CAD FOR GEARS

Part 1

Spur, Helical and Double Helical Gears

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ABSTRACT

The aim of this paper is to construct e 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. The main elements of the design in this program are : input data, type end shape of gear tooth system (IS, US, BS and DIN), module, minimum number of teeth to aviod interference, tooth profile modification, centre distance, contact ratio, face width, helix angle, material, equations for bending strength (Lewis, modified Lewis and Buckingham) and surface durability, and load carrying capacity for bending strength and surface durability using ISO, AGMA 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, gear with two walls 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 sava in time and accuracy of the results.

NOMENCLATURE

English letters

- a centre distance, mm
- CR contact ratio
- C_m durability load-distribution factor
- C_a application factor
- C_v velocity factor
- d pitch diameter of pinion or wheel, mm
- $E_{1,2}$ modulus of elasticity for pinion and wheel respectively, Kp/mm^2
- F face width, mm
- h_f dedendum, mm
- J bending geometry factor
- K_A application factor
- $K_{F\alpha}$ Transverse load-distribution factor for bending stress
- $K_{H\beta}$ longitudinal load-distribution factor for pitting
- k_p pitch factor $= P^{0.a}$
- k_1 service factor
- k_2 form-factor coefficient
- L hub length, mm
- $m_{1,2}$ effective masses of the pinion and wheel respectively, slugs
- b dedendum, mm
- C_k geometry factor for durability
- C_p coefficient for elastic properties of the materials used
- c_f factor
- $d_{P,G}$ pitch diameter of pinion and wheel respectively, mm
- e measured error in action, mm
- h_k addendum, mm
- I durability geometry factor
- j number of arms
- $K_{F\beta}$ longitudinal load-distribution factor for bending stress
- $K_{H\alpha}$ transverse load-distribution factor for pitting
- k_f fatigue stress-concentration factor
- $k_{p,w}$ tooth correction factor for pinion and wheel respectively
- K_v dynamic factor
- m module, mm
- m_n normal module, mm
- m_p profile contact ratio

m_G gear ratio
P_d diametral pitch
P_t transverse circular pitch, mm
$R_{1,2}$ pitch circle radius of pinion and wheel respectively, mm
r_a, r_b addendum circle radius and base circle radius, mm
S_{bo} basic bending stress, lb/in^2
S_{co} basic surface stress, lb/in^2
t number of wheel teeth
T_b blank temperature, ^oF
V tangential velocity, mt/sec
W_a acceleration load, N
W_t transmitted load, N
W_1 average force required to accelerate the masses, N
W_w limiting load for wear, N
$x_{1,2}$ tooth correction factor for pinion and wheel respectively
Y form factor
Y_β helix angle factor
Y_s stress-concentration factor (HPSTC)
Y_{Fa} tooth form factor (tip loading)
Y_ϵ contact ratio factor
Z_E coefficient for elastic properties of the materials used
Z' zone factor

P circular pitch, mm
P_n normal diametral pitch
r pitch circle radius, mm
r_{ao} edge radius ,mm
r_t edge radius ,in
r_f fillet radius, mm
S_t calculated tensile stress at root tooth, lb/in^2
T number of teeth of pinion
T_f flash temperature, ^oF
U number of wheel teeth/ number of pinion teeth
W_d dynamic load, N
W_{tmax} max. transmitted load, N
W_2 force required to deform the teeth through amount of effective error, N
x tooth correction factor
X_b speed factor for strength
X_c speed factor for wear
Y' strength factor
Y_F tooth form factor (HPSTC)
Y_{sa} stress-concentration factor (tip loading)
Z minimum number of teeth
$Z_{H,\beta,\epsilon}$ total durability geometry factor
Z_1, Z_2 number of pinion and wheel teeth
Z_{min} minimum number of teeth

Greek letters

α pressure angle, (metric) deg
α_t transverse pressure angle, deg
λ factor
σ_b bending stress, N/mm^2 (lb/in^2)
σ_{ef} surface endurance limits, N/mm^2(lb/in^2)
α_w working pressure angle, deg
β helix angle
ϕ pressure angle (English), deg
σ_e endurance limits, N/mm^2
ρ_1, ρ_2 tooth radius of curvature of the pinion and wheel respectively, mm

INTRODUCTION

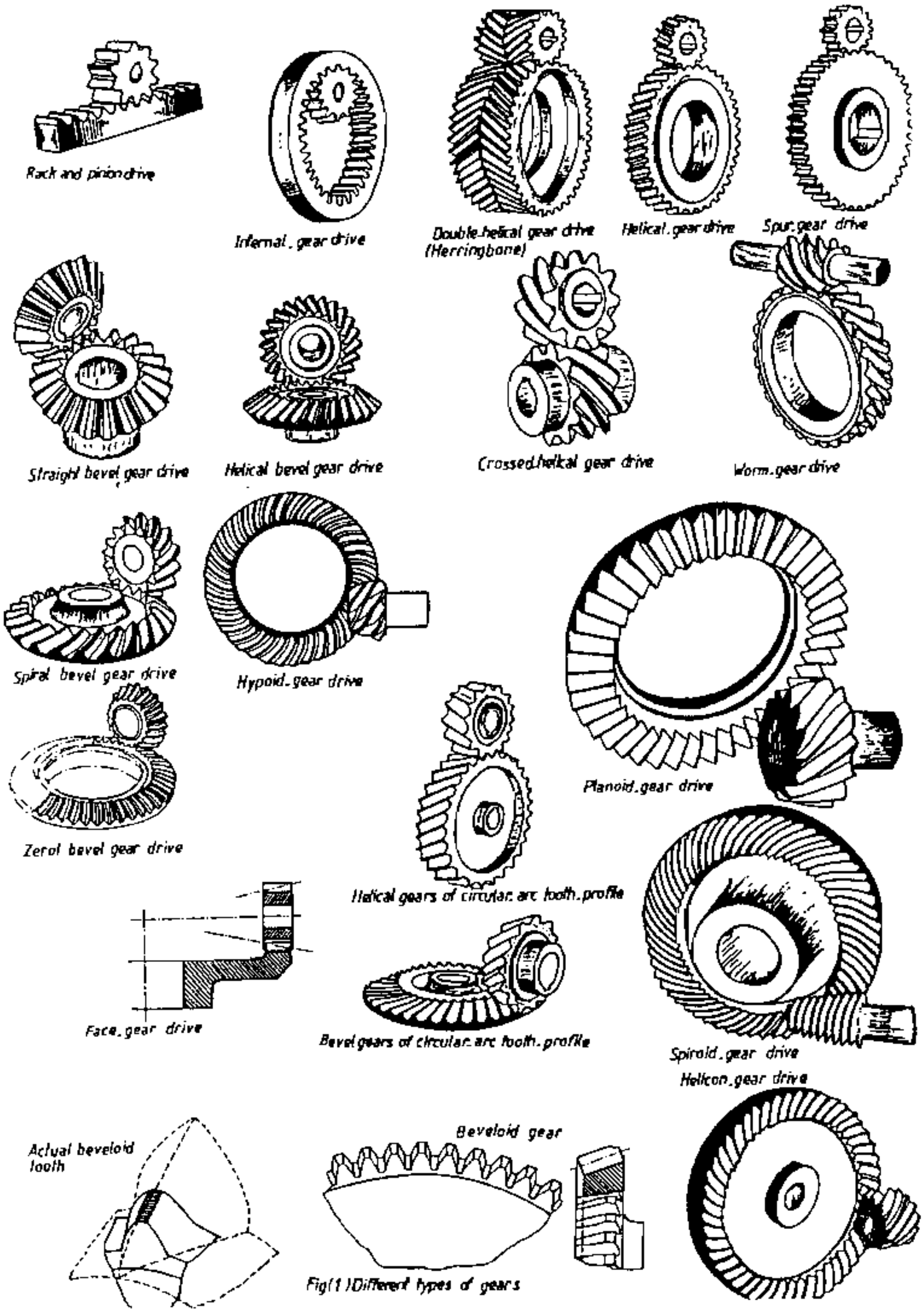
Gear drives are the most commonly used of all types of drives. They are used for parallel, intersecting, and crossed shafts, for the lowest to the highest horsepower, speeds, and overall speed ratios. They are distinguished for their slip-less force transmission, high reliability, long life, ability to take overloads, easy maintenance, compactness, and high efficiency, despite, their higher price and somewhat higher noise level. Different types of gears are shown in Fig(1). These gears are generally categorized into three distinct types according to the types of shafts;

a- For parallel shafts, namely,
spur gears (external and internal), helical gears (external and internal) and herringbone or double-helical gears (external and internal).

b- For intersecting shafts using
bevel gears (straight, zerol and spiral), face gears and beveloid gears.

c- For crossed shafts (non intersecting and/or non parallel)
crossed helical gear, single and double-enveloping worm, hypoid gear, spiroid gear, planoid gear, beveloid gear, face gear and helicon gear.

In recent times, the gear design has become a highly complicated and comprehensive subject. Designer of a modern gear drive system must remember that the main objective of a gear drive is to transmit higher power with comparatively smaller overall dimensions of the driving system which can be constructed with minimum possible manufacturing cost, run reasonably free of noise and vibration, and require little maintenance. He has to satisfy, among others, the above conditions and design accordingly, so that the design is sound as well as economically viable.



Fig(1)Different types of gears

The most important stresses which should be considered for the 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 pitting and scoring are also considered.
The precise computation of gear capacity is an extremely difficult process since a large number of veriables are involved such as gear cutting accuracy, mounting errors, elastic deflections, material quality, tooth stiffness, characteristics of connected machinery, and many others.

