Computer Applications in Mechanical Engineering Education-Case 2: Helical Gear Design and Analysis with VISUAL BASIC

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***Abstract*— A visual basic developed program, EHgear, for sizing, stress and strength analysis of the full depth teeth, standard helical gears is described. The Design basis is on the American Gear Manufacturer's Association (AGMA) recommended standard. Test and validation of the program through numerical examples with typical helical gear operational parameters is shown.**

***Keywords*— Gears, helical gears, gear design, gearing, engineering programs.**

I. Introduction

This article forms a second part (Case 2) of an Engineering Department's efforts at exposing mechanical engineering students to hands-on practices of applying computers through software development to solutions of real world engineering problems. The first article (Case 1) was a Diesel Engine performance analysis program developed based on MATLAB [1]. Menu driven software programs for helical gears design and load analysis have been reported in the literature [2]. While some programs are purely for gear parameters calculations, others have additional functionality of converting the calculated gear parameters into animated drawing for computer-aided-manufacturing (CAM) tooling [2], [3]. Here is another effort at developing a more user-friendly helical gear program using VISUAL BASIC.

II. Equations for Helical Gear Sizing

Equations for design and analysis of gears are available in the open literature [2-7]. Most Machine Design textbooks treat the subject of gears and gearing design to some degree. For stress and strength analysis, charts have been developed and provided for critical gear parameters determination by gear standards regulators such as the American Gear Manufacturers Associations (AGMA), Japanese Gear Manufacturers Association (JGMA) of the Japan Industrial Standards (JIS), British Standards (BS), and German Deutsches Institut für Normung (DIN). These charts are very useful for hand calculations, since the numbers of variables under consideration are often many and prone to possible errors if the equations for determining such parameters are directly applied by the gear designer. For computer applications, with a careful step-by-step systems analysis approach to programming, such errors are easily traced and debugged. As a measure of general standardisation to handle the deluge of different Standards, the International Standards Organisation (ISO) provide the ISO Standards for metric gearing which is generally accepted by all countries [4].

A. Determination of the Number of Teeth for Pinion and Gear

To avoid undercut, possible interference and weakness of a gear tooth in the addendum area of the root fillet area, standard gears are cut with teeth numbers below a critical value [4]. For a helical gear pair of a given gear ratio, m_G, the minimum number of teeth on the pinion to avoid interference is as given in eq. (1), by [5]. For the helical gear, the minimum number of teeth to avoid interference increases with decreasing helix angle for a fixed gear ratio and normal ISO recommended pressure angle of 20°, though marginally. For gear ratio, m_G = 1, 30° helix angle, N_{min} is approximately 9, for 25° helix angle, about 10, for 20° helix angle, the approximate value is 11, for a 15° helix angle, about 11, and for a 10° helix angle, about 12.

$$N_{min} = \frac{2k\cos\psi}{(1+2m_G)\sin^2\emptyset_t}\left[m_G + \sqrt{{m_G}^2 + (1+2m_G)\sin^2\emptyset_t}\right] \quad (1)$$

$k = 1$ (For full-depth teeth)

$N_p = N_{min}$

$N_G = m_G N_P$

N_p = number of teeth on pinion

N_G = the number of teeth on gear

$\emptyset_t$=transverse pressure angle

m_G =gear ratio

B. Virtual Number of Teeth for Pinion and Gear N'_P, N'_G

Used in determining helical gear tooth strength, the geometric basis of the virtual number of teeth is on an equivalent spur gear in the normal plane of larger number of teeth depending on the helix angle [2].

$$N'_P = \frac{N_P}{\cos^3\psi}, \qquad N'_G = \frac{N_G}{\cos^3\psi} \quad (2)$$

ψ =helix angle

C. Pitch Diameter for Pinion and Gear d_P, d_G

$$d_P = \frac{N_P m_n}{\cos\psi} = N_P m_t, \qquad d_G = \frac{N_G m_n}{\cos\psi} = N_G m_t \tag{3}$$

m_n = Normal module, used in metric gearing, it is defined as the length of pitch diameter per tooth [4]. The unit of module is mm.

m_t=transverse module

D. Outside Diameter for Pinion and Gear d_{oP}, d_{oG}

$$d_{oP} = d_P + 2a_w \tag{4}$$

$$d_{oG} = d_G + 2a_w \tag{4a}$$

a_w=addendum

E. Base Circle Diameter for Pinion and Gear d_{bP}, d_{bG}

$$d_{bP} = d_P \cos\emptyset_t\,, \qquad d_{bG} = d_G \cos\emptyset_t \tag{5}$$

F. Root Diameter for Pinion and Gear $d_{rP},\ d_{rG}$

$$d_{rP} = d_P - 2b_w, \qquad d_{rG} = d_G - 2b_w \tag{6}$$

$b_w = Dedendum$

G. Transverse Module m_t

$$m_t = \frac{m_n}{\cos\psi} \tag{7}$$

H. Addendum a_w

$$a_w = m_n \tag{8}$$

I. Dedendum b_w

$$b_w = 1.25 m_n \tag{9}$$

J. Transverse Pressure Angle $\emptyset_t$

$$\emptyset_t = \tan^{-1}\left(\frac{\tan\emptyset_n}{\cos\psi}\right) \tag{10}$$

