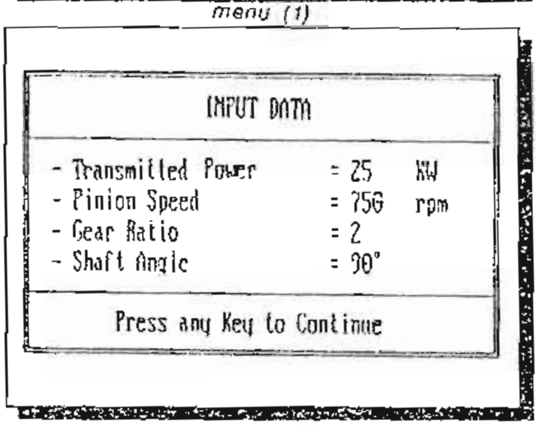

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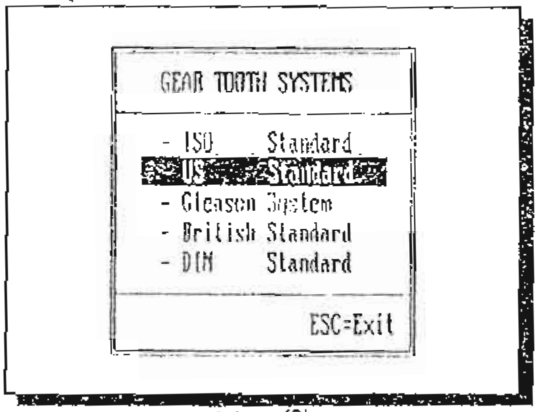

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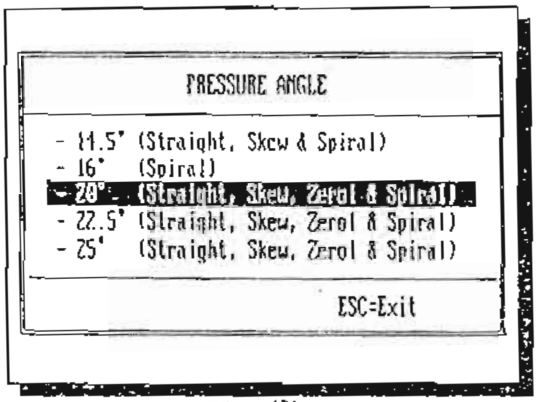

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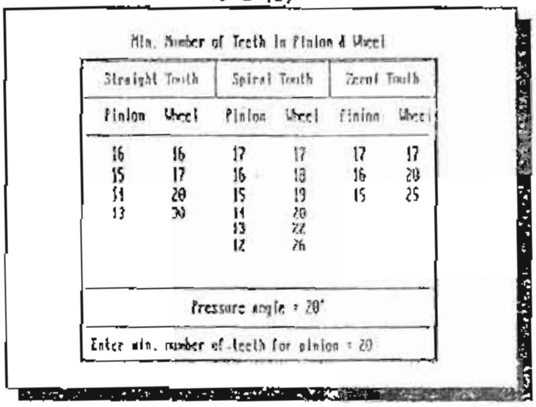

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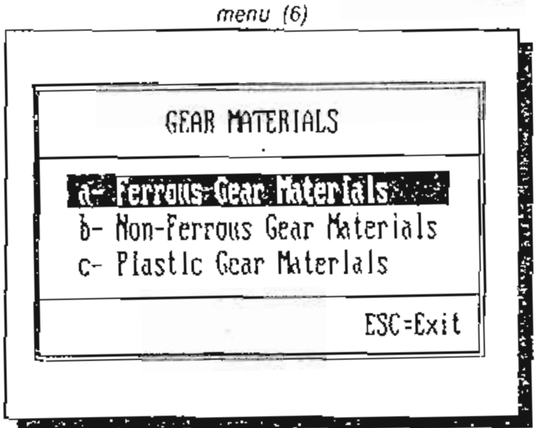

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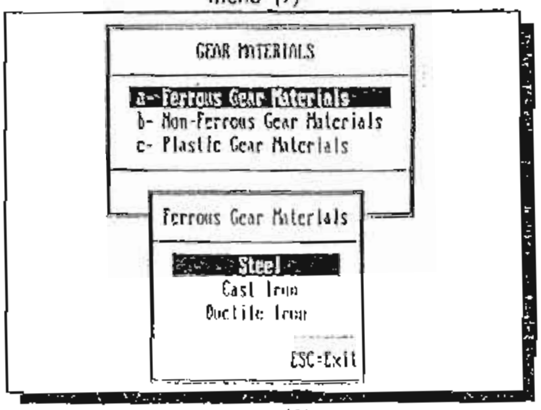

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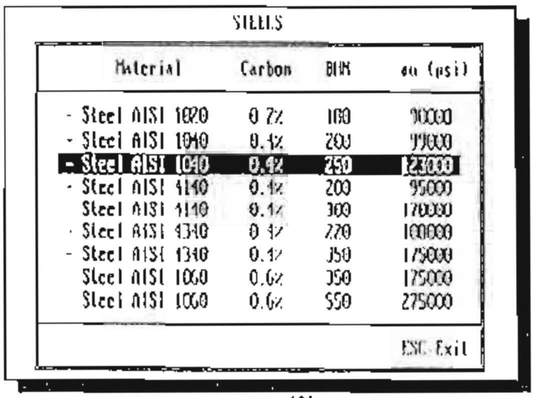

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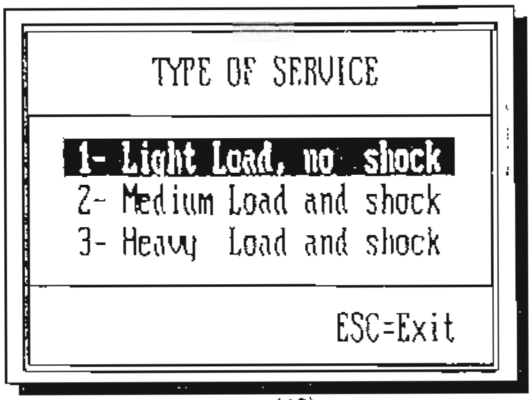