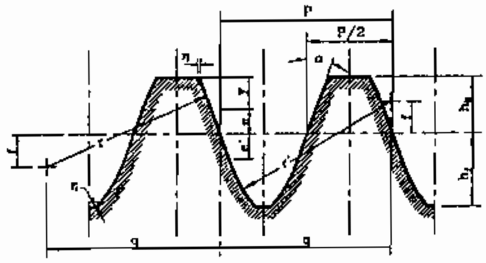
Due to the above problems, gear design is a complex and time-consuming task which must satisfy numerous design constraints. It is desireble to use CAD techniques to accomplish this task. Many investigators [1-6] have attempts for the gear tooth design using the computer, but to the author's knowledge there is no complete work that has been done on the gear design using CAD technique.

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

DESIGN APPROACH

1- Types of Gear Tooth Systems

The reference profiles of the tooth (basic racks) of IS, US st., BS and DIN [7-14] are shown in table (1). Full depth 20° involute system is the most widely used tooth system. This system elleviates the interference and undercutting problems and gives a stronger root section. Full depth 14.5° involute system gives a number of gear teeth which are large enough to avoid undercutting. Stub-tooth system reduces the interference problem and is of greater strength, lower cost and small sliding velocity, but the contact ratio is less then that for the full depth. This system gives better results when the pinion has less than 25 teeth.



Des.	Standards	Tooth system	α°	h_a/m	h_f/m	r_f/m	r/m	η/m	y/m
IS St.	IS:2535-1978	20° Full Depth Syst.	20	1.00	1.25	0.400	without	0.02	≤0.6
US St.	ASA-B 6.1/1932	14.5° Composite Syst.	14.5	1.00	1.157	0.209	3.7287	—	—
	ASA-B 6.1/1932	14.5° Full Depth Syst.	14.5	1.00	1.175	≥0.209	without	—	—
	ASA-B 6.1/1932	20° Full Depth Syst.	20	1.00	1.175	≥0.235	without	—	—
	ASA-B 6.1/1932	20° Stub Tooth Syst.	20	0.80	1.000	≥0.300	without	—	—
	ASA-B 6.7/1950	20° Fine Pitch Syst.	20	1.00	1.200	0.000	without	—	—
	AGMA 218.01	25° Full Depth Syst.	25	1.00	1.350	≥0.235	without	—	—
BS St.	BSS 436/1940	Class A 1	20	1.00	1.440	0.295	≥15.750	0.009	0.493
	BSS 436/1940	Classes A 2 and a	20	1.00	1.250	0.390	≥15.750	0.009	0.493
	BSS 436/1940	Classes C and D	20	1.00	1.250	0.390	≥12.875	0.019	0.628
DIN St.	DIN 867		20	1.00	1.1:1.3	0.200	without	—	—

$q = 3.72783m$, $f = 0.56278m$

Table (1) Reference profiles (basic racks) of IS, USA, British and DIN standards

2- 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, tooth profile modification and gear reduction ratio according to the following equations [8,10,11] :

$Z=2/\sin^2\alpha$ (1)

$Z=2.h_{1x}/m.\sin^2\alpha$ Metric where $h_{1x}=h_1-r_{ao}(1-\sin\alpha)$ (2-a)

$N=2x.P_1/\sin^2\phi$ English where $x = b-r_t(1-\sin\phi)$ (2-b)

$Z_1^2+2Z_1Z_2=4(1+Z_2)/\sin^2\alpha$ or $(Z_1^2/Z_2)+2Z_1=(4/Z_2\sin^2\alpha)+(4/\sin^2\alpha)$ (3)

3- Tooth Profile Modification

According to the design aspects, there are two broad categories of the tooth profile modifications depending on the required design for changing or not changing the centre distance. For not changing the centre distance, the two components of the mating pair of gears receive numerically equal correction factors, but these two factors are algebraically of opposite signs. The pinion is provided with positive correction and the gear with negative correction. This type is also known as the "long and short addendum" system [11].

$x=(Z_{min}-Z)/Z_{min}$, $x_1+x_2 = 0$ (4)

New dimensions of tha gearing system are as follows;

	Pinion	Wheel	
Tip circle diameter	$d_1+2m+2x_1m$	$d_2+2m-2x_1m$	(5)
Root circla diameter	$d_1-2(1.25-x_1)m$	$d_2-2(1.25+x_1)m$	(6)
Tooth thickness on the pitch circle	$(\pi m/2)+2x_1m\ \tan\alpha$	$(\pi m/2)-2x_1m\ \tan\alpha$	(7)

For changing the centre distance the sum of the profile corrections of the two mating gears is not equal to zero. It is eithar positive or negative

$x_1+x_2=(Z_1+Z_2)(inv\alpha_w-inv\alpha)/(2\ \tan\alpha)$ (8)

Actual centre distance, a $=m(Z_1+Z_2)\cos\alpha/(2\ \cos\alpha_w)$ (9)

Addendum modification , y_m $=m(Z_1+Z_2)/2+(x_1m+x_2m)-a$ (10)

Also addendum modification of spur gears is given as follows according to [8,10]. When the tooth sum is equal to 60 or more, addendum coefficient of the pinion is

$k_p=0.4(1-t/T)$ with a minimum value of $k_p=0.02(30-t)$ and $k_w=-k_p$ (11)

When the tooth-sum is less than 60, the sum of the addendum coefficients is equal to

$k_p+k_w=0.02[60-(T+t)]$ and $k_w=0.02(30-T)$ (12)

Another addendum modification of spur gear is given in table (2) according to [15].

Tooth ratio Z_1/Z_2	20° pressure angle		25° pressure angle	
	Pinion add.	Wheel add.	Pinion add.	Wheel add.
12/35	1.24	0.76	1.16	0.84
12/50	1.32	0.68	1.22	0.78
12/75	1.38	0.62	1.25	0.75
12/125	1.44	0.56	1.25	0.75
12/∞	1.48	0.52	1.25	0.75
16/35	1.15	0.85	1.10	0.90
16/50	1.22	0.78	1.15	0.85
16/75	1.27	0.73	1.18	0.82
16/125	1.32	0.68	1.21	0.79
16/∞	1.36	0.64	1.24	0.76
24/35	1.06	0.94	1.04	0.96
24/50	1.11	0.89	1.08	0.92
24/75	1.15	0.85	1.10	0.90
24/125	1.20	0.80	1.13	0.87
24/∞	1.24	0.76	1.16	0.84

Table (2) Long and short addendum for speed-reducing spur gears.

4- Contact Ratio

To ensure smooth and continuous running, the contact ratio CR must be as high as possible, which the limiting factors permit. Definite values are difficult to specify, a lower contact ratio also necessitates a higher degree of accuracy in cutting.

For spur gear, CR
$$CR = \frac{(r_{a_1}^2 - r_{b_1}^2)^{0.5} + (r_{a_2}^2 - r_{b_2}^2)^{0.5} - (a.\sin\alpha)}{(\pi m.\cos\alpha)} \quad (13)$$

For a gear mating with a rack,
$$CR = \frac{(r_a^2 - r_b^2)^{0.5} - (r.\sin\alpha) + (h_r/\sin\alpha)}{(\pi m.\cos\alpha)} \quad (14)$$

Contact ratio for the helical gearing is the summation of the profile and face contact ratios as;

$$CR = \frac{(r_{a_1}^2 - r_{b_1}^2)^{0.5} + (r_{a_2}^2 - r_{b_2}^2)^{0.5} - (a.\sin\alpha_1)}{P_t \cos\alpha_t} + \frac{F.\tan\beta}{P_t} \quad (15)$$

5- Face Width

Face width is a very important factor for designing and determining the load carrying capacity of the gears. Selection of this value depends on multi variables such as pitch diameter of the gear, module, centre distance, circular pitch and axial pitch. Relations between the face width and the above variables are written as follows:

a- *Face width with pitch diameter:*
- Face width should not exceed twice pinion diameter
- Face width should not be less than 1/10 wheel diameter
- With rigid straddle mounting $F/d \leq 1.2$, With overhanging gears $F/d \leq 0.75$

b- *Face width with centre distance:*
- $F = 0.5a$ for single step gearing
- For double reduction gear box, $F_1 = a/3$ for the first step, $F_2 = 2F_1$, for the second step. The ratio F/a may however be taken the same for all the steps and the centre distances in geometric progression $a_I : a_{II} : a_{III} = 100 : 125 : 160$

c- *Face width with circular pitch:*
- $F = c_f P$, c_f should not exceed 5 or 6 except for profile group and shaved gears where c_f may be as much as 8.

d- *Face width with axial pitch:*
- Face width should be at least twice the axial pitch.

e- *Face width with module:*
- $F_{max} = \lambda m$, λ depends on the surface finish of the gear tooth and type of bearing carrying the gear shaft;
- $\lambda = 10$ for cast clean, smooth tooth and bearing fitted on steel construction
- For machined smooth or ground tooth;
- $\lambda = 15$ bearing is fitted on steel construction
- $\lambda = 15$ bearing with overhung pinion
- $\lambda = 25$ bearing is fitted in gear box casing
- $\lambda = 30$ bearing with rigid base using shaft with sufficient stiffness.

Normally, the width of the pinion is made 3 to 4 mm greater than of the wheel to ensure complete engagement during service.

6- Helix Angle

The helix angle is an important criterion of the design of helical gears as the gear dimensions, centre distance of drive and thrust forces depend on its magnitude and orientation.

Hence, helix angle can be calculated from the equation

$$\beta = \tan^{-1}(1.15\frac{P_t}{F}) \quad (16)$$

For normal application helix angle ranging from 8° to 20°. Helix angle should not cross 30° to avoid a large resultant axial thrust.

Double helix gears have helix angles that typically range from 30° through 45°. Although higher helix angles provide smoother operations, the tooth strength is lower.