K. Transverse Pitch p_t

$$p_t = \pi m_t \tag{11}$$

L. Axial Pitch p_x

$$p_x = \frac{p_t}{\tan\psi} \tag{12}$$

M. Chordal Tooth Thickness $\overline{S_P}, \overline{S_G}$

$$\overline{S_P} = N_P' m \sin\psi_{vP}\,, \qquad \overline{S_G} = N_G' m \sin\psi_{vG} \tag{13}$$

$$\psi_{vP} = \frac{90}{N_P'}, \qquad \psi_{vG} = \frac{90}{N_G'} \tag{13a}$$

N. Normal Circular Pitch p_n

$$p_n = \pi m_n \tag{14}$$

O. Centre Distance C

$$C = \frac{d_G + d_P}{2} \tag{15}$$

P. Base Helix Angle ψ_b

$$\psi_b = \tan^{-1}(\tan\psi \sin\emptyset_t) \tag{16}$$

Q. Gear Ratio m_G

$$m_G = \frac{N_G}{N_P} \tag{17}$$

R. Face Width b

For safe operation, [5] provides a range of values for selecting the face width as:

$$3\pi m_n \le b \le 5\pi m_n \tag{18}$$

This compares with the recommendation of [8] for a minimum face width in line with the eq. (18a):

$$b_{\min} = 1.15 p_t \cos\psi \tag{18a}$$

Shigley [5], suggests, a median face width as defined by eq,. (18b) be initially selected for the approximate gear sizing calculation. This is applied in program discussed in this article.

$$b = 4\pi m_n \tag{18b}$$

The median face width value of eq. (18b), may later be adjusted in the Analysis Program during the iteration process. When the face width falls out of range, the program displays an error check out of range dialogue message box.

S. Face Contact Ratio m_F

$$m_F = \frac{b}{p_x} \tag{19}$$

T. Tooth Fillet Radius r_f

$$r_f = 0.38m_n \tag{20}$$

U. Whole Depth h_t

$$h_t = 2.25m_n \tag{21}$$

V. Rim Thickness below the Tooth t_R

$$t_R = 1.2h_t \tag{22}$$

III. EQUATIONS FOR HELICAL GEAR FORCE ANALYSIS

The forces acting upon the gear teeth, on gear shafts and bearings are an important consideration. The equations relating them are given in eq. (23) to eq. (26):

A. Pitch Line Velocity

$$V = \frac{\pi d_p n}{60000} \tag{23}$$

B. Tangential Force

$$F_t = \frac{1000P}{V} \tag{24}$$

C. Axial Force

$$F_a = F_t \tan\psi \tag{25}$$

D. Radial Force

$$F_r = F_t \tan\phi_t \tag{26}$$

IV. EQUATIONS FOR HELICAL GEAR STRESS AND STRENGTH ANALYSIS

The model for stress and strength analysis is based on the American Gear Manufacturers Association (AGMA) standard [9], [5]. This compares to the ISO and other analysis methods with slight variations [4]. Strength considerations in gears are based on tooth bending fatigue failure, and Hertz (contact) or surface fatigue failure as a result of wear due to the several repetitive rolling actions of the meshing gears and the induced stresses [5], [4]. Surface durability is concerned with avoiding pitting or wear failure.

Recommended equations are:

A. AGMA Bending Stress Equation for Pinion and Gear

Estimating the bending stress in the pinion and gear is obtained by the relations of eq. (27)

$$\sigma_P = F_t K_o \acute{K}_v K_{sP} \frac{1}{b m_t} \frac{K_H K_B}{Y_{JP}}, \qquad \sigma_G = F_t K_o \acute{K}_v K_{sG} \frac{1}{b m_t} \frac{K_H K_B}{Y_{JG}} \tag{27}$$

B. AGMA Pitting Resistance Equation (Hertzian-Contact Stresses)

Hertz contact stresses can be estimated in the AGMA method by the relations of eq. (28) for the pinion and gear:

$$\sigma_{CP} = Z_E \sqrt{F_t K_o \acute{K}_v K_{sP} \frac{K_H}{d_p b} \frac{Z_R}{Z_I}}, \qquad \sigma_{CG} = Z_E \sqrt{F_t K_o \acute{K}_v K_{sG} \frac{K_H}{d_p b} \frac{Z_R}{Z_I}} \tag{28}$$

