

Computer Aided Design and Drafting of Helical Gears

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Abstract

An interactive user friendly low cost software called “CADDgear” was developed in this study to facilitate the design and drafting of helical and spur gears thereby generating reliable data for use in manufacturing process. The software was developed, using JAVA programming language, as a tool for determining the design parameters and producing accurate and efficient 3D (three dimensional) and 2D (two dimensional) detail working drawings of helical gears. The study considered the existing approaches in use for the design of helical gears and then established a design analysis procedure for helical gear design. The established procedure was implemented through the developed software so that a substantial saving in term of time and cost of production of the design is obtained. The outcome of this research would enhance the designer’s productivity by reducing the time required to synthesis, analyze and document helical gear design. This would permit a thorough analysis of a large number of design alternatives. Results generated by the software shows very good agreement with that obtained through manual calculation using the established procedure. It was observed that the developed software successfully increase productivity over manual gear design and drafting by approximately thirty-four times in term of the time required for the design.

Keywords: gear design, interactive, efficient, reliability, computer aided, design and drafting

INTRODUCTION

In recent times, gear design has become a highly complicated and comprehensive subject. A designer of a modern gear drive system must remember that the main objectives 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, runs reasonably free of noise and vibrations and which requires little maintenance. Gear problems are common occurrences in the gear industry and are often the result of improper design, wrong selection of material for a given application. However, increase in demand for gears with high load carrying capacity and increased fatigue life require improvements in tooth form. The strength of the gear teeth has to be improved to meet increase load. For a better understanding of the gearing system Figure 1 (a) and (b) depict the basic geometry and nomenclature of the spur gear and helical gear respectively. Computer aided drafting (CAD) technology was first introduced in the mid 1960’s as a tool for the production of drawings without the use of traditional drafting tools. This technology had gone through lots of development so that Computer-aided Design and Drafting (CADD) system now exists. The CADD system allows design modification to be performed easily and efficiently. Also, the analysis and optimization phases of the design are easily and accurately performed by the computer while the designer will find these task times consuming and tedious without the use of computer.

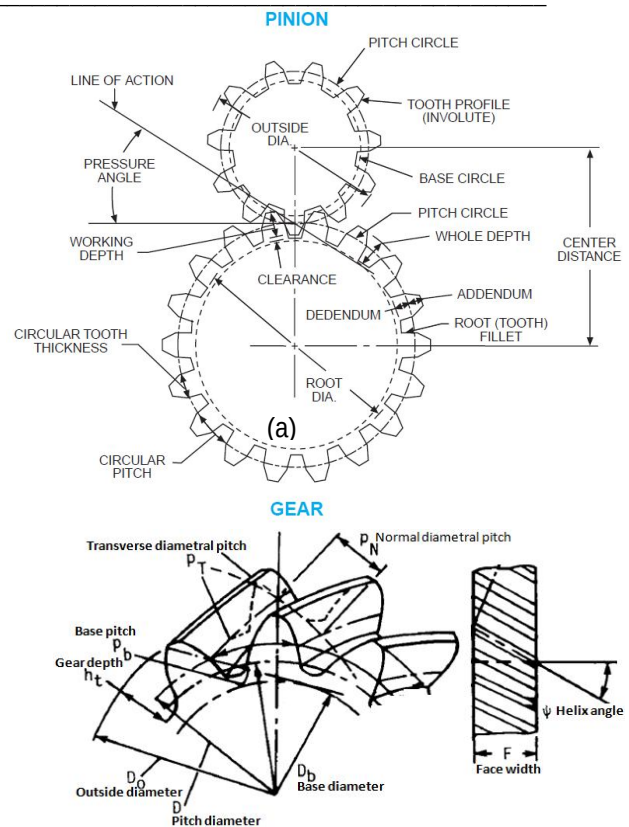


Figure 1 : (a) Spur Gears Geometry (Roymech, 2009); (b) Helical Gear Geometry (adapted from Brown, 2004)