7- Design of Gear Tooth According to Bending Strength

Geer tooth design according to bending strength is divided into three steps using Lewis, modified Lewis and Buckingham equations as follows :-

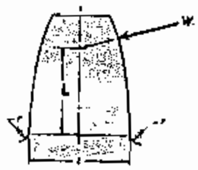
- *Lewis Equation,* $\sigma_b = W_t / (F\ m\ Y)$ (17)

$Y = \pi(0.124 - 0.684/Z)$ For 14.5° composite and Full depth involute system
$Y = \pi(0.154 - 0.912/Z)$ For 20° Full depth involute system
$Y = \pi(0.175 - 0.841/Z)$ For 20° stub system

In helical gears, the virtual number of teeth must be used to determine the form factor Y.

$Z_v = Z/\cos^3\beta$

- Modified Lewis equation,



$$W_{t_{max}} = \frac{F.m.K_2.Y.C_v}{K_f.K_1}\sigma_e \qquad \text{For spur gear} \qquad (18\text{-}a)$$

$$W_{t_{max}} = \frac{F.m.Y.C_v}{K_f.K_1.m_p}\sigma_e \qquad \text{For helical gear} \qquad (18\text{-}b)$$

$K_f = 0.18 + (t/r)^{0.15}(t/L)^{0.45}$ For 20° Involute teeth
$K_f = 0.22 + (t/r)^{0.20}(t/L)^{0.40}$ For 14.5° Involute teeth
K_1 = 1.25 For steady loads, 1.35 For pulsating loads, and 1.50 for shock loads.
K_2 = 1.7 For full-depth teeth , 1.6 For stub teeth
$C_v = 3/(3+V)$, V up to 12.5 m/sec , ordinary cut gears
$C_v = 4.5/(4.5 + V)$, V up to 12.5 m/sec , carefully cut gears
$C_v = 6/(6 + V)$, V up to 20 m/sec, very accurately cut and ground metallic gears
$C_v = 0.75/(0.75 + \sqrt{V})$, V up to 20 m/sec , precision cut gears with high accuracy
$C_v = (0.75/(1+V)) + 0.25$ non-metallic gears

- *Buckingham Equation,* $W_d = W_1 + \sqrt{W_a(2W_2 - W_a)}$ (19)

$$W_a = \frac{W_1.W_2}{(W_1 + W_2)}$$

$$W_1 = \frac{c_1 m_1 m_2}{(m_1+m_2)}(1/R_1 + 1/R_2)V^2 \ , \quad W_2 = \frac{F.e}{c_2(1/E_1 + 1/E_2)} + W_t \qquad \text{For spur gear}$$

$$W_1 = \frac{c_1 m_1 m_2}{(m_1+m_2)}(1/R_1 + 1/R_2)V^2\cos^2\beta \ , \quad W_2 = \frac{F.e}{c_2(1/E_1 + 1/E_2)}\cos^2\beta + W_t \qquad \text{For helical gear}$$

C_1 = 0.00086 For 14.5° gears , 0.00120 For 20° gears
C_2 = 9.345 For 14.5° gears . 9.000 For 20° Full-depth gears , 8.700 For 20° stub gears

8- Design of Gear Tooth According to Surface Failure

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

- *Limiting load for wear :-*

$$W_w = \frac{1.43 F d_p d_G}{(d_p + d_G)}(1/E_1 + 1/E_2)\sigma_{es}^2 \sin\alpha \qquad \text{For spur gear} \qquad (20)$$

$$W_w = \frac{1.43 F d_p d_G}{(d_p + d_G)\cos^2\beta}(1/E_1 + 1/E_2)\sigma_{es}^2 \sin\alpha \qquad \text{For helical gear} \qquad (21)$$

- *Contact stress for spur gear;* $S_c = \sqrt{\dfrac{0.7}{(1/E_1 + 1/E_2)\cos\phi\sin\phi}}\ \sqrt{\dfrac{W_t}{Fd}\dfrac{(m_G + 1)}{m_G}}$ (22)

- *Contact stress for helical gear;* $S_c = \sqrt{\dfrac{0.7\ \cos^2\beta}{m_p(1/E_1 + 1/E_2)\cos\phi_n\sin\phi_n}}\ \sqrt{\dfrac{W_t}{Fd}\dfrac{(m_G + 1)}{m_G}}$ (23)

- *Scoring-criterion number* $= (\frac{W_t}{F})^{0.75} Z_p^{0.5} m^{0.25}$ (24)

- *Flash temperature;* $T_f = T_b + 0.0175(\frac{W_t}{F})^{0.75} Z_p^{0.5} \dfrac{[\sqrt{\rho_1} - \sqrt{\rho_2/m_G}]}{(\cos\phi_t)^{0.75}[\rho_1\rho_2/(\rho_1+\rho_2)]^{0.25}}$ (25)

9- Design According to Load Carrying Capacity

After the 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 [15] and BS [8] gear rating formulae for bending strength and surface durebility are used and given as follows :

- Total rating stress in bending:

$$S_t=\frac{W_t.K_a.P_n}{K_u.F}K_m\frac{\cos\psi}{J} \qquad AGMA \qquad (26)$$

$$\sigma_F=\frac{W_t}{F.m_n}\ Y_F\ Y_s\ Y_\beta\ K_A\ K_v\ K_{F\beta}\ K_{F\alpha} \qquad \text{for load at HPSTC} \qquad ISO \qquad (27)$$

$$\sigma_F=\frac{W_t}{F.m_n}\ Y_{Fa}\ Y_{sa}\ Y_\varepsilon\ Y_\beta\ K_A\ K_v\ K_{F\beta}\ K_{F\alpha} \qquad \text{for load at tip of tooth} \qquad ISO \qquad (28)$$

Total rating stress for surface durability:

$$S_c=C_p(\frac{W_t.C_a.C_m}{d.F.I.C_v})^{0.5} \qquad AGMA \qquad (29)$$

$$\sigma_H=Z_E(W_t.K_A\frac{(U+1)}{d.F.U}\ K_v\ K_{H\alpha}\ K_{H\beta})^{0.5}\ (Z_H\ Z_\beta\ Z_\varepsilon) \qquad ISO \qquad (30)$$

British standard rating formulae:

$$W_i=S_{bo}.Y.X_b/P \qquad \text{For the strength rating,} \qquad (31)$$

$$W_i=S_{co}.Z.X_c/K_p \qquad \text{For the wear rating} \qquad (32)$$

Horse-power formulae at normal rating

$$\text{Pinion, strength ; h.p.} =\frac{S_{bop}.X_{bp}.Y_p.t.n.F}{126000P^2} \qquad (33)$$

$$\text{Wheel, strength ; h.p.} =\frac{S_{bow}.X_{bw}.Y_w.T.N.F}{126000P^2} \qquad (34)$$

$$\text{Pinion, wear ; h.p.} =\frac{S_{cop}.X_{cp}.Z.t.n.F}{126000P.K_p} \qquad (35)$$

$$\text{Wheel, wear; h.p.} =\frac{S_{cow}.X_{cw}.Z.T.N.F}{126000P.K_p} \qquad (36)$$

For variable Loading:

Duration other than 12 hours per day, torque and speed are constant,

normal rating = actual load x "running time factor", $K_u={}^{7}\sqrt{U/12}$ for strength , $K_u={}^{3}\sqrt{U/12}$ for wear

For variable torque and/or speeds:

U_1 hours at 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,...

The equivalent running time U_e is given by :

$$U_e=U_1+U_2(N_2/N_1)(M_2/M_1)^3+U_3(N_3/N_1)(M_3/M_1)^3+\ \ldots\ldots\ldots \qquad (37)$$

The rating formulae for gear tooth strength and gear surface durability have become very long and difficult in use. Short and simple formulae of AGMA and ISO are used with or without the exact formulas according to the selected condition of the design.

Strength formula :

$$S_t=K_t.U_t.K_d \qquad N/mm^2 \quad or \quad psi \qquad (38)$$

K_t geometry factor for bending strength = c . Cosψ / J

U_t unit load, index for tooth breakage = $W_t/(F.m_n)$ N/mm^2 or $(W_t.P_n)/F$ psi

K_d overall derating for bending strength = $(K_a.K_m.K_s)/K_v$ Metric or English

Durability formula:

$$S_c=C_k(K.C_d)^{0.5} \qquad N/mm^2 \quad or \quad psi \qquad (39)$$

C_k geometry factor for durability = c . $[(m_G+1)\ /\ I.m_G]^{0.5}$

K = k factor, index for pitting

$= W_t(U+1)/(U.F.d_{p_1})$ metric or $= W_t(m_G+1)/(F.d.m_G)$ English

C_d = Overall derating for durability $= C_a.C_m.C_s/C_v$ Metric or English

10- 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 gear constructions are given as follows [16-21] and shown in Fig(2).

10-1 *Integar Gear;* Fig(2-a): this type is used in the case of $d < 2d_1$ or if $d_2 - d_1 = 3m$

10-2 *Solid Gear;* Fig(2-b): this type is used in the case of $2d_1 < d < [14.75m + 60mm]$ or $d \leq 4.7P_c + 2.35$ in

10-3 *Gear with Web;* Fig(2-c): this type is used for $d < 250$ mm or $d \leq 7.5\ P_c + 3.35$ in , $t = 1.75m$ or $t = 0.5P_c + 0.125$ in , $m_1 = \sqrt{0.2Z_{1,2}}/P_d$ in. $d_5 = d_3 - 10m$

hub diameter and length are given

Type of service	Diameter, d_4		Length, L
	Cast Iron	Steel	
Light load, no shock	$1.75d_1$	$1.6d_1$	$L \geq 1.50d_1$
Medium load and shock	$1.85d_1$	$1.7d_1$	$L \geq 1.75d_1$
Heavy load and shock	$2.00d_1$	$1.8d_1$	$L \geq 2.00d_1$

10-4 *Gear with Web end Holes;* Fig(2-d) this type is used for $d \cong 250$ mm , $d_4 = 1.6d_1$, $d_6 = (d_5 + d_4)/2$, $d_7 = (d_5 - d_4)/5$, $t = 0.3F$, $m_1 > 1.5m$, $N = 3$ for $d_7 < 40$ mm , $N = 4$ for $d_7 < 100$ mm , $N = 6$ for $d_7 < 180$ mm.