C. AGMA Bending Endurance Strength Equation for Pinion and Gear

$$\sigma_{allP} = \frac{\sigma_{FPP}}{FOS} \frac{Y_{NP}}{Y_\theta Y_Z}, \qquad \sigma_{allG} = \frac{\sigma_{FPG}}{FOS} \frac{Y_{NG}}{Y_\theta Y_Z} \tag{29}$$

The condition for avoiding bending fatigue failure is that the significant tooth bending stress, eq. (27), be very much less than either the gear material yield strength or the allowable bending endurance strength of eq. (29) [5].

D. AGMA Contact Endurance Strength equation for Pinion and Gear

$$\sigma_{CallP} = \frac{\sigma_{HPP}}{FOS} \frac{Z_{NP}}{Y_\theta Y_Z}, \qquad \sigma_{CallG} = \frac{\sigma_{HPG}}{FOS} \frac{Z_{NG} Z_W}{Y_\theta Y_Z} \tag{30}$$

The condition for avoiding contact or surface fatigue failure is that the significant contact stress, of eq. (28), be very much less than either the gear material allowable contact or surface endurance strength of eq. (30) [5]

V. Parameters Equations for the AGMA Helical Gear Stress and Strength Determination

A. Overload Factor, K_o

TABLE I
ESTIMATING AGMA OVERLOAD FACTORS. *SOURCE: [5], [6].*

Power Source	Uniform	Moderate Shock	Heavy Shock
uniform	1.00	1.25	1.75
Light shock	1.25	1.50	2.00
Medium shock	1.50	1.75	2.25

B. Dynamic Factor, $\acute{K}_v$

$$\acute{K}_v = \left(\frac{A+\sqrt{200V}}{A}\right)^B \quad (31)$$

$$A = 50 + 56(1-B) \quad (31a)$$

$$B = 0.25(12 - Q_V)^{\frac{2}{3}} \quad (31b)$$

Where, V, is the pitch line velocity in [m/s].

A model for specifying quality number Q_V is shown in Table II as follows:

TABLE II
DETERMINING QUALITY NUMBERS. *SOURCE: AGMA*

PITCH LINE VELOCITY [M/S]	QUALITY NUMBER
10 – 19	3 – 6
19-28	6 – 8
28 – 41	8 – 10
41 – 66	10 – 12

C. Size Factor for Pinion and Gear. K_{sP}, K_{sG}

$$K_{sP} = 0.904\left(bm_n\sqrt{Y_P}\right)^{0.0535}, \qquad K_{sG} = 0.904\left(bm_n\sqrt{Y_G}\right)^{0.0535} \quad (32)$$

A modified relation for the form factors, Y_p and Y_G of the pinion and gear respectively is given by [10] as in eq. (32a):

$$Y_p = \pi\left(0.154 - \frac{1.35}{(N_p+6)}\right), \qquad Y_G = \pi\left(0.154 - \frac{1.35}{(N_G+6)}\right) \quad (32a)$$

This differs slightly from other applications where the virtual number of teeth is applied.

D. Load Distribution Factor K_H

$$K_H = 1 + C_{mc}\left(C_{pf}C_{pm} + C_{ma}C_e\right) \quad (33)$$

$$C_{mc} = 1, \qquad C_{pm} = 1, \qquad C_e = 1$$

TABLE III
DETERMINING CONSTANTS IN FACTOR CMA. *SOURCE: [5]*

CONDITION	A	B	C
Open gearing	0.247	$6.5748(10^{-4})$	$-1.1858(10^{-7})$
Commercial, enclosed units	0.127	$6.2205(10^{-4})$	$-0.1442(10^{-7})$
Precision, enclosed units	0.0675	$5.0394(10^{-4})$	$-1.4353(10^{-7})$
Extraprecision, enclosed gear units	0.00360	$4.0157(10^{-4})$	$-1.2741(10^{-7})$

$$C_{ma} = A + Bb + Cb^2 \quad (33a)$$

$$C_{pf} = \begin{cases} \frac{b}{10d_p} - 0.025 & b \le 25 \ mm \\ \frac{b}{10d_p} - 0.0375 + 4.92(10^{-4})b & 25 < b \le 425\,mm \\ \frac{b}{10d_p} - 0.1109 + 8.15(10^{-4})b - 3.53(10^{-7})b^2 & 425 < b \le 1000\,mm \end{cases} \quad (33b)$$