In gear system, theories relating to failure, wear, and strength of material form the basis of the design procedures. Many researchers had studied gear design and analysis using computer aided system approach. Venkatesh et al., (2010) worked on design, modeling and manufacturing of helical gear for marine engine. CATIA software was utilized to study the structural analysis of a high speed helical gear used in marine engine. This work was carried out to investigate the stress generated and the deflection of the tooth for different materials. Evgueni et al., (2003) studied design of high contact ratio spur gear to reduce static and dynamic transmission error. A computer program was developed to calculate static and dynamic transmission error of gears under load. The analysis of the gear using this program showed that high contact ratio which can be achieved by decreasing pressure angle and increasing tooth height have much less static and dynamic transmission error than standard gear. Nordiana et al., (2007) studied computer aided design of a spur gear called Cadgear (2007), an interactive computer graphics able to generate accurate data and produce precise and efficient 2D design drawings based on strength and wear calculations. Shanmugasundaram et al. (2010) studied profile modification for increasing the tooth strength in spur gear using CAD. They examined the tooth failure in spur gear and the work revealed that the circular root fillet design is particularly suitable for lesser number of teeth in pinion and whereas the triochoidal root fillet gear is more opt for higher number of teeth

Nwosu and Iwuoha (2010) analyzed the failure of the gearbox unit after about 10,000 service hours when the design service life of 45,000 hours was expected. The observed failure was due to design and manufacturing errors. This work came out with findings to prevent these failures. These include adequate sizing of the gear teeth and face width as well as better choice of materials with higher allowable static stresses to bear load imposed. However, automatic design of helical gears and the automatic generation of the three dimensional drawing of both the spur and helical gears require attentions to further improves on the design and drafting of these gears. Hence, this study developed an interactive computer graphic system for the design and drafting of both spur and helical gears. The software determines the geometrical (design) parameters of the appropriate gear, use calculated values of face width to check for adequate strength of gear tooth, compare wear load and dynamic load for gear check for specific area of application such as paper box machine, cement kiln, steel mill drives, clothes washing machine, printing press, computing mechanism, automotive transmissions, radar antenna drive, marine propulsion drive and gyroscope and finally produces accurate 2D (two dimensional) and

3D (three dimensional) detail working drawings of the designed gear.

RESEARCH METHODOLOGY

Visitations were made to some manufacturing companies in Oshogbo, Akure, Ondo Ibadan and Lagos, to examine how helical gears are been designed and produced. Also, consultations were made with Engineers having very good experienced in gear design through oral interview during the course of the visitations. Thereafter standard Machine design textbooks were consulted in respect of the analysis and design of gears. Based on the knowledge and information that was gained and obtained from the visitations and consultations, a procedure for designing helical gears was established in the design analysis section.

The established procedure was used to design and implement a software for helical gear design and drafting using JAVA programming language. This is because Java is a simple, object-oriented, network-savvy, robust, secure, architecture neutral, portable, high-performance and dynamic language. It possesses automatic garbage collection, thereby simplifying the task of Java programming. The programming task is devoid of a lot of esoteric training and user friendly software can be developed. The software was designed to validate users input, check for adequate strength of gear tooth, check for adequate wear resistance, calculate gear design parameters for use in the manufacturing process and draft the 2D and 3D drawings of the designed gears. The software design approach ensures that it is menu driven and user friendly.

DESIGN ANALYSIS

General Design Considerations

The proper design of gears for power transmission in a particular application is a function of (a) the expected transmitted power, (b) the driving gear's speed, (c) the driven gear's speed or speed ratio and (d) the center distance (Khurmi and Gupta 2009).

The essential conditions that must be met in the design of a gear drive are:

- (i) The gear teeth will not fail under static loading or dynamics loading during normal conditions
- (ii) The gear teeth should possess good wear resistance so that they will not fail under static loading or dynamics loading during normal conditions and so that gear life is satisfactory
- (iii) Gears with smaller modules are preferred in order to optimize the space requirement.
- (iv) The gears must operate together without tooth interference with proper length of contact and without noise (Singh, 1997).

Generally, the design of gear tooth involves essentially the determination of proper pitch and face width for adequate strength, durability and economy

of manufacture. Thus helical gears are designed from the standpoint of strength and wear.