10-5 *Gear with Arms;* Fig(2-e) this type is used for $d > 400$ mm , Fig(2-f) shows the different cross sections of the arms, Fig(2-g) shows the cast wheels at $d \leq 1000$ mm , $F < 200$ mm *and Fig(2-h) shows the wheel at* $d > 1000$ mm and $F > 200$ mm. $t_r = 1.6m$ to $2m$ or $(0.5P_c - 0.6P_c)$ in , the rim should be taper at the rate of about one inch per ft. toward the centre or $t_r = 1/p_d\ ^3\sqrt{Z_{1,2}/2j}$, $q = t_r$, $j = (1/7$ to $1/8)\ d^{0.5}$ mm $= 4$ to 8, $h_1 = 8m$ to $11m$, $h_2 = 6.5$m to 9m or $h_1 = 0.8d_1$, $h_2 = 0.8h_1$, $c = 0.2h_1$ (but < 10mm) , $s = 0.8c$, $e = 0.2d_1$, $K = 0.8e$, $r = 10mm$ $R \geq 0.5h_1$.

The required section modulus of the arm at its centre, $Z = W_t d\ /\ 2j\ \sigma_{b\ all}$.

10-6 *Gear with Two-walls;* this type is shown in Fig(2-i), dimensions are in inches. $A = d_3/80$, $B = 0.4 + 0.2d_1$, $C = 0.25 + 0.6P_n(1 + Z/1000)$. d_1 = *largest of the three quantities* {$F/2$, $0.3[d_3^2.F]^{1/3}$, $0.18.\ ^3\sqrt{torque}$ lb.in }, F_1 = *largest of the three quantities* [$(d_3 - 1.7d_1)/40$, $0.25 + F/20$, $0.25 + d_1/40$ }, $G = 0.13 + 0.3P_n$ or $F/8$ *wichever is less*, $J_1 = [0.5d_3 - 0.5d_1 - h_1 - B - C - 2M_2]$, $K = F - 2G - 2F_1$, $M_2 = 0.1 + d_3/500$

10-7 *Composite Gear;* this type is shown in Fig(2-j,k), the wheel centre is made of cast iron and the rim is made of steel or alloy steel.

FLOW CHART AND COMPUTER PROGRAM

Construction of the software containing design and drawing of spur, helical and double helical gears covers all requirements of the designers and users of the gears. These requirements are divided and specified into the following:

1- Experienca 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(3).

2- Experience and expartise of the user is efficient, some items are selected or assumed such as module and/or material, type of gear and helix angle. This facility gives minimum running time of the program and minimum cost for the design.

3- Improving the design and performance of the gear in service or old design by feeding the software with some information. The program calculates the new required dimensions, specifications and new drawings.

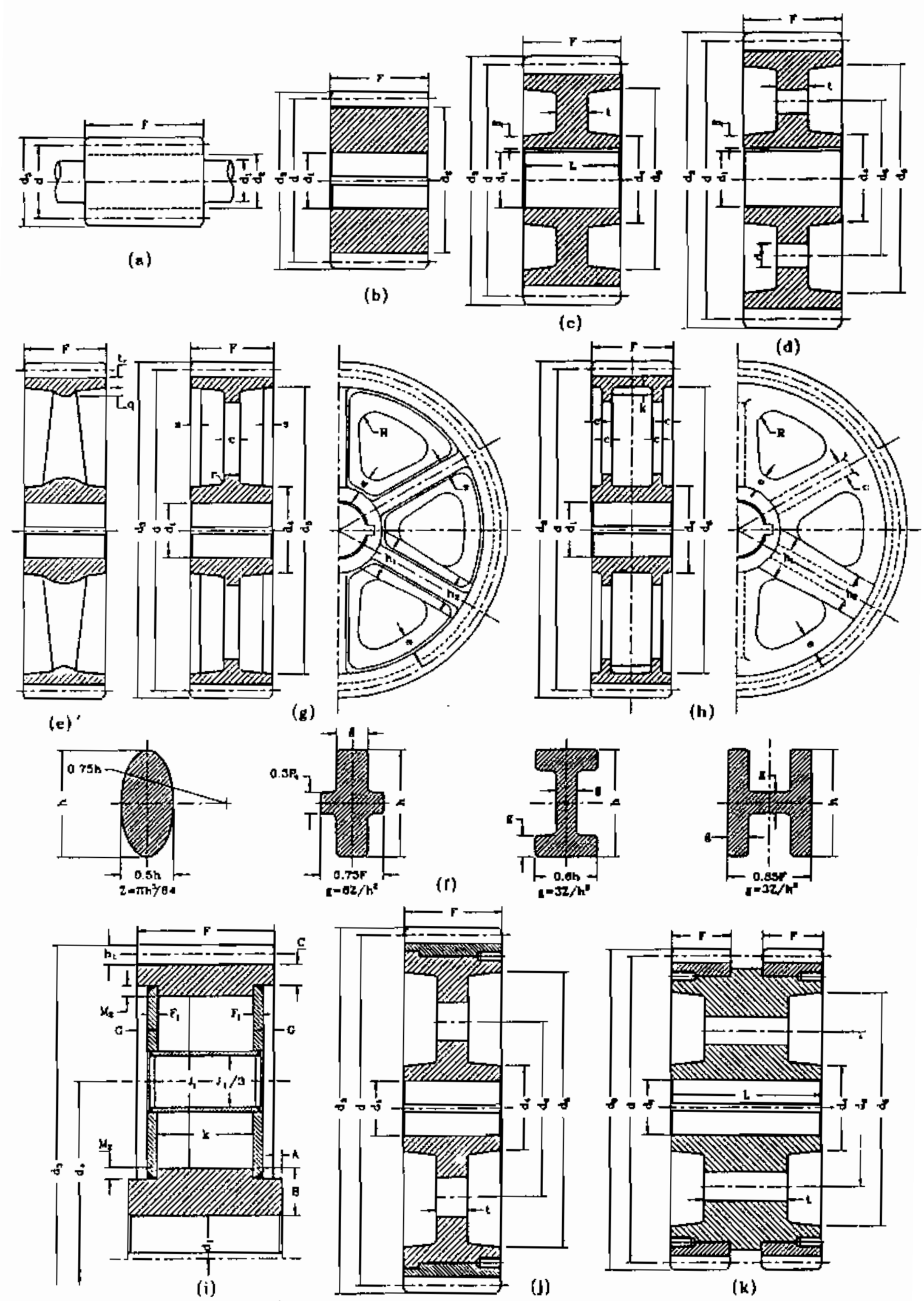


Fig.(2) Different types of gear constructions.

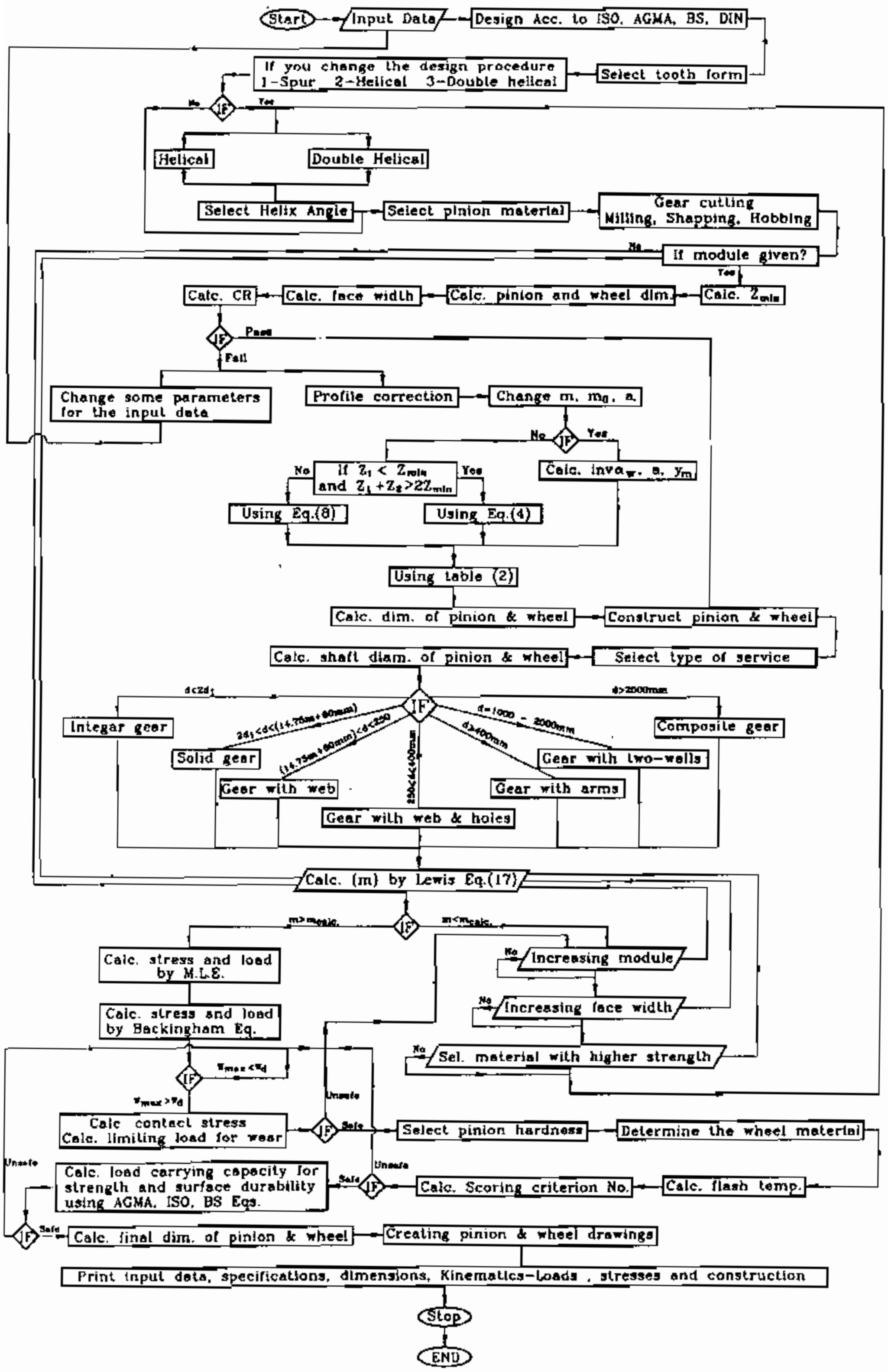


Fig.(3) Flow chart.