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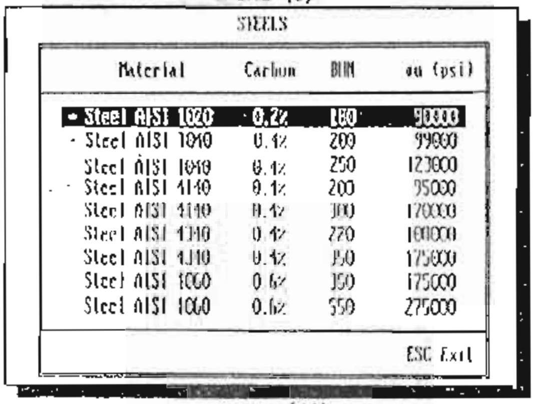

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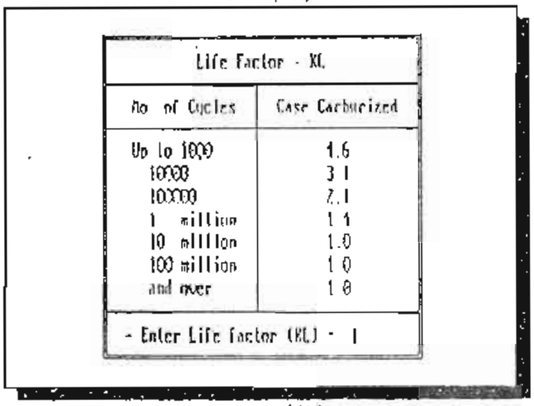

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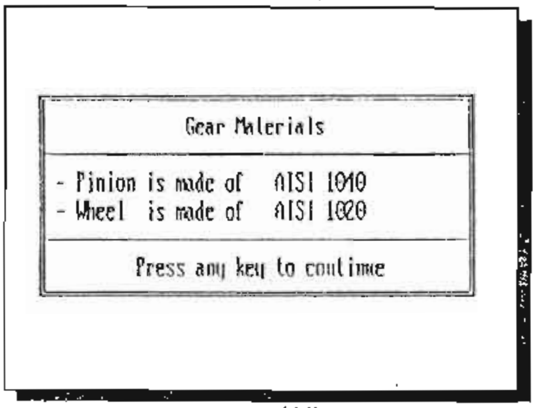

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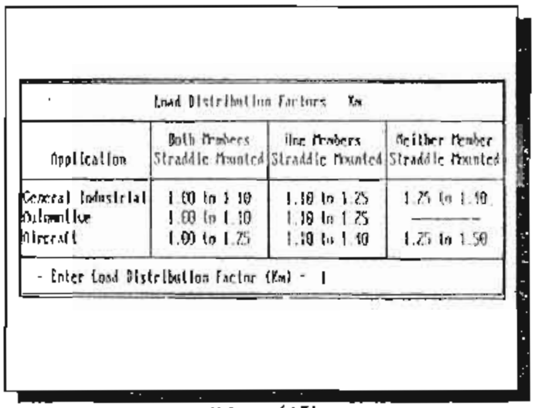

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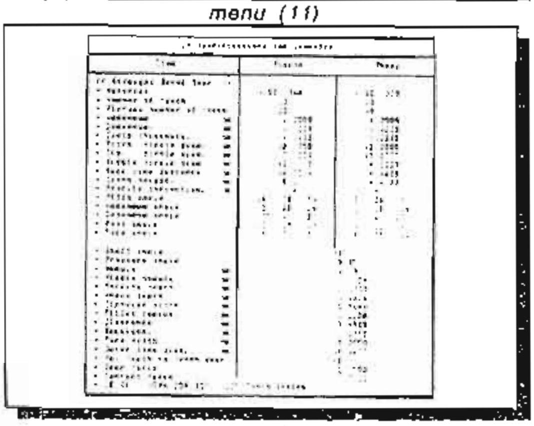

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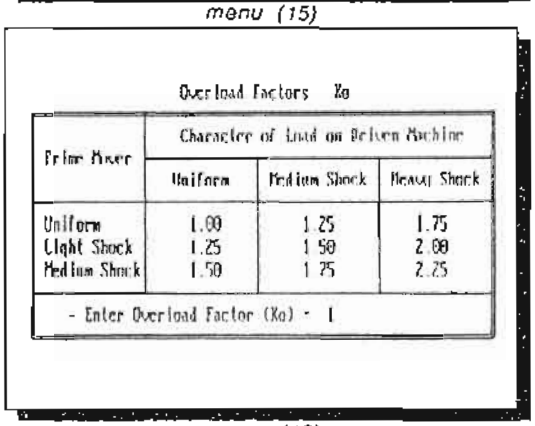

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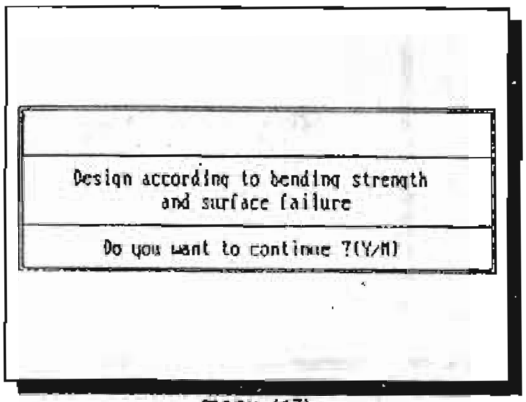

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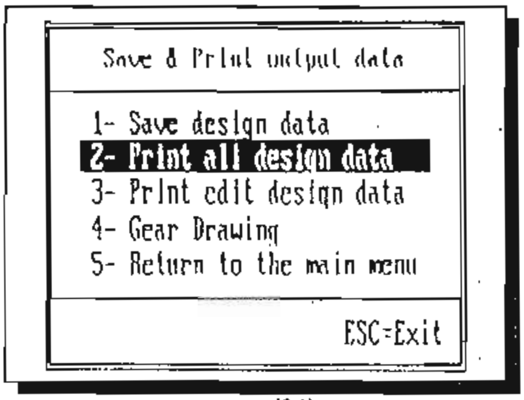