Designs for Adequate Strength of Helical Gear Tooth

In helical gears the contact between mating gear starts at one end move along the line of contacts. Thus in order to find the strength of helical gears, a modified Lewis equation reported by Singh,(1997) for determined strength of gear tooth is given by Equation (1).

$$b = \frac{F_t}{(\sigma_b \times C_v) m y^1 \pi} \quad (1)$$

Where b is the face width (mm), F_t is tangential force (N), m is the module, y^1 is the Lewis form factor corresponding to formative number of teeth, σ_b is the allowable static stress (N/mm²), σ_b

is $\frac{\text{Permissible stress}}{\text{Safety factor}} \text{ N/mm}^2$ and C_v is the

velocity factor. The factor of safety applicable to a variety of drive and load conditions are as presented in Table 1. The Face width are commonly used for

strength check and it lies between ranges $\frac{1.15\pi m}{\tan \phi}$ to

20m for helical gears and between 9m to 13m for spur gears

Table 1: Factor of Safety for Gears

Types of drive and load	Factor of safety
Steady load on a single pair	3
Suddenly applied on a single pair	4
Steady load of gears of train beyond first mesh	5
Sudden load on gears of train beyond first mesh	6

Source: (Khurmi and Gupta, 2009)

Gear Parameters for Strength Consideration.

Velocity factor

When gears are running at high speed, the gear may be subjected to dynamic effect. To account for this a velocity factor C_v is considered, thus Barth equations for velocity factor for helical gears as given in Equations (2) – (5).

$$C_v = \frac{6}{6 + v} \quad \text{for peripheral velocities from 5m/s to 10m/s} \quad (2)$$

$$= \frac{15}{15 + v} \quad \text{for peripheral velocities from 10m/s to 20m/s.} \quad (3)$$

$$= \frac{0.75}{0.75 + \sqrt{v}} \quad \text{for peripheral velocities greater than 20m/s} \quad (4)$$

$$= \frac{0.75}{1 + v} + 0.25, \quad \text{for non-metallic gears} \quad (5)$$

Also, the Velocity factor C_v for spur gears is given is as given in Equations (6) – (9).

$$C_v = \frac{3}{3 + v}, \quad \text{for ordinary cut gear operating at velocities up to 12.5m/s} \quad (6)$$

$$= \frac{4.5}{4.5 + v}, \quad \text{For carefully cut gear operating at velocities up to 12.5m/s} \quad (7)$$

$$= \frac{6}{6 + v}, \quad \text{for ground metallic gear operating at velocities up to 20m/s} \quad (8)$$

$$= \frac{0.75}{0.75 + \sqrt{v}}, \quad \text{for precision gears cut with accuracy operating at 20m/s} \quad (9)$$

Where, v is the pitch line velocity given by Equation (10)

$$v = \frac{\pi D n}{60} \quad (10)$$

Where,

D is pitch diameter (mm), n is number of revolution per min (rpm).

Tangential Force (F_t)

The tangential force F_t can be related to power transmitted. To account for gear service conditions, a modified equation is given as in Equation (11).

$$F_t = \frac{1000 \times P \times C_s}{V} \quad (11)$$

Where: P is power transmitted (kw), V is pitch line velocity (m/s) and C_s is service factor

The service factor (see Table 2) is a multiplier applied to the known load which redefines the load in accordance with the conditions at which the drive will be used or it can be a divisor which defines the rating in accordance with the drive conditions.

Table 2: Service Factor for Gears

Type of load	Intermittent on 3hrs /day	8-hrs/day	Continuous 24/hrs/day
Light shock	1.00	1.54	0.80
Medium shock	1.25	1.54	1.80
heavy shocks	1.54	1.80	2.00

Source: (Khurmi and Gupta, 2009)

Lewis Form Factor

The form factor based on the formative number of teeth, y^1 , can be obtained from the relation in Equations (12) – (15).

$$y^1 = 0.124 - \frac{0.684}{N_F} \quad \text{for } 14 \frac{1}{2}^\circ \text{ full depth} \quad (12)$$

$$= 0.154 - \frac{0.912}{N_F} \text{ for } 20^\circ \text{ full depth} \quad (13)$$

$$= 0.175 - \frac{0.841}{N_F} \text{ for } 20^\circ \text{ stub system} \quad (14)$$