4- Obtaining the specifications and the dimensions of the gear to manufacture a new gear instead of an old one broken in service.

The program is written in Turbo Basic Language [22]. The compiler of the program with Turbo Basic gives an executable file to run on the Dos prompt. The program has created and constructed autometically two files which include all information on designing end drawing the gear. Types of these files are the DXF file and SCR file. The DXF and SCR files format are famillar with AutoCAD program [23,24]. 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(4) shows the different menua of the software. For example input power=25 Kw, speed=1500 rpm, number of pinion teeth=20 and gear ratio=4.

COMPUTATIONAL RESULTS AND DISCUSSION

Complete output of any run is divided into three items;

1. Specificetions 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 1500 rpm input data are shown in table (3,4) and Fig.(5); those of 50 Kw and 2000rpm input deta are shown in table (5,6) and Fig.(6) and those of 100 Kw and 3000 rpm input data table (7,8) and Fig.(7).

From these tables and figures, it is clear that the variety of output results according to the input data. Spur gear are given for the smallest power. For Increasing power and speed helical gears are given. For more increasing power and speed double helical gears are given.

Tables (9,10) and Fig(6) show the specifications, geometry, kinematics, loads, stresses, pinion and wheel constructions of 25 Kw, 1500 rpm and 250 mm centre distance input data. These results show that, when using fixed centre distance. the program calculates automaticelly minimum number of teeth to avoid interference and profile correction is made to give this centre distance (250 mm).

CONCLUSION

It is possible to construct a software containing a complete design and make a detailed drawing of spur, helical and double helical 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 gear can be obtained according to the input power and speed, generally speaking, or entering some information to get a special design of the gear such as dimensions, materlal, 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 rasults show that:

1- Obtaining a remarkebly 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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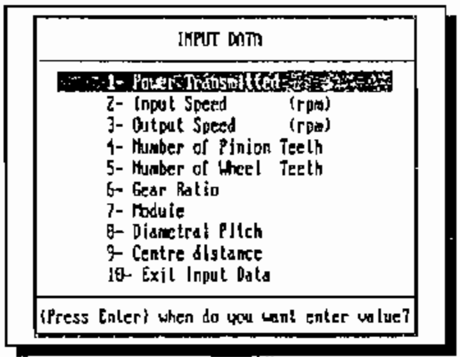

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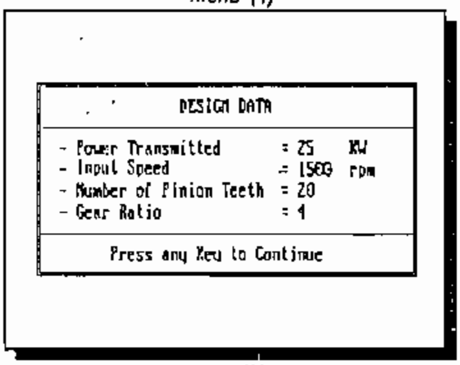

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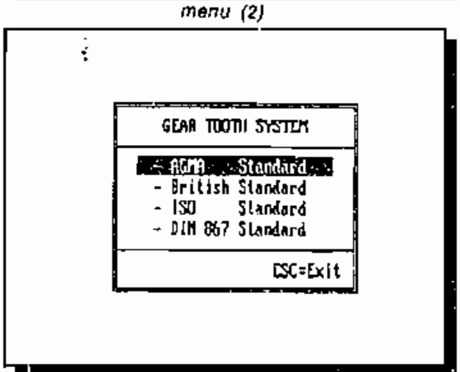

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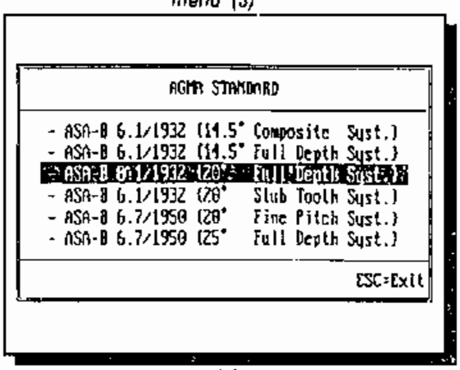

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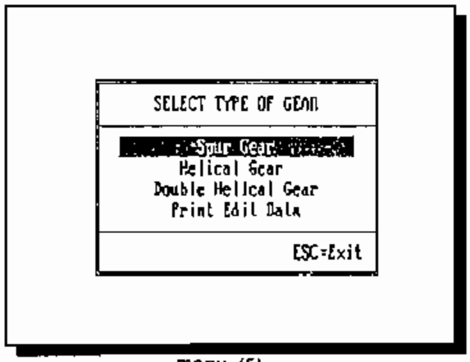

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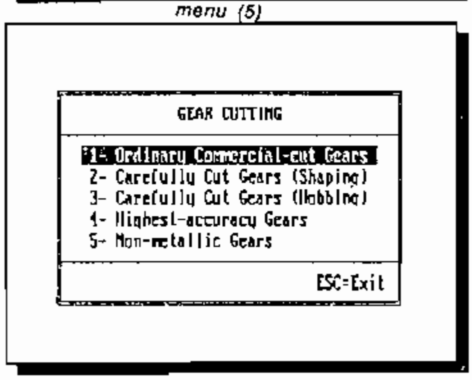

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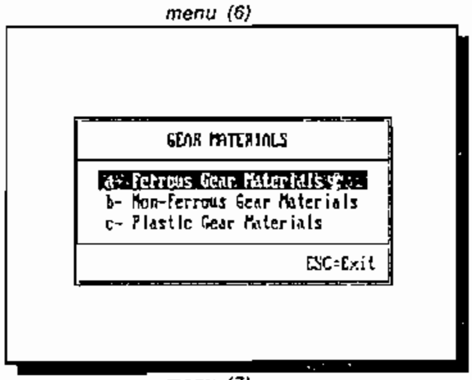

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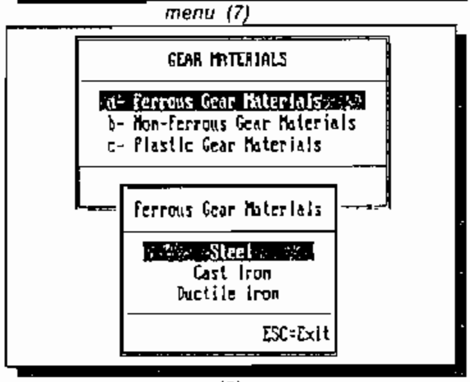

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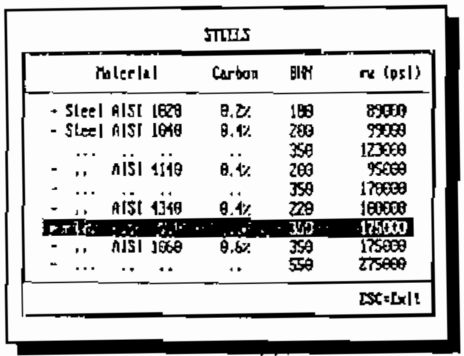
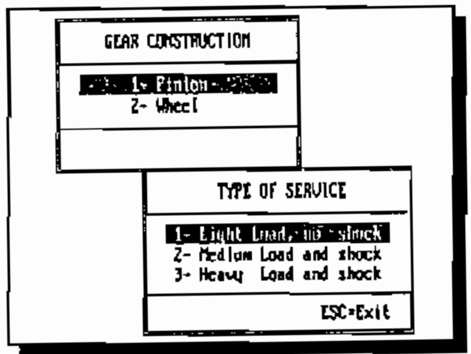


menu (9)



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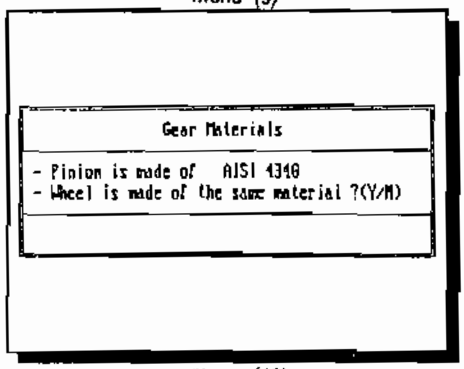
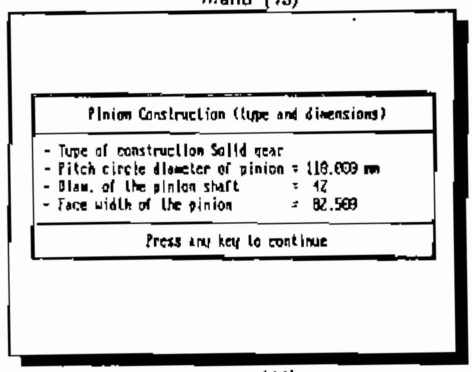


menu (10)