Where b is face width and d_p is the pinion pitch circle diameter

E. Rim Thickness Factor k_B

$$m_b = \frac{t_R}{h_t} \quad (34)$$

$$K_B = \begin{cases} 1.6\ln\frac{2.242}{m_b} & m_b < 1.2 \\ 1 & m_b \le 1.2 \end{cases} \quad (34a)$$

$$h_t = 2.25m_n \quad (35)$$

$$t_R = 1.2h_t \quad (35a)$$

F. Surface Condition Factor Z_R

$$Z_R = 1 \tag{36}$$

G. Hardness Ratio Factor Z_W

$$Z_W = 1.0 + \acute{A}(m_G - 1.0) \tag{37}$$

$$\acute{A} = \begin{cases} 8.98(10^{-3})\left(\frac{H_{BP}}{H_{BG}}\right) - 8.29(10^{-3}) & 1.2 \leq \frac{H_{BP}}{H_{BG}} \leq 1.7 \\ 0 & \frac{H_{BP}}{H_{BG}} < 1.2 \\ 0.00698 & \frac{H_{BP}}{H_{BG}} > 1.7 \end{cases} \tag{38}$$

Where, H_{BP}, H_{BG} are Brinell hardness values for pinion and gear materials respectively

H. Stress Cycle Factors in Bending for Pinion and Gear Y_{NP}, Y_{NG}

$$Y_{NP} = 1.3558(N_c)^{-0.0178}, \quad Y_{NG} = 1.3558\left(\frac{N_c}{m_G}\right)^{-0.0178} \tag{39}$$

I. Stress cycle Factors in Pitting for Pinion and Gear Z_{NP}, Z_{NG}

$$Z_{NP} = 1.4488(N_c)^{-0.023}, \quad Z_{NG} = 1.4488\left(\frac{N_c}{m_G}\right)^{-0.023}$$

N_c is the number of life load cycles repeated applied. (40)

J. Reliability Factor, Y_Z

$$Y_z = \begin{cases} 0.658 - 0.0759\ln(1-R) & 0.5 < R < 0.99 \\ 0.50 - 0.109\ln(1-R) & 0.99 \leq R \leq 0.9999 \end{cases}$$

R is the reliability constant. (41)

K. Temperature Factor

$$Y_\theta = 1 \tag{42}$$

L. Surface Strength Geometric Factor

$$Z_I \begin{cases} \frac{\cos\phi_t \sin\phi_t}{2m_N} \frac{m_G}{m_G+1} & \text{external gears} \\ \frac{\cos\phi_t \sin\phi_t}{2m_N} \frac{m_G}{m_G-1} & \text{internal gears} \end{cases} \tag{43}$$

$$m_N = \frac{p_N}{0.95Z} \tag{43a}$$

$$p_N = p_n \cos\phi_n, \quad p_n = \pi m_n \tag{43b}$$

$$Z = \sqrt{(r_p + a_w)^2 - r_{bp}^2} + \sqrt{(r_G + a_w)^2 - r_{bG}^2} - (r_p + r_G)\sin\phi_t \tag{43c}$$

$$r_{bP} = (r_p)\cos\phi_t, \quad r_{bG} = (r_G)\cos\phi_t \tag{43d}$$

M. Bending Strength Geometric Factor for Pinion and Gear Y_{JP}, Y_{JG}

$$Y_{JP} = \frac{Y'_P . C_\psi}{K_{fp} . m_N} \tag{44}$$

$$Y_{JG} = \frac{Y'_G . C_\psi}{K_{fG} . m_N} \tag{44a}$$

Where,

K_{fP}, K_{fG} = fatigue stress-concentration factor for pinion and gear respectively

Y'_P and Y'_G = modified form factors for pinion and gear respectively as given by eq. (47).

The program applies the formulae in [9], which is based on calculations for the bending strength geometry factor on a unit normal module, i.e. m_n=1.

N. Fatigue Strength Concentration Factor for Pinion and Gear K_{fp}, K_{fG}

$$K_{fp} = H + \left(\frac{t_p}{r}\right)^L \left(\frac{t_p}{l}\right)^M \quad (45a)$$

$$K_{fG} = H + \left(\frac{t_G}{r}\right)^L \left(\frac{t_G}{l}\right)^M \quad (45b)$$

Where

$$H = 0.34 - 0.4583662\phi_n \quad (45c)$$

$$L = 0.316 - 0.4583662\phi_n \quad (45d)$$

$$M = 0.290 + 0.4583662\phi_n \quad (45e)$$

The root radius as given in eq. (45a) and eq. (45b) is based on a tooth profile with dimensions as shown in the fig. (1), as given by [5]. The radius of curvature, *r* and the whole depth, *l*, have been taken as.

$$r = 0.4276m_n,\ l = 2.355m_n \quad (45f)$$

This relates with the chart model selected by [5] for computation of the helical gear geometry factor. The whole depth is also slightly larger than that given by eq. (21). AGMA [9] observes that actual generated whole depth is larger due to tooth thinning for backlash.