Where N_F is the formative number of teeth, N is number of teeth, ϕ is helix angle and is given as

$$N_F = \frac{N}{\cos^3 \phi} \quad (15)$$

Design for Helical Gear Tooth Dynamic Load

When a pair of gears is running at moderate or high speeds, there is generation of noise and inaccuracies in the meshing of gear teeth in action. As a result, the gear is subjected to dynamic effect. Shigley (2001), gives the dynamic tooth load on the helical gear as in Equation (16).

$$F_d = F_t + \frac{21V(C b \cos^2 \phi + F_t) \cos \phi}{21V + \sqrt{C b \cos^2 \phi + F_t}} \quad (16)$$

Where: F_d is the dynamic load (N), F_t is Tangential force (N), V is pitch line velocity (m/s), b is face width of the pinion (mm), ϕ is helix angle (degree) and C is values of deformation factor based on the tooth form material and the degree of accuracy with which the tooth is cut (KN/m). The pitch line velocity depending on the area of application of the gear being designed could be obtained from Table 3.

Table 3: Recommended AGMA Quality Numbers Versus Pitch Line Speed and Area of Applications

Pitch line Speed (m/s)	Quality Number	Area of Application
0-4	6-8	Paper box making machine, Cement kiln and steel mill drives,
4-10	8-10	Clothes washing machine, Printing press, Computing mechanism,
10-20	10-12	Automotive transmission, Radar antenna drive, Marine propulsion drive,
Over 20	12-14	Gyroscope

Source: AGMA Standard (2001)

Design for Helical Gear Tooth Wear Loads

The maximum load that gear teeth can carry without premature wear depends upon the radii of curvature of the tooth profiles and on the elasticity and surface fatigue limits of the material. The limiting wear load F_w for helical gears may be determined by the Buckingham equation for wear given by Equations (17) – (20)

$$F_w = \frac{D_p b Q K}{\cos^2 \phi} \quad (17)$$

$$Q = \frac{2Dg}{Dp + Dg} = \frac{2Ng}{Np + Dg} \quad (18)$$

$$K = \frac{(\sigma_{es})^2 \sin \phi_N}{1.4} \left[\frac{1}{E_p} + \frac{1}{E_g} \right] \quad (19)$$

For steel gears appropriate value of δ_{es} is given as

$$\delta_{es} = (2.75(BHN) - 70) \quad (20)$$

Where: F_w is the wear load (N), D_p is pitch diameter of pinion (mm), D_g is pitch diameter of gear (mm), b is face width of the pinion (mm), Q is ratio factor, K is load stress factor (N/m²), σ_{es} is surface endurance limit (N/m²), ϕ_N is normal pressure angle (in degree), E_p and E_g are Young's modulus of pinion and gear material (N/mm²), N_p and N_g are the Numbers of teeth of pinion and gear and BHN is Brinell hardness. It has been observed that for the design to be satisfactory from the consideration of wear the value of F_w must not be less than F_d . If otherwise, then the hardness of the designed gear is increased by equating $F_w = F_d$ to find desired value of Brinell hardness.

SOFTWARE DEVELOPMENT

A graphics 2D (java 2D) object maintain a whole heap of information that determine how the 2-D module of the software was designed using the six JAVA attributes of paint, stroke, font, transformation, clip and composite. The 3D design of the gears was achieved by the use of AutoCAD with the help of affine transformation rendering and the 3-D images were dynamically loaded into the JAVA workspace of the application. The graphical representation of the algorithm is as presented in Figure 2. Programming of the software was done in respect of the flowchart using the JAVA programming language. Based on the design steps analyzed in the previous section, the following procedural steps/algorithm was adopted within the developed software for the design and drafting of helical gears.

(i) From design requirement, identify the driver's speed in rev/min and the power to be transmitted P in kw

(ii) Select factor of safety for gears F_{os} and value of service factor for type of load C_s using Table 1 and Table 2 respectively.