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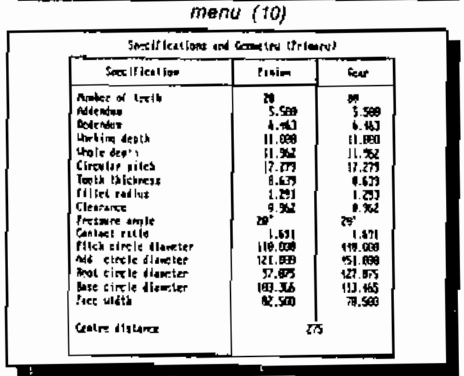

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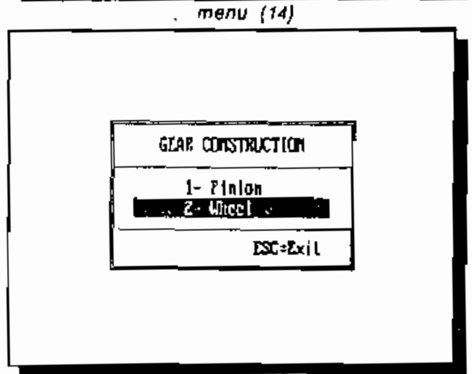

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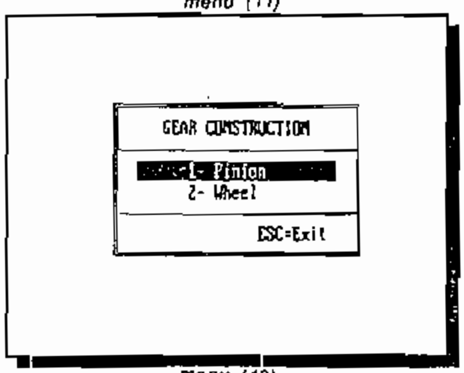

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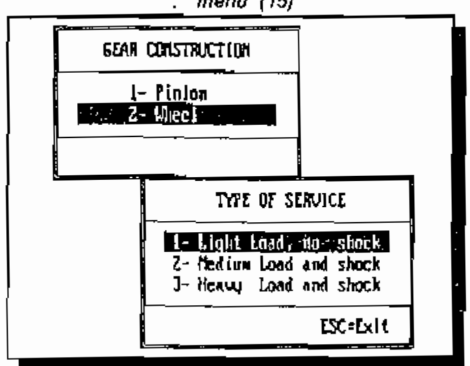

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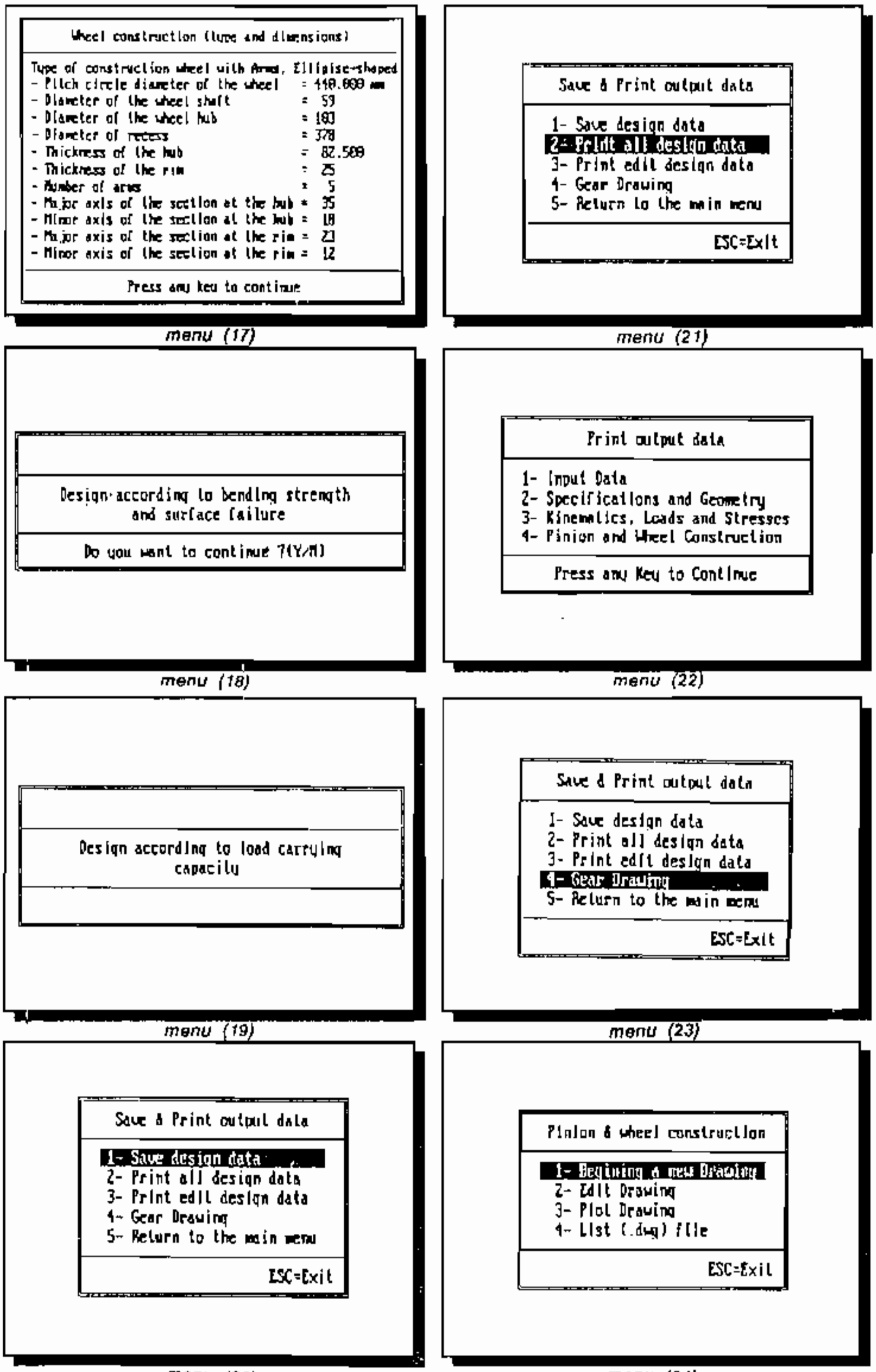

Fig.(4) Different menus of software for run of 25 Kw, 1500 rpm power and speed respectively.

Input Data	
- Power, Kw	= 25
- Input speed, rpm	= 1500
- Number of pinion teeth	= 20
- Gear ratio	= 4.000
- Design according to "AGMA"	

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

1- Specifications and Geometry

Specifications & Geometry	Pinion	Wheel
<< Spur Gear >>		
- Material	AISI 4340	AISI 4340
- Number of teeth	20	80
- Addendum, mm	5.500	5.500
- Dedendum, mm	6.463	6.463
- Working depth, mm	11.000	11.000
- Whole depth, mm	11.962	11.962
- Tooth thickness, mm	8.639	8.639
- Face width, mm	82.500	78.5
- Pitch circle diam., mm	110.000	440.000
- Add. circle diam., mm	121.000	451.000
- Root circle diam., mm	97.075	427.075
- Base circle diam., mm	103.366	413.465
- Profile correction, mm	-	-
- Module mm	5.5	
- Circular pitch, mm	17.279	
- Base Circular pitch, mm	16.237	
- Fillet radius, mm	1.293	
- Clearance, mm	0.962	
- Backlash, mm	.1975	
- Contact ratio	1.691	
- Gear ratio	4.000	
- Pressure angle	20°	
- Working pressure angle	20° 00'	
- Centre distance, mm	275.000	
- St. (ASA-B), 20° Full Depth Tooth System		

Table (3) specification and geometry of pinion and wheel for 25 Kw, 1500 rpm power and speed respectively.

2- Kinematics-Loads and Stresses

- Speed for pinion & wheel, rpm	= 1500 , 375
- Pitch line velocity, m/sec	= 8.639
- Tan., Rad. & Nor. load, N	= 2896, 1054, 3081
- Calc. module Acc. to L.E.	3.69 < 5.5 Safe
- Tang. load Acc. to L.E., N	= 15298
- Tang. load Acc. to M.L.E., N	= 33737
- Dynamic load Acc. to Backingham, N	= 16873
- Limiting load for wear, N	= 28765
- U.T.S., N/mm² (Ib/in²)	= 1207(175000)
- All. Tensile strength, N/mm² (Ib/in²)	= 402(58332)
- All. surface strength, N/mm² (Ib/in²)	= 322(46666)
- Calc. bending stress Acc. L.E., N/mm²	= 43
- Calc. .. .. Acc. M.L.E., N/mm²	= 166
- Calc. contact stress, N/mm²	= 292
- ** AGMA **	
- Tooth Bending stress, (S.V.) Ib/in²	= 11757
- .. .. .. (E.V.) Ib/in²	= 7838
- Tooth-Surface Durability, (S.V.) Ib/in²	= 32192
- .. .. .. (E.V.) Ib/in²	= 26285
- ** ISO **	
- Tooth Bending stress, (S.V.) N/mm²	= 81
- .. .. .. (E.V.) N/mm²	= 14
- Tooth-Surface Durability, (S.V.) N/mm²	= 222
- .. .. .. (E.V.) N/mm²	= 122
- ** British **	
- Tooth Bending stress, Ib/in²	= 2096
- Tooth Surface stress for wear, Ib/in²	= 2995
- Scoring-criterion number	= 365
- Flash Temperature, °F	= 104.021

Table (4) kinematics, loads and stresses of gearset for 25 Kw and 1500 rpm power and speed respectively.