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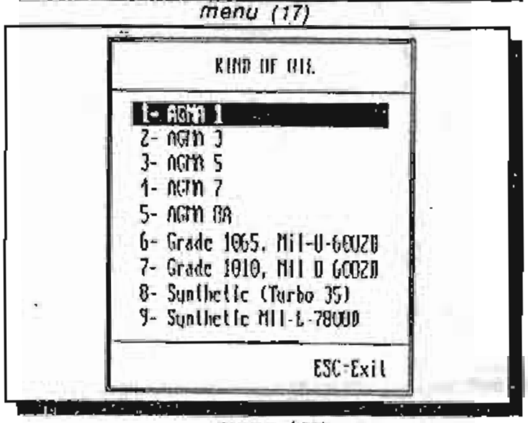

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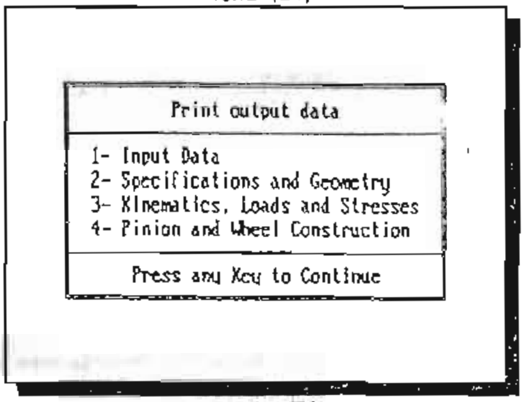

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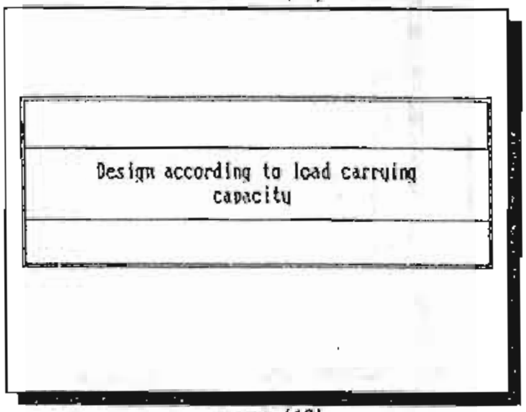
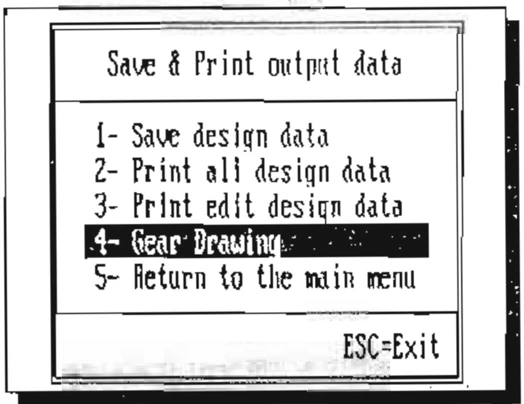
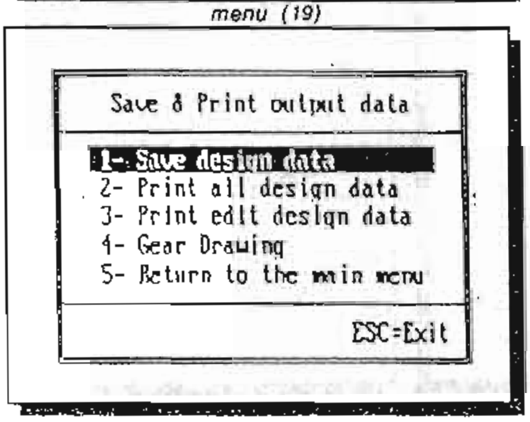


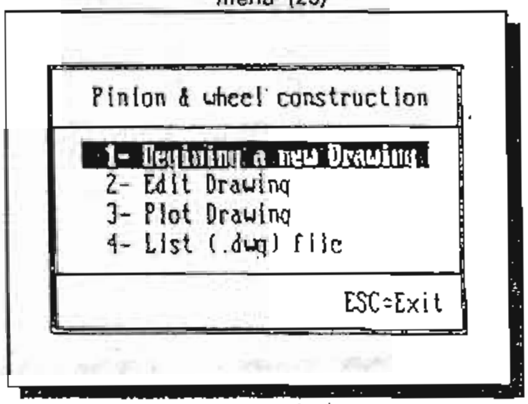
menu (19)



menu (23)



menu (20)



menu (24)

Fig.(7) Different menus of software for run of 25 Kw power, 750 rpm speed, 2 gear ratio and 90° shaft angle

Input Data		
- Power.	Kw	= 25
- Input speed.	rpm	= 750
- Gear ratio		= 2.0000
- Shaft angle		= 90°
- Design according to "US"		

Output Data
1- Specifications and Geometry
2- Kinematics, Loads and Stresses
3- Gear Construction

1- Specifications and Geometry			
Item		Pinion	Wheel
<< Straight Bevel Gear >>			
- Material		AISI 1040	AISI 1020
- Number of teeth		20	40
- Virtual number of teeth		22	89
- Addendum.	mm	4.0000	4.0000
- Dedendum.	mm	4.8028	4.8028
- Tooth thickness.	mm	6.2832	6.2832
- Pitch circle diam..	mm	80.0000	160.0000
- Tip circle diam..	mm	87.1554	163.5777
- Middle circle diam..	mm	62.1115	124.2229
- Back cone distance.	mm	44.7214	178.8855
- Crown height.	mm	78.2111	36.4223
- Profile correction.	mm	-	-
- Pitch angle		26° 33' 54"	63° 26' 5"
- Addendum angle		2° 33' 38"	2° 33' 38"
- Dedendum angle		3° 4' 25"	3° 4' 25"
- Root angle		23° 29' 29"	60° 21' 40"
- Face angle		29° 7' 32"	65° 59' 44"
- Shaft angle		90°	
- Pressure angle		20.0°	
- Module	mm	4.00	
- Middle module	mm	3.1056	
- Working depth.	mm	8.0000	
- Whole depth.	mm	8.8028	
- Circular pitch.	mm	12.5664	
- Fillet radius.	mm	1.2000	
- Clearance.	mm	0.8028	
- Backlash.	mm	0.1975	
- Face width.	mm	40.0000	
- Outer cone dist..	mm	89.4427	
- No. teeth in crown gear		45	
- Gear ratio		2.0000	
- Contact ratio		1.7122	
- US St. (AGMA 208.02), 20° Tooth System			

Table (5) Specifications and geometry of straight bevel gears for 25 Kw power, 750 rpm input speed, 2 gear ratio and 90° shaft angle.