(iii) Ensure that the stress on the pinion and/or the gear shall have lower value than the permissible stress.

(iv) Select type of material for the driver and driven gears and compute gear parameters (the modulus of elasticity, poisons ratio and strengths values of the materials available on the software had been stored in the software database).

(v) Quality number verifies area of application chosen from Table 3 if not OK a module range is

suggested in order to have satisfactory design based on area of application

(vi) Computed face width is used to check for adequate strength of the gear tooth. The face width

should lie between $\frac{1.15\pi m}{\tan \phi}$ to $20m$ for helical gear

and between $9m$ to $13m$ for spur gear, where m is the module. If the face width lies within the computed range the design is satisfactory from stand point of strength otherwise a different material should be selected and re-computation of gear parameters is done until the adequacy in strength of the designed gear is achieved.

(vii) Compute the endurance stress, wear load and dynamic load and check to make sure that wear load F_w is greater than or equal to F_d otherwise increase the hardness of the pinion.

(viii) When all conditions required for designed gears to be adequate have been satisfied with respect to the calculation in the seven steps above, the design parameters are then used to draft the 2D as well as the 3D diagram of the designed gears. This is done through the drafting modules wherein the geometries had been mathematically represented within the developed software.

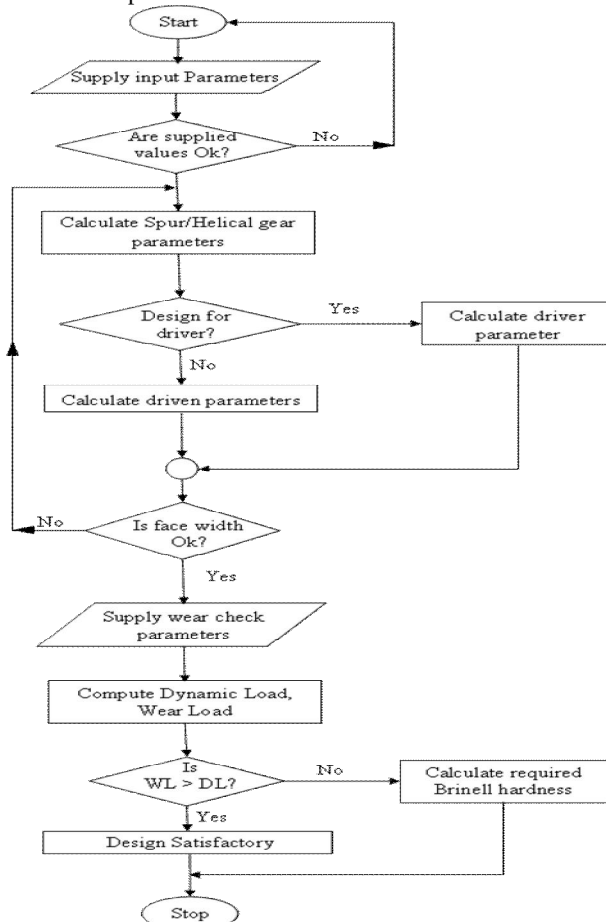


Figure 2: Flowchart Showing Algorithm of Helical and Spur Gear Detailed Design

RESULTS AND DISCUSSION

CADDgear Operational Procedure

Launch the application and the output screen for the helical gear geometric parameter appears as shown in Figure 3. Enter the required parameters on the window at the first Row Tabs basically the driver teeth, driven teeth, module and the area of application. The area of application is optional. Any of the area of application can be chosen provided it satisfies the quality number required as recommended by the AGMA Quality Number versus Pitch line speed and area of application, otherwise the application suggest an alternative range of module.

Select the required pressure angle, helix angle and the manufacturing process as desired. Then, proceed by clicking "Calculate parameter". This gives the result for the basic helical gear parameters for driver and driven and other geometric parameters at the right panel. At the bottom left are two buttons with the inscription "Go to Detailed Design" and "Exit Window". Proceed by clicking on "Go to Detailed Design", which loads the output screen for helical gear detailed design as shown in Figure 4. Enter the required driver's speed, power, service factor and factor of safety for the load type through the text box at the top left corner of the Graphical