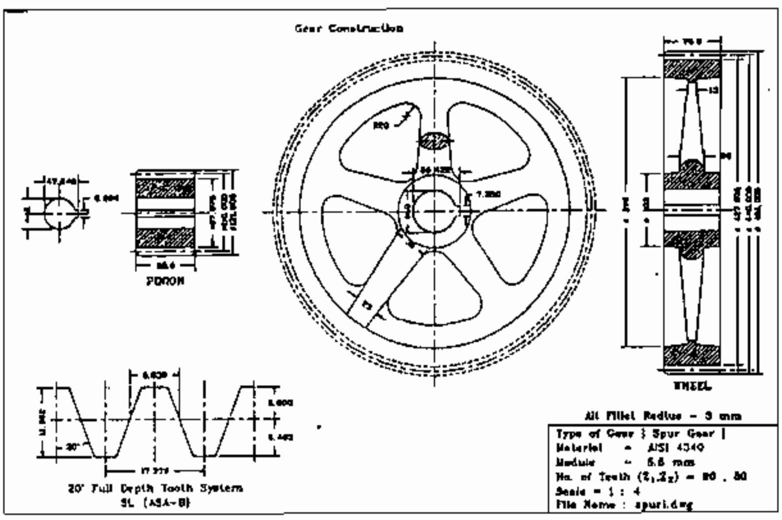

Fig.(5) Construction drawing of pinion and wheel for 25 kw, 1500 rpm power and speed respectively

Input Data	
- Power, Kw	= 50
- Input speed, rpm	= 2000
- Number of pinion teeth	= 20
- Gear ratio	= 4.000
- Design according to "AGMA"	

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

1- Specifications and Geometry		
Specifications & Geometry	Pinion	Wheel
<< Helical Gear >>		
- Material	AISI 4340	AISI 4340
- Number of teeth	20	80
- Virtual number of teeth	30	123
- Addendum mm	5.000	5.000
- Dedendum, mm	5.875	5.875
- Working depth, mm	10.000	10.000
- Whole depth, mm	10.875	10.875
- Normal tooth thick.. mm	7.854	7.854
- Trans. tooth thick.. mm	9.069	9.069
- Face width, mm	75	71
- Pitch circle diam.. mm	115.470	461.880
- Add. circle diam.. mm	125.470	471.880
- Root circle diam.. mm	103.720	450.130
- Base circle diam.. mm	106.451	425.803
- Profile correction. mm	-	-
- Nor. & Tr. module. mm	5, 5.774	
- Nor., Tr. & Ax. C.P..mm	15.708, 18.138, 31.416	
- Nor., Tr. & Ax. B.P..mm	14.761, 16.721, 29.521	
- Fillet radius, mm	1.175	
- Clearance. mm	0.875	
- Backlash. mm	1975	
- Contact ratio	3.766	
- Gear ratio	4.000	
- Pressure angle	20°	
- Trans. pressure angle	22° 47'	
- Working pressure angle	20° 00'	
- Helix angle	30°	
- Centre distance, mm	288.675	
- St. (ASA-B). 20° Full Depth Tooth System		

Table (5) specification and geometry of pinion and wheel for 50 Kw, 2000 rpm power and speed respectively.

2- Kinematics-Loads and Stresses	
- Speed for pinion & wheel, rpm	= 2000 , 500
- Pitch line velocity, m/sec	= 12.092
- Tan. & Rad. loads, N	= 4138 1739
- Ax. & Nor. loads, N	= 2389, 5084
- Calc. module Acc. to L.E.	3.23 < 5 Safe
- Tang. load Acc. to L.E.. N	= 11286
- Tang. load Acc. to M.L.E.. N	= 16962
- Dynamic load Acc. to Backingham., N	= 16446
- Limiting load for wear, N	= 36601
- U.T.S. , N/mm^2 (lb/in^2)	= 1207(175000)
- All. Tensile strength, N/mm^2 (lb/in^2)	= 402(58332)
- All. surface strength, N/mm^2 (lb/in^2)	= 322(46666)
- Calc. bending stress Acc. L.E.. N/mm^2	= 72
- Calc. .. .. Acc. M.L.E.. N/mm^2	= 362
- Calc. contact stress. N/mm^2	= 253
- ** AGMA **	
- Tooth Bending stress. (S.V.) lb/in^2	= 9938
- .. .. (E.V.) lb/in^2	= 9035
- Tooth-Surface Durability, (S.V.) lb/in^2	= 33734
- .. .. (E.V.) lb/in^2	= 32164
- ** ISO **	
- Tooth Bending stress. (S.V.) N/mm^2	= 69
- .. .. (E.V.) N/mm^2	= 28
- Tooth-Surface Durability. (S.V.) N/mm^2	= 233
- .. .. (E.V.) N/mm^2	= 149
- ** British **	
- Tooth Bending stress, lb/in^2	= 2019
- Tooth Surface stress for wear. lb/in^2	= 3630
- Scoring-criterion number	= 605
- Flash Temperature. °F	= 105.424

Table (6) kinematics, loads and stresses of gearset for 50 Kw and 2000 rpm power and speed respectively.

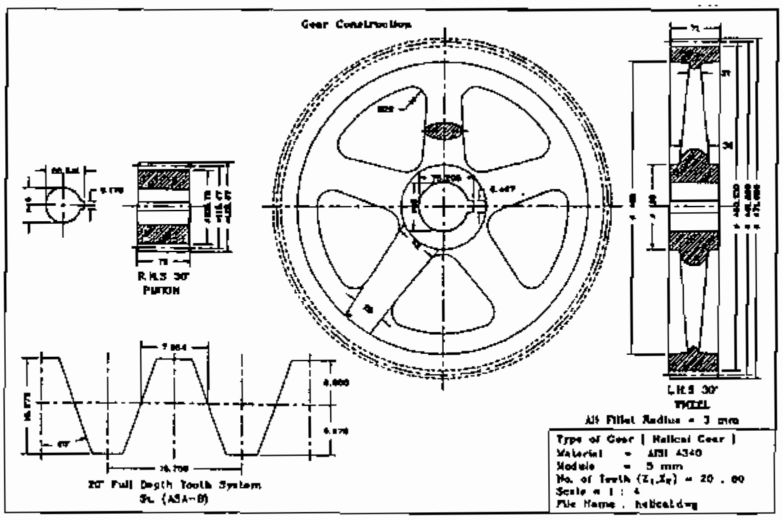

Fig.(6) Construction drawing of pinion and wheel for 50 kw, 2000 rpm power and speed respectively

Input Data	
- Power, Kw	= 100
- Input speed, rpm	= 3000
- Number of pinion teeth	= 20
- Gear ratio	= 4.000
- Design according to "AGMA"	

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

1- Specifications and Geometry		
Specifications & Geometry	**Pinion**	**Wheel**
<< Double Helical Gear >>		
- Material	AISI 4340	AISI 4340
- Number of teeth	20	80
- Virtual number of teeth	36	145
- Addendum, mm	5.000	5.000
- Dedendum, mm	5.875	5.875
- Working depth, mm	10.000	10.000
- Whole depth, mm	10.875	10.875
- Normal tooth thick., mm	7.854	7.854
- Trans. tooth thick., mm	9.588	9.588
- Face width, mm	2X 63	2X 59
- Pitch circle diam., mm	122.077	488.310
- Add. circle diam., mm	132.077	498.310
- Root circle diam., mm	110.327	476.560
- Base circle diam., mm	111.561	446.243
- Profile correction, mm	-	-
- Nor. & Tr. module, mm	5. 6.104	
- Nor., Tr. & Ax. C.P., mm	15.708, 19.176, 27.386	
- Nor., Tr. & Ax. B.P., mm	14.761, 17.524, 25.734	
- Fillet radius, mm	1.175	
- Clearence, , mm	0.875	
- Backlash, mm	.1975	
- Contact ratio	3.838	
- Gear ratio	4.000	
- Pressure angle	20°	
- Trans. pressure angle	23° 57'	
- Working pressure angle	20° 00'	
- Helix angle	35°	
- Centre distance, mm	305.194	
- St. (ASA-B), 20° Full Depth Tooth System		

Table (7) specification and geometry of pinion and wheel for 100 Kw, 3000 rpm power and speed respectively.

2- Kinematics-Loads and Stresses		
- Speed for pinion & wheel, rpm		= 3000 , 750
- Pitch line velocity, m/sec		= 19.176
- Tan., Rad. & Nor. load, N		= 5218, 2319, 6779
- Calc. module Acc. to L.E.		2.91 < 5 Safe
- Tang. load Acc. to L.E.,	N	= 13298
- Tang. load Acc. to M.L.E.,	N	= 38195
- Dynamic load Acc. to Backingham,	N	= 36897
- Limiting load for wear,	N	= 36042
- U.T.S. ,	N/mm² (lb/in²)	= 1207(175000)
- All. Tensile strength,	N/mm² (lb/in²)	= 402(58332)
- All. surface strength,	N/mm² (lb/in²)	= 322(46666)
- Calc. bending stress Acc. L.E.,	N/mm²	= 40
- Calc. ,, ,, Acc. M.L.E.,	N/mm²	= 396
- Calc. contact stress,	N/mm²	= 202
- ** AGMA **		
- Tooth Bending stress, (S.V.)	lb/in²	= 7520
- ,, ,, ,, (E.V.)	lb/in²	= 6836
- Tooth-Surface Durability, (S.V.)	lb/in²	= 28539
- ,, ,, ,, (E.V.)	lb/in²	= 27211
- ** ISO **		
- Tooth Bending stress, (S.V.)	N/mm²	= 52
- ,, ,, ,, (E.V.)	N/mm²	= 21
- Tooth-Surface Durability, (S.V.)	N/mm²	= 197
- ,, ,, ,, (E.V.)	N/mm²	= 126
- ** British **		
- Tooth Bending stress,	lb/in²	= 1033
- Tooth Surface stress for wear,	lb/in²	= 3649
- Scoring-criterion number		= 602
- Flash Temperature, °F		= 105 036

Table (8) kinematics, loads and stresses of gearset for 100 Kw and 3000 rpm power and speed respectively.