2- Kinematics-Loads and Stresses		
- Speed for pinion & wheel.	rpm	= 750 , 375
- Pitch line velocity.	m/sec	= 3.142
- Sliding velocity.	m/sec	= 0.942
- Efficiency		= 84.35%
Force Analysis :-		
- Normal load .	N	= 564
- Tan., Rad. & Ax. loads for pinion.	N	= 812, 264, 132
- Tan., Rad. & Ax. loads for wheel.	N	= 812, 132, 264
Design Acc. to Bending Strength Failure :-		
- Calc. module Acc. to M.L.E..	mm	= 3.34< 4.00 Safe
A- Loads		
- Tan. load Acc. to M.L.E..	N	= 8482
- Dynamic load Acc. to Backingham.	N	= 6404
- Tan. load Acc. to AGMA.	N	= 10850
- Tan. load Acc. to Gleason.	N	= 11715
- Tan. load for pinion Acc. to B S.	Ib(N)	= 2294(10207)
- Tan. load for wheel. Acc. to B.S.	Ib(N)	= 1928(8579)
B- Stresses		
- Max. allowable stress.	N/mm²	= 287
- Bending stress Acc. to M.L.E..	N/mm²	= 147
- Bending stress Acc. to AGMA.	N/mm²	= 208
- Bending stress Acc. to Gleason.	N/mm²	= 192
Design Acc. to Surface Failure :-		
A- Loads		
- Wear load Acc. to Buckingham.	N	= 10871
- Wear load Acc. to AGMA.	N	= 10042
- Wear load Acc. to Gleason.	N	= 3165
- Wear load for pinion Acc. to B.S.	Ib(N)	= 2370(10544)
- Wear load for wheel Acc. to B S.	Ib(N)	= 2042(9084)
B- Stresses		
- Max. allowable contact stress.	N/mm²	= 841
- Hertzian contact stress.	N/mm²	= 630
- Calc. contact stress Acc. to AGMA.	N/mm²	= 504
- Gleason (Pitting).	N/mm²	= 521
- Blank temperature		= 150 °F
- Flash temperature		= 181.397 °F
- Scoring Index		= 3377
- Critical Scoring criterion number		= 6000
Design Acc. to Load Carrying Capacity :-		
<< ISO >>		
- Tooth bending stress.	N/mm²	= 207
- Tooth-surface durability.	N/mm²	= 338
<< AGMA >>		
- Tooth bending stress.	Ib/in²	= 29950
- Tooth-surface durability.	Ib/in²	= 48988
<< Power rating >>		
- Transmitted power	Kw	= 25
- Max. power Acc. to AGMA	Kw	= 43
- Max. power Acc. to Gleason	Kw	= 34
- Max. power Acc. to AGMA (based on wear)	Kw	= 31
- Max. power Acc. to AGMA for durability	Kw	= 27

Table (6) Kinematics, loads, stresses and power rating of straight bevel gears for 25 Kw power, 750 rpm input speed, 2 gear ratio and 90° shaft ang.

Input Data		
- Power.	Kw	= 30
- Input speed.	rpm	= 800
- Gear ratio		= 3.0000
- Shaft angle		= 90°
- Design according to "US"		

Output Data
1- Specifications and Geometry
2- Kinematics, Loads and Stresses
3- Gear Construction

1- Specifications and Geometry

Item		Pinion	Wheel
<< Skew Bevel Gear >>			
- Material		AISI 1040	AISI 1020
- Number of teeth		20	60
- Virtual number of teeth		21	190
- Addendum.	mm	4.0000	4.0000
- Dedendum.	mm	4.8028	4.6028
- Tooth thickness.	mm	6.2832	6.2832
- Pitch circle diam..	mm	80.0000	240.0000
- Tip circle diam..	mm	87.5895	242.5298
- Middle circle diam..	mm	67.3509	202.0527
- Back cone distance.	mm	42.1637	379.4733
- Crown height.	mm	118.7351	36.2053
- Profile correction.	mm	-	-
- Pitch angle		18° 26' 5"	71° 33' 54"
- Addendum angle		1° 48' 40"	1° 48' 40"
- Dedendum angle		2° 10' 28"	2° 10' 28"
- Root angle		16° 15' 37"	69° 23' 26"
- Face angle		20° 14' 46"	73° 22' 34"
- Helix angle		20°	
- Shaft angle		90°	
- Pressure angle		20.0°	
- Module	mm	4.00	
- Middle module	mm	3.3675	
- Working depth.	mm	8.0000	
- Whole depth.	mm	8.8028	
- Circular pitch.	mm	12.5664	
- Fillet radius.	mm	1.2000	
- Clearance.	mm	0.8028	
- Backlash.	mm	0.1975	
- Face width.	mm	40.0000	
- Outer cone dist..	mm	126.4911	
- No. teeth in crown gear		63	
- Gear ratio		3.0000	
- Contact ratio		1.7393	
- US St. (AGMA 208.02). 20° Tooth System			

Table (7) Specifications and geometry of skew bevel gears for 30 Kw power, 800 rpm input speed, 3 gear ratio and 90° shaft angle.