The radius of curvature, *r*, differs from the, *r*, relation of eq. (45g), also given by [5]. Use of eq. (45g) gave very much lower values for the radius of curvature. It is not clear why such magnitude of difference.

$$r = \frac{(b_w - r_f)^2}{(d/2) + b_w - r_f} \quad (45g)$$

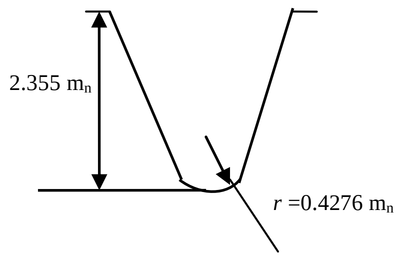

Fig. 1 Helical Gear tooth profile geometry adopted for computation of helical gear geometry factors. *Source: Shigley, [5]*

$$t_p = \sqrt{4lx_p},\qquad t_G = \sqrt{4lx_G} \quad (45h)$$

$$x_p = \frac{3Y_p m_n}{2},\qquad x_G = \frac{3Y_G m_n}{2} \quad (45i)$$

O. Helix Overlap Factor, C_ψ

$$C_\psi = 1 \quad (46)$$

P. Modified Form Factor for Pinion and Gear, Y_P', Y_G',

$$Y_P' = \frac{K_\psi}{\frac{\cos\phi_{nL}}{\cos\phi_{nr}}\left(\frac{6l}{t_p^2.ch} - \frac{\tan\phi_{nL}}{t_p}\right)} \quad (47)$$

$$Y_G' = \frac{K_\psi}{\frac{\cos\phi_{nL}}{\cos\phi_{nr}}\left(\frac{6l}{t_G^2.ch} - \frac{\tan\phi_{nL}}{t_G}\right)} \quad (47a)$$

Q. Helix Angle Factor, K_ψ

$$K_\psi = \cos\psi_r \cos\psi \quad (48)$$

ψ_r is the operating helix angle

ψ is the helix angle

$$\psi_r = \tan^{-1}\left(\frac{\tan\psi_b}{\cos\phi_{tr}}\right) \quad (48a)$$

ψ_b is the base helix angle

$$\psi_b = \tan^{-1}(\tan\psi \cos\phi_t) \quad (48b)$$

ϕ_{tr} is the operating transverse pitch

$$\phi_{tr} = \cos^{-1}\left(\frac{r_{bG} + r_{bp}}{C_r}\right) \quad (48c)$$

$$C_r = \frac{C}{m_n} \quad (48d)$$

C_r = Operating centre distance

R. Operating Normal Pressure Angle, Φ_{nr}

$$\phi_{nr} = \sin^{-1}(\cos\psi_b \sin\phi_{tr}) \quad (49)$$

S. Helical Factor, ch

$$ch = \frac{1}{1 - \sqrt{\frac{w}{100}\left(1 - \frac{w}{100}\right)}} \quad (50)$$

$$w = \tan^{-1}(\tan\psi \sin \emptyset_n) \quad (50a)$$

W = the angle of inclination of helical contact line

T. Load Angle ϕ_{nL}

$$\phi_{nL} = \tan\phi_{nw} - Inv\phi_{np} \quad (51)$$

ϕ_{nw} =pressure angle at load application point

$$\phi_{nw} = \tan^{-1}\left(\sqrt{\left(\frac{r'_{ap}}{r'_{bP}}\right)^2 - 1}\right) \quad (51a)$$

r'_{bP} =pinion virtual base radius

$$r'_{bP} = r'_P \cos\phi_n, \quad r'_P = \frac{N'_P}{2}, \quad N'_P = \frac{N_P}{\cos^3\psi} \quad (51b)$$

$$r'_{bP} = r'_P \cos\phi_n$$

$$r'_P = \frac{N'_P}{2}$$

$$N'_P = \frac{N_P}{\cos^3\psi}$$

N'_P = pinion virtual teeth

$$r'_{ap} = r'_P + r_{op} - r_p \quad (51c)$$

$$r_{op} = r_p + a_w \quad (51d)$$

$$Inv\phi_{np} = \tan\phi_n - \phi_n + \frac{S_n}{N'_P} \quad (51e)$$

$$S_n = \left(\frac{\pi}{2} + 2x_n \tan\phi_n\right) m_n \quad (51f)$$

$x_n = 0$ (For non-profile shifting gears)

A backlash allowance is allowed, calculated with a 0.024 mm reduction in normal circular tooth thickness of pinion and gear to provide 0.048mm total for one normal module [9], [5].