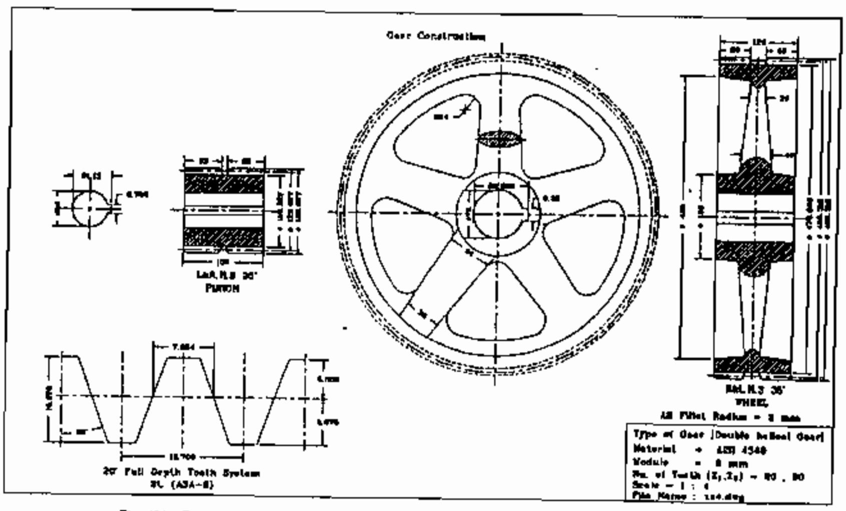

Fig.(7) Construction drawing of pinion and wheel for 100 kw, 3000 rpm power and speed respectively

Input Data	
- Power, Kw	= 25
- Input speed, rpm	= 1500
- Gear ratio	= 4.000
- Actual center distance, mm	= 250
- Design according to "AGMA"	

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

1- Specifications and Geometry

Specifications & Geometry	Pinion	Wheel
<< Spur Gear >>		
- Material	AISI 4340	AISI 4340
- Number of teeth	18	72
- Addendum, mm	7.580	5.832
- Dedendum, mm	4.295	6.042
- Working depth, mm	11.000	11.000
- Whole depth, mm	11.962	11.962
- Tooth thickness, mm	10.217	8.945
- Face width, mm	88	84
- Pitch circle diam., mm	99.000	396.000
- Add. circle diam., mm	114.159	407.665
- Root circle diam., mm	90.410	383.916
- Base circle diam., mm	93.030	372.118
- Profile correction, mm	+0.3941	+0.0764
- Module, mm	5.5	
- Circular pitch, mm	17.279	
- Base Circular pitch, mm	16.237	
- Fillet radius, mm	1.293	
- Clearance, mm	0.963	
- Backlash, mm	.1975	
- Contact ratio	1.951	
- Gear ratio	4.000	
- Pressure angle	20°	
- Working pressure angle	21° 31'	
- Centre distance, mm	247.500	
- Actual Centre dist., mm	250	
- Add. modification, mm	0.0881	
- St. (ASA-B)., 20° Full Depth Tooth System		

Table (g) specification and geometry of pinion and wheel for 25 Kw, 1500 rpm power and speed respectively.

2- Kinematics-Loads and Stresses

- Speed for pinion & wheel, rpm		= 1500 , 375
- Pitch line velocity, m/sec		= 7.775
- Tan., Rad. & Nor. load, N		= 3217, 1171, 3424
- Calc. module Acc. to L.E.		3.80 < 5.5 Safe
- Tang. load Acc. to L.E.,	N	= 16746
- Tang. load Acc. to M.L.E.,	N	= 33336
- Dynamic load Acc. to Backingham,	N	= 15206
- Limiting load for wear,	N	= 27615
- U.T.S.,	N/mm² (Ib/in²)	=1207(175000)
- All. Tensile strength,	N/mm² (Ib/in²)	= 402(58332)
- All. surface strength,	N/mm² (Ib/in²)	= 322(46666)
- Calc. bending stress Acc. L.E.,	N/mm²	= 48
- Calc. ,, ,, Acc. M.L.E.,	N/mm²	= 173
- Calc. contact stress,	N/mm²	= 314
- ** AGMA **		
- Tooth Bending stress, (S.V.)	Ib/in²	= 12247
- ,, ,, ,, (E.V.)	Ib/in²	= 8164
- Tooth-Surface Durability, (S.V.)	Ib/in²	= 34633
- ,, ,, ,, (E.V.)	Ib/in²	= 28278
- ** ISO **		
- Tooth Bending stress, (S.V.)	N/mm²	= 84
- ,, ,, ,, (E.V.)	N/mm²	= 17
- Tooth-Surface Durability, (S.V.)	N/mm²	= 239
- ,, ,, ,, (E.V.)	N/mm²	= 131
- ** British **		
- Tooth Bending stress,	Ib/in²	= 1965
- Tooth Surface stress for wear,	Ib/in²	= 2140
- Scoring-criterion number		= 376
- Flash Temperature, °F		= 105.695

Table (h) kinematics, loads and stresses of gearset for 25 Kw and 1500 rpm power and speed respectively.

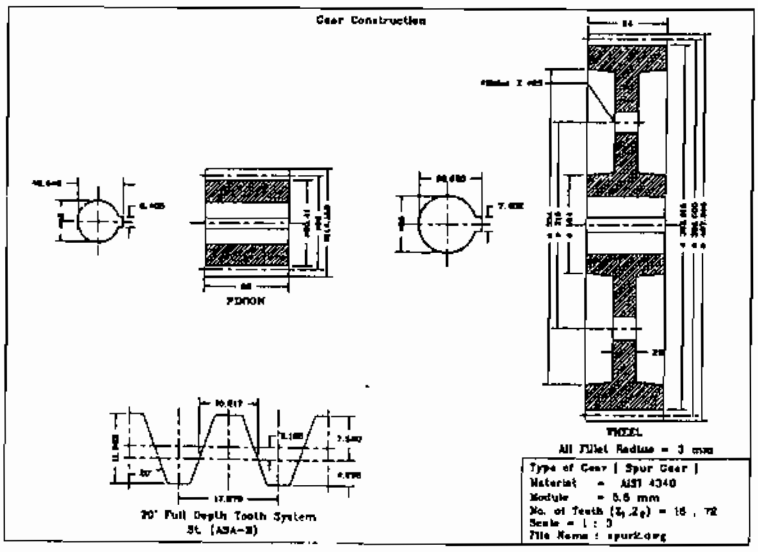

Fig.(8) Construction drawing of pinion and wheel for 25 kw, 1500 rpm power and speed respectively (profile correction)

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استخدام الحاسب الآلى فى تصميم المسننات
الجزء الأول
المسننات العدلة والحلزونية والحلزونية المزدوجة

الخلاصة : الغرض من هذا البحث هو انشاء برنامج جاهز (software) يحتوى على خطوات لتصميم ورسم المسننات العدلة والحلزونية والحلزونية المزدوجة . هذا البرنامج الجاهز يمكن استخدامه وتشغيله على أى حاسب آلى شخصى . العناصر الرئيسية للتصميم والتى يتكون منها هذا البرنامج هى : البيانات الدخله 'شكل ونوع السنه طبقا للمواصفات (الدولية – الامريكية – البريطانية – الألمانية)' الموديول' أقل عدد من الأسنان لتلاشى التداخل' تصحيح بروفيل السنة' المسافة بين المحورين ' نسبة التغطية' عرض المسنن' زاوية اللولبة الحلزونية ' نوع الخام المستخدم' معادلات التصميم طبقا لمقاومة الإنحناء للسنة (ليوس وليوس المعدلة وباكنجهام)' معادلات التصميم طبقا لانهيار سطح السنة ' معادلات سعة التحميل لمقاومة انحناء وانهيار سطح السنة للمسنن(النظام الدولى والجمعية الامريكية لإنتاج المسننات والنظام البريطانى مع اعتبار أن الحمل ثابت أو متغير) . تم اختيار واستنتاج مجموعة من المعادلات والعلاقات العملية لإنشاء ورسم المسننات (مخلق فى العمود ' مصمط' له جدار كامل مصمت' له جدار به ثقوب' له أعصاب' بجدارين' من أجزاء مربوطة مع بعضها) .

ان تشغيل هذا البرنامج الجاهز يعطى النتائج التالية : هندسية ومواصفات المسننات (القائد والمنقاد)' كينيماكية المسنن' الأحمال والإجهدات للسنه والمسنن والرسم الكامل للمسننات (القائد والمنقاد) وذلك طبقا للبيانات الداخلة . تم عمل أمثله مختلفه ومتنوعه لبيان مدى قدرة واستجابة هذا البرنامج الجاهز لإتمام عملية التصميم بالكامل والحصول بيانات ورسومات كامله صحيحة . ولقد أظهرت هذه النتائج الدقه الكبيره فى أبعاد ومواصفات المسنن وكذلك الوفر الكبير فى الوقت والتكاليف