2- Kinematics-Loads and Stresses

- Speed for pinion & wheel.	rpm =	800 , 267
- Pitch line velocity.	m/sec =	3.351
- Sliding velocity.	m/sec =	0.962
- Efficiency	=	85.24%
Force Analysis :-		
- Normal load	N =	972
- Tan., Rad. & Ax. loads for pinion.	N =	913, 315, 105
- Tan., Rad. & Ax. loads for wheel.	N =	913, 105, 315
Design Acc. to Bending Strength Failure :-		
- Calc. module Acc. to M.L.E..	mm =	3.59< 4.00 Safe
A- Loads		
- Tan. load Acc. to M.L.E..	N =	10244
- Dynamic load Acc. to Backingham.	N =	6682
- Tan. load Acc. to AGMA.	N =	10594
- Tan. load Acc. to Gleason.	N =	11441
- Tan. load for pinion Acc. to B.S.	Ib(N) =	2801(12463)
- Tan. load for wheel. Acc. to B.S.	Ib(N) =	2554(11360)
B- Stresses		
- Max. allowable stress.	N/mm² =	287
- Bending stress Acc. to M.L.E..	N/mm² =	169
- Bending stress Acc. to AGMA.	N/mm² =	239
- Bending stress Acc. to Gleason.	N/mm² =	221
Design Acc. to Surface Failure :-		
A- Loads		
- Wear load Acc. to Buckingham.	N =	17295
- Wear load Acc. to AGMA.	N =	9804
- Wear load Acc. to Gleason.	N =	3640
- Wear load for pinion Acc. to B.S.	Ib(N) =	2894(12876)
- Wear load for wheel Acc. to B.S.	Ib(N) =	2704(12030)
B- Stresses		
- Max. allowable contact stress.	N/mm² =	841
- Hertzian contact stress.	N/mm² =	676
- Calc. contact stress Acc. to AGMA.	N/mm² =	804
- Gleason (Pitting).	N/mm² =	830
- Blank temperature	=	150 °F
- Flash temperature	=	194.458 °F
- Scoring Index	=	3810
- Critical Scoring criterion number	=	6000
Design Acc. to Load Carrying Capacity :-		
<< ISO >>		
- Tooth bending stress.	N/mm² =	202
- Tooth-surface durability.	N/mm² =	358
<< AGMA >>		
- Tooth bending stress.	Ib/in² =	29349
- Tooth-surface durability.	Ib/in² =	51959
<< Power rating >>		
- Transmitted power	Kw =	30
- Max. power Acc. to AGMA	Kw =	48
- Max. power Acc. to Gleason	Kw =	36
- Max. power Acc. to AGMA (based on wear)	Kw =	32
- Max. power Acc. to AGMA for durability	Kw =	32

Table (8) Kinematics, loads, stresses and power rating of skew bevel gears for 30 Kw power, 800 rpm input speed, 3 gear ratio and 90° shaft angle.

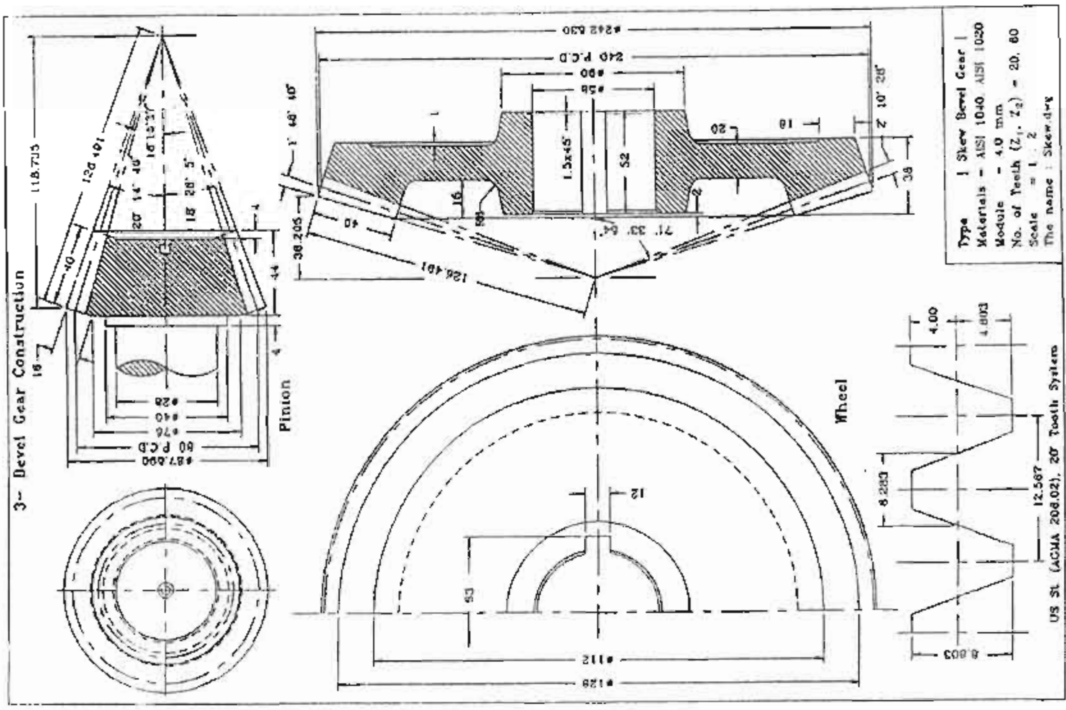


Fig.(9) Construction drawing of skew bevel gears pinion and wheel for 30 Kw power, 800 rpm input speed, 3 gear ratio and 90° shaft angle.