S_n =normal circular tooth thickness

U. Allowable Bending Stress for Through-Hardened Steel σ_{FPp}, σ_{FPG}

TABLE IV
BRINELL HARDNESS OF MATERIAL EQUATION FOR BENDING STRESS DETERMININATION. *SOURCE: SHIGLEY [5].*

	Grade 1	Grade 2
σ_{FP_p} (MPa)	$0.533H_{BP} + 88.33$	$0.703H_{BP} + 113$
σ_{FP_G} (MPa)	$0.533H_{BG} + 88.33$	$0.703H_{BG} + 113$

V. Contact Fatigue Stress for Hardened Steel, σ_{HPp}, σ_{HPG}

TABLE V
BRINELL HARDNESS OF MATERIAL EQUATION FOR CONTACT FATIGUE STRESS DETERMININATION. *SOURCE: SHIGLEY [5].*

	Grade 1	Grade 2
σ_{HP_p} (MPa)	$2.22H_{BP} + 200$	$2.41H_{BP} + 237$
σ_{HP_G} (MPa)	$2.22H_{BG} + 200$	$2.41H_{BG} + 237$

W. Elastic Coefficient

$$Z_E = \sqrt{\frac{1}{\pi\left(\frac{1-\mu_P{}^2}{E_P} + \frac{1-\mu_G{}^2}{E_G}\right)}} \quad (52)$$

E_P, E_G = modulus of elasticity for pinion and gear

Fig. 2 EHgear Helical gear sizing and analysis welcome

Fig. 3 A screen shot of helical gear sizing interactive user-interface

Fig. 4 A screen shot of helical gear analysis interactive user-interface

VI. Helical Gear Sizing Numerical Example 1

A pair of helical gears is to be designed for the following requirements: Gear ratio = 2.5; Helix angle = 30 degrees; Normal module, m_n = 2.5 mm

Fig. 5 Helical gear sizing solution for numerical example 1

VII. Helical Gear Stress and Strength Analysis Numerical Example 2

Conduct a failure analysis for the helical gear pair with the following meshing data: Normal module, m_n = 2.5; Normal pressure angle, $\Phi_n = 20^0$; Helix angle = 30^0, N_P = 17; N_G = 52; Pinion speed, N = 1800 rpm; Face width, b = 38 mm; Quality Number, Q_v = 6; transmitting power, P = 3 kW Brinell hardness value on pinion, HB_p = 240; Brinell hardness value on gear, HB_G = 200; both materials are made of through -hardened steel; Pinion life = 10^8 cycles; Reliability = 0.9. Factor of safety = 2.

(Data for the numerical example 2 was taken from Shigley [5], Example 14-5, pages 928-931, as a program validation check).

Fig. 6 Helical gear analysis solution for numerical example 2

Fig. 7 Accepted Helical gear after iterative analysis

VIII. CONCLUSIONS

The final accepted gear parameters from the analysis are shown on a separate screen as displayed in fig. (7). The EHgear program described is still undergoing extension and tests for robustness. The program considered only external standard gears of full depth teeth, through-hardened steel only, though allows for users to specify user-defined materials. Future extension will allow for internal gears to be handled, stub teeth, and a database of several material types. The handling of plastic gears will also be considered. Because of space limitations, only the subroutine codes of the helical sizing calculations, form event classes, and types of error checks provided in the program are shown in the appendix.

IX. APPENDIX: VISUAL BASIC SUBROUTINES AND CODES

```
Public Class sing

    Private Sub Button2_Click(ByVal sender As System.Object, ByVal e As System.EventArgs) Handles Button2.Click
        Close()
    End Sub

    Private Sub Button1_Click(ByVal sender As System.Object, ByVal e As System.EventArgs) Handles ok.Click

        If design.Checked = True Then
            Form2.ShowDialog()
        Else
            Form3.ShowDialog()
        End If

        If analysis.Checked = True Then

            Form3.anspvp.Text = ""
            Form3.anstlp.Text = ""
            Form3.ansrl.Text = ""
            Form3.ansal.Text = ""
            Form3.anscsp.Text = ""
            Form3.anscsg.Text = ""
            Form3.anscesp.Text = ""
            Form3.anscesg.Text = ""
            Form3.ansbsg.Text = ""
            Form3.ansbsp.Text = ""
            Form3.ansallbg.Text = ""
            Form3.ansallbp.Text = ""
            Form3.wearremarkg.Text = ""
            Form3.wearremarkp.Text = ""
            Form3.strengthrmkg.Text = ""
            Form3.strengthrmkp.Text = ""
            Form3.txtpwr.Clear()
            Form3.nmodule.Clear()
            Form3.facew.Clear()
            Form3.txtr1.Text = ""
            Form3.txtr2.Text = ""
            Form3.txtsp.Clear()
            Form3.txtng.Clear()
            Form3.txtnp.Clear()
            Form3.safetyfactor.Clear()
            Form3.allowbg.Clear()
            Form3.allowbp.Clear()
            Form3.allowcg.Clear()
            Form3.allowcp.Clear()
            Form3.selectbhg.Clear()
            Form3.selectbhp.Clear()
            Form3.brinnelg.Clear()
            Form3.brinnelp.Clear()

        End If

        If design.Checked = True Then

            Form2.gearratio.Clear()
            Form2.tm.Clear()
            Form2.centerd.Clear()
            Form2.hel.Clear()
            Form2.m2.Checked = True

            Form2.ansnp.Text = ""
            Form2.ansng.Text = ""
            Form2.ansdp.Text = ""
            Form2.ansdg.Text = ""
            Form2.ansmt.Text = ""
            Form2.ansaw.Text = ""
            Form2.ansdd.Text = ""
            Form2.ansqt.Text = ""
            Form2.anspt.Text = ""
            Form2.anspx.Text = ""
            Form2.ansb.Text = ""
            Form2.ansmn.Text = ""
            Form2.anschordal.Text = ""
            Form2.anspn.Text = ""
            Form2.ansbcp.Text = ""
            Form2.ansbcg.Text = ""
            Form2.ansbha.Text = ""
            Form2.ansgr.Text = ""
            Form2.ansoutp.Text = ""
            Form2.ansoutg.Text = ""
            Form2.ansmf.Text = ""
            Form2.anschrdg.Text = ""
```

```
            Form2.ansnvg.Text = ""
            Form2.ansnvp.Text = ""
            Form2.txttfr.Text = ""
            Form2.ansrtp.Text = ""
            Form2.ansrtg.Text = ""
            Form2.answholeht.Text = ""
            Form2.ansrimtr.Text = ""

        End If

    End Sub
End Class

Public Class Form2

    Private Sub GroupBox1_Enter(ByVal sender As System.Object, ByVal
e As System.EventArgs) Handles GroupBox1.Enter

    End Sub

    Private Sub m1_CheckedChanged(ByVal sender As System.Object,
ByVal e As System.EventArgs) Handles m1.CheckedChanged
        tm.Visible = False
        centerdist.Visible = True
    End Sub

    Private Sub m2_CheckedChanged(ByVal sender As System.Object,
ByVal e As System.EventArgs) Handles m2.CheckedChanged
        tm.Visible = True
        centerdist.Visible = False
    End Sub

    Private Sub Button1_Click(ByVal sender As System.Object, ByVal e
As System.EventArgs) Handles Button1.Click
        Close()
    End Sub

    Private Sub Button4_Click(ByVal sender As System.Object, ByVal e
As System.EventArgs) Handles transfer.Click

        Form3.hlx.Text = hel.Text
        Form3.nmodule.Text = tm.Text
        Form3.facew.Text = ansb.Text

        Form3.txtnp.Text = ansnp.Text
        Form3.txtng.Text = ansng.Text

        Dim f1, f2 As Double
        f1 = (ansb.Text / 4) * 3
        f2 = (ansb.Text / 4) * 5
        Form3.txtr1.Text = Math.Round(f1, 2)
        Form3.txtr2.Text = Math.Round(f2, 2)

        Form3.kop.SelectedIndex = 0
        Form3.kog.SelectedIndex = 0
        Form3.quality.SelectedIndex = 3

        Form3.ShowDialog()

    End Sub

    Private Sub calc_Click(ByVal sender As System.Object, ByVal e As
System.EventArgs) Handles calc.Click
        If gearratio.Text = Nothing Or hel.Text = Nothing Then
            MessageBox.Show("One or more input field(s) empty!!")
            Exit Sub
        End If

        If Not IsNumeric(gearratio.Text) Or Not IsNumeric(hel.Text) Then