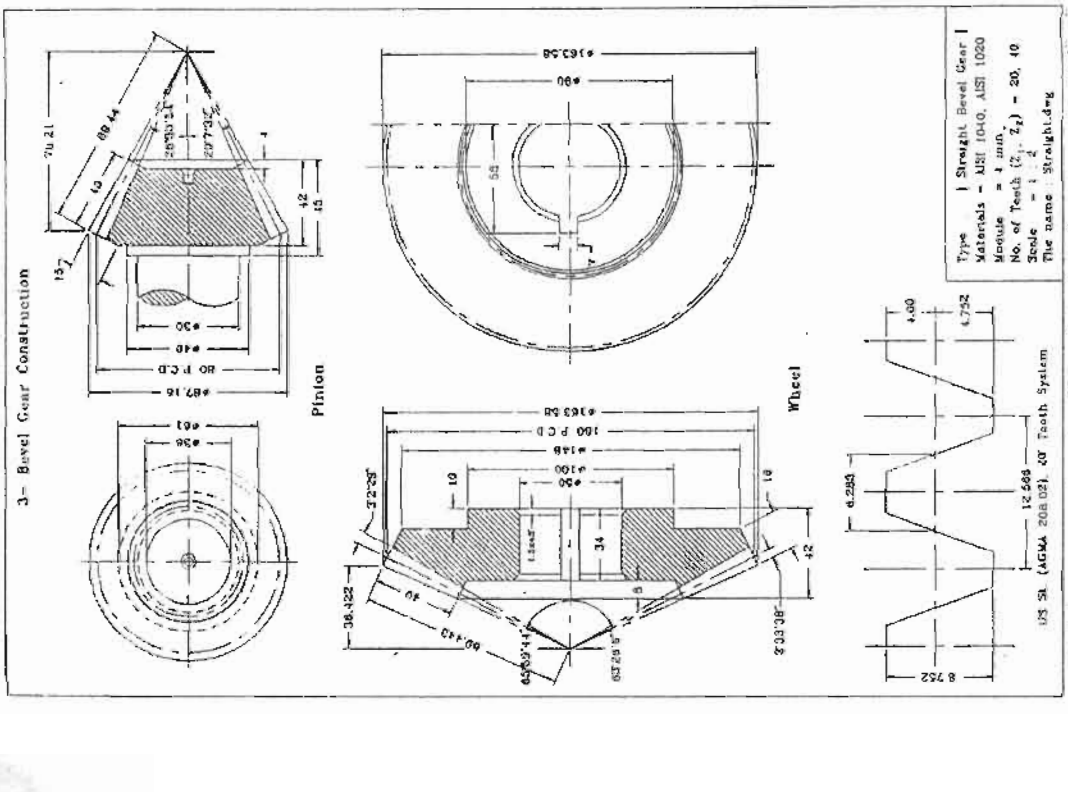


Fig.(8) Construction drawing of straight bevel gears pinion and wheel for 25 Kw power, 750 rpm input speed, 2 gear ratio and 90° shaft angle.

Input Data		
- Power.	Kw	= 100
- Input speed.	rpm	= 3000
- Gear ratio		= 4.0000
- Shaft angle		= 90
- Design according to "US"		

Output Data
1- Specifications and Geometry
2- Kinematics, Loads and Stresses
3- Gear Construction

1- Specifications and Geometry			
Item		Pinion	Wheel
<< Spiral Bevel Gear >>			
- Material		AISI 1040	AISI 1020
- Number of teeth		20	80
- Virtual number of teeth		38	600
- Addendum.	mm	2.8000	2.8000
- Dedendum.	mm	4.0000	4.0000
- Tooth thickness.	mm	6.2832	6.2832
- Pitch circle diam..	mm	80.0000	320.0000
- Tip circle diam..	mm	85.4328	321.3582
- Middle circle diam..	mm	70.2986	281.1943
- Back cone distance.	mm	41.2311	659.6972
- Crown height.	mm	159.3209	37.2836
- Profile correction.	mm	-	-
- Pitch angle		14° 2' 10"	75° 57' 49"
- Addendum angle		0° 58' 21"	0° 58' 21"
- Dedendum angle		1° 23' 21"	1° 23' 21"
- Root angle		12° 38' 48"	74° 34' 27"
- Face angle		15° 0' 31"	76° 56' 11"
- Spiral angle		35°	
- Shaft angle		90°	
- Pressure angle		20.0°	
- Module	mm	4.00	
- Middle module	mm	3.5149	
- Working depth.	mm	6.8000	
- Whole depth.	mm	7.5520	
- Circular pitch.	mm	12.5664	
- Fillet radius.	mm	1.2000	
- Clearance.	mm	0.7520	
- Backlash.	mm	0.1975	
- Face width.	mm	40.0000	
- Outer cone dist..	mm	164.9242	
- No. teeth in crown gear		82	
- Gear ratio		4.0000	
- Profile contact ratio		1.1190	
- Face contact ratio		2.5522	
- Contact ratio		2.7867	
- US St. (AGMA 209.02).	20°	Tooth System	

Table (9) Specifications and geometry of spiral bevel gears for 100 Kw power, 3000 rpm input speed, 4 gear ratio and 90° shaft angle.