            MessageBox.Show("text not allowed, Values only!!")
            Exit Sub
        End If

        If Val(gearratio.Text) < 1 Then
            MessageBox.Show("Gear ratio is out of range")
            Exit Sub
        End If


        If m2.Checked = True Then
            If tm.Text = Nothing Then
                MessageBox.Show("Please specify value of Module!!")
                Exit Sub
            End If

            If tm.Text < 0 Then
                MessageBox.Show("module value must be positive")
                Exit Sub
            End If

            If Not IsNumeric(tm.Text) Then
                MessageBox.Show("text not allowed, Values only!!")
                Exit Sub
            End If
        End If

        If m1.Checked = True Then
            If centerd.Text = Nothing Then
                MessageBox.Show("Please specify value for Center Distance!!")
                Exit Sub
            End If

            If centerd.Text < 0 Then
                MessageBox.Show("value of center distance must be positive")
                Exit Sub
            End If

            If Not IsNumeric(centerd.Text) Then
                MessageBox.Show("text not allowed, Values only!!")
            End If
        End If

        If 15 >= Val(hel.Text) Or 30 < Val(hel.Text) Then
            MessageBox.Show("helix angle is out of range!!")
            Exit Sub
        End If

        Dim c, dp, dg, mn, mt, aw, dw, b, qt, qn, U, mg, kt, cd As Double
        Dim np, ng, nmin, nmax As Integer
        c = Val(centerd.Text)
        U = (Val(hel.Text) * Math.PI) / 180
        qn = (20 * Math.PI) / 180

        mg = Val(gearratio.Text)

        qt = Math.Atan(Math.Tan(qn) / Math.Cos(U))

        kt = (qt * 180) / Math.PI

        Dim n1, n2, n3, k1, k2, k3, k4, bc, nh, nh2, JL As Double

        n1 = Math.Cos(U)
        n2 = (Math.Sin(qt)) ^ 2
        n3 = (Math.Cos(U)) ^ 2
        k1 = 1 + (2 * mg)
        k2 = Math.Sqrt((mg ^ 2) + (k1 * n2))


        bc = ((2 * n1) / (k1 * n2)) * (mg + k2)
        nmin = ((2 * n1) / (k1 * n2)) * (mg + k2)

        nh = bc - nmin
        If nh > 0 Then
            nmin = nmin + 1
```

```
Else
   nmin = nmin
End If

np = nmin

JL = np * (mg)
ng = np * (mg)

nh2 = JL - ng
If nh2 > 0 Then
   ng = ng + 1
Else
   ng = ng
End If

k3 = (np ^ 2)
k4 = (k3 * n2) - (4 * n3)

nmax = k4 / ((4 * n1) - (2 * np * n2))

'determination of module and  pitch diameter of pinion and gear'
      dg = ((2 * c * mg) / (mg + 1))

      dp = (2 * c) - dg

   mn = (dp * n1) / np
   lblmn.Text = "Normal Module (mm)"
   ansmn.Text = Math.Round(mn, 2)

   mn = Val(tm.Text)
   dg = (ng * mn) / n1
   dp = (np * mn) / n1

cd = (dp + dg) / 2

lblmn.Text = "Center Distance (mm)"
ansmn.Text = Math.Round(cd, 2)

'determining transverse module'
mt = mn / (Math.Cos(U))

'determining addendum'
aw = 1 * mn

'determining dedendum'
dw = 1.25 * mn

'determination of root diameter for pinion and gear'
Dim rtp, rtg As Double
rtp = dp - (2 * dw)
rtg = dg - (2 * dw)

'determining transverse pitch'
Dim pt As Double
pt = Math.PI * (mt)

'determining axial pitch'
Dim px As Double
px = pt / (Math.Tan(U))

'determining face width'
b = 4 * Math.PI * mn

'determining normal circular pitch'
Dim ncp As Double
ncp = mn * Math.PI

'determining pinion and gear base diameter'
Dim dbp, dbg As Double
dbp = dp * Math.Cos(qt)
dbg = dg * Math.Cos(qt)

'detremining base helix angle'
Dim ba, bha As Double
ba = Math.Atan(Math.Tan(U) * Math.Cos(qt))
bha = (ba * 180) / Math.PI

'determining gear ratio'
Dim gr As Double
gr = ng / np

'determining pinion virtual number of teeth'
Dim nvp As Double
nvp = (np / (Math.Cos(U)) ^ 3)

'determining gear virtual number of teeth'
Dim nvg As Double
nvg = (ng / (Math.Cos(U)) ^ 3)

'detremining pinion tooth thickness half angle'
Dim uv1, uvp As Double
uv1 = 90 / nvp
uvp = (uv1 * Math.PI) / 180

'detremining gear tooth thickness half angle'
Dim uv2, uvg As Double
uv2 = 90 / nvg
uvg = (uv2 * Math.PI) / 180

'detremining pinion chordal tooth thickness'
Dim cttp As Double
cttp = nvp * mn * Math.Sin(uvp)

'detremining pinion chordal tooth thickness'
Dim cttg As Double
cttg = nvg * mn * Math.Sin(uvg)

'Determination of root fillet radius rf'
Dim rf As Double
rf = 0.38 * mn

'Determination of speed ratio'
Dim spr As Double
spr = 1 / mg

'outside diameter for gear and pinion'
Dim outp, outg As Double

outp = dp + (2 * mn)
outg = dg + (2 * mn)

'determination of face-contact ratio mf'
Dim mf As Double
mf = b / px

'determination of whole tooth depth ht'
Dim wholeht As Double
wholeht = 2.25 * mn
```


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