2- Kinematics-Loads and Stresses		
- Speed for pinion & wheel.	rpm	= 3000 . 750
- Pitch line velocity.	m/sec	= 12.566
- Sliding velocity.	m/sec	= 4.762
- Efficiency		= 96.51%
Force Analysis :-		
- Normal load .	N	= 1054
- Tan., Rad. & Ax. loads for pinion.	N	= 812. 488. -464
- Tan., Rad. & Ax. loads for wheel.	N	= 812. -464. 488
Design Acc. to Bending Strength Failure :-		
- Calc. module Acc. to M.L.E..	mm	= 2.61< 4.00 Safe
A- Loads		
- Tan. load Acc. to M.L.E..	N	= 13450
- Dynamic load Acc. to Backingham.	N	= 13549
- Tan. load Acc. to AGMA.	N	= 12556
- Tan. load Acc. to Gleason.	N	= 13561
- Tan. load for pinion Acc. to B.S.	Ib(N)	= 5104(22706)
- Tan. load for wheel. Acc. to B.S.	Ib(N)	= 4928(21923)
B- Stresses		
- Max. allowable stress.	N/mm²	= 287
- Bending stress Acc. to M.L.E..	N/mm²	= 127
- Bending stress Acc. to AGMA.	N/mm²	= 179
- Bending stress Acc. to Gleason.	N/mm²	= 166
Design Acc. to Surface Failure :-		
A- Loads		
- Wear load Acc. to Buckingham.	N	= 35144
- Wear load Acc. to AGMA.	N	= 11621
- Wear load Acc. to Gleason.	N	= 5790
- Wear load for pinion Acc. to B.S.	Ib(N)	= 8964(39970)
- Wear load for wheel Acc. to B.S.	Ib(N)	= 8891(39555)
B- Stresses		
- Max. allowable contact stress.	N/mm²	= 841
- Hertzian contact stress.	N/mm²	= 585
- Calc. contact stress Acc. to AGMA.	N/mm²	= 696
- Gleason (Pitting).	N/mm²	= 719
- Blank temperature		= 150 °F
- Flash temperature		= 215.974 °F
- Scoring Index		= 6754
- Critical Scoring criterion number		= 14000
Design Acc. to Load Carrying Capacity :-		
<< ISO >>		
- Tooth bending stress.	N/mm²	= 139
- Tooth-surface durability.	N/mm²	= 336
<< AGMA >>		
- Tooth bending stress.	Ib/in²	= 20203
- Tooth-surface durability.	Ib/in²	= 48986
<< Power rating >>.		
- Transmitted power	Kw	= 100
- Max. power Acc. to AGMA	Kw	= 180
- Max. power Acc. to Gleason	Kw	= 158
- Max. power Acc. to AGMA (based on wear)	Kw	= 144
- Max. power Acc. to AGMA for durability	Kw	= 107

Table (10) Kinematics, loads, stresses and power rating of spiral bevel gears for 100 Kw power, 3000 rpm input speed, 4 gear ratio and 90° shaft angle.

Input Data		
- Power.	Kw	= 100
- Input speed.	rpm	= 3000
- Gear ratio		= 3.0000
- Shaft angle		= 120
- Design according to "US"		

Output Data
1- Specifications and Geometry
2- Kinematics, Loads and Stresses
3- Gear Construction

1- Specifications and Geometry			
Item		Pinion	Wheel
<< Spiral Bevel Gear >>			
- Material		AISI 1040	AISI 1020
- Number of teeth		20	60
- Virtual number of teeth		39	578
- Addendum.	mm	4.8560	1.9440
- Dedendum.	mm	3.9468	6.6588
- Tooth thickness.	mm	7.6131	4.9533
- Pitch circle diam..	mm	80.0000	240.0000
- Tip circle diam..	mm	89.1770	240.7348
- Middle circle diam..	mm	66.9069	200.7208
- Back cone distance.	mm	42.3320	634.9805
- Crown height.	mm	113.8806	21.1850
- Profile correction.	mm	-	-
- Pitch angle		19° 6' 23"	100° 53' 36"
- Addendum angle		2° 16' 32"	0° 54' 41"
- Dedendum angle		1° 50' 59"	3° 12' 44"
- Root angle		17° 15' 24"	97° 40' 51"
- Face angle		21° 22' 55"	101 48' 17"
- Spiral angle		35°	
- Shaft angle		120°	
- Pressure angle		20.0°	
- Module	mm	4.00	
- Middle module	mm	3.3453	
- Working depth.	mm	6.8000	
- Whole depth.	mm	8.8028	
- Circular pitch.	mm	12.5664	
- Fillet radius.	mm	1.0000	
Clearance.	mm	2.0028	
Backlash.	mm	0.1975	
- ace width.	mm	40.0000	
- ter cone dist..	mm	122.2020	
- N teeth in crown gear		61	
- G ratio		3.0000	
- Pr le contact ratio		1.2658	
- Fac contact ratio		2.6961	
- Cont t ratio		2.9785	
- USA s (AGMA-209.02), 20° Tooth System			

Table (11) Specifications and geometry of spiral bevel gears for 100 Kw power, 3000 rpm input speed, 3 gear ratio and 120° shaft angle.

2- Kinematics-Loads and Stresses		
- Speed for pinion & wheel.	rpm	= 3000 , 1000
- Pitch line velocity.	m/sec	= 12.566
- Sliding velocity.	m/sec	= 4.355
- Efficiency		= 96.59%
Force Analysis :-		
- Normal load .	N	= 1054
- Tan., Rad. & Ax. loads for pinion.	N	= 812, 527, -419
- Tan., Rad. & Ax. loads for wheel.	N	= 812, -490, 462
Design Acc. to Bending Strength Failure :-		
- Calc. module Acc. to M.L.E..	mm	= 2.57< 4.00 Safe
A- Loads		
- Tan. load Acc. to M.L.E..	N	= 13007
- Dynamic load Acc. to Backingham.	N	= 13700
- Tan. load Acc. to AGMA.	N	= 12622
- Tan. load Acc. to Gleason.	N	= 13631
- Tan. load for pinion Acc. to B.S.	lb(N)	= 5033(22390)
- Tan. load for wheel. Acc. to B.S.	lb(N)	= 4587(20409)
B- Stresses		
- Max. allowable stress.	N/mm²	= 287
- Bending stress Acc. to M.L.E..	N/mm²	= 126
- Bending stress Acc. to AGMA.	N/mm²	= 178
- Bending stress Acc. to Gleason.	N/mm²	= 165
Design Acc. to Surface Failure :-		
A- Loads		
- Wear load Acc. to Buckingham.	N	= 44927
- Wear load Acc. to AGMA.	N	= 11681
- Wear load Acc. to Gleason.	N	= 5636
- Wear load for pinion Acc. to B.S.	lb(N)	= 8859(39413)
- Wear load for wheel Acc. to B.S.	lb(N)	= 8277(36823)
B- Stresses		
- Max. allowable contact stress.	N/mm²	= 841
- Hertzian contact stress.	N/mm²	= 584
- Calc. contact stress Acc. to AGMA.	N/mm²	= 694
- Gleason (Pitting).	N/mm²	= 717
- Blank temperature		= 150 °F
- Flash temperature		= 251.969 °F
- Scoring Index		= 6754
- Critical Scoring criterion number		= 14000
Design Acc. to Load Carrying Capacity :-		
<< ISO >>		
- Tooth bending stress.	N/mm²	= 153
- Tooth-surface durability.	N/mm²	= 338
<< AGMA >>		
- Tooth bending stress.	lb/in²	= 22188
- Tooth-surface durability.	lb/in²	= 48968
<< Power rating >>		
- Transmitted power	Kw	= 100
- Max. power Acc. to AGMA	Kw	= 172
- Max. power Acc. to Gleason	Kw	= 159
- Max. power Acc. to AGMA (based on wear)	Kw	= 145
- Max. power Acc. to AGMA for durability	Kw	= 107

Table (12) Kinematics, loads, stresses and power rating of spiral bevel gears for 100 Kw power, 3000 rpm input speed. 3 gear ratio and 120° shaft angle.

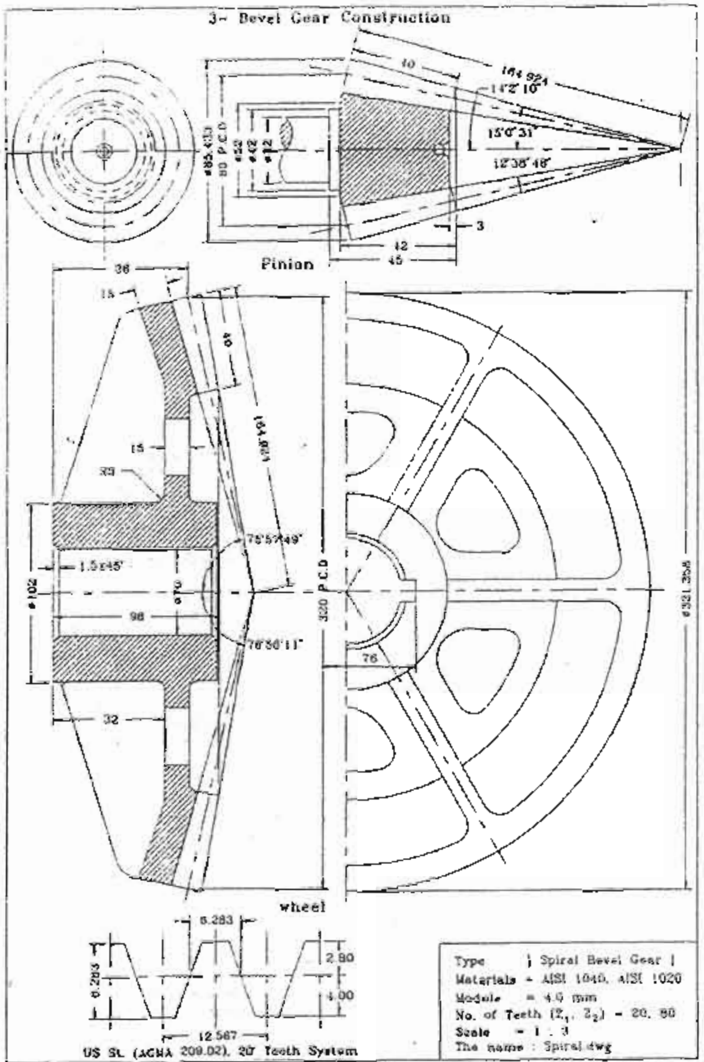


Fig.(10) Construction drawing of spiral bevel gears pinion and wheel for 100 Kw power, 3000 rpm input speed, 4 gear ratio and 90° shaft angle.

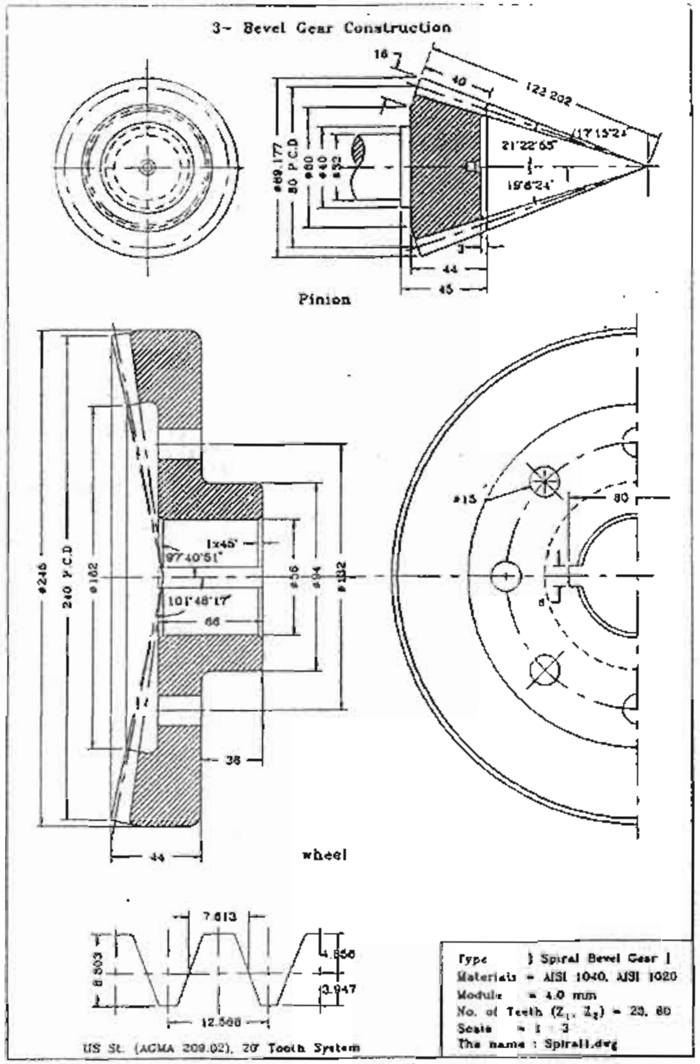


Fig.(11) Construction drawing of spiral bevel gears pinion and wheel for 100 Kw power, 3000 rpm input speed, 3 gear ratio and 120° shaft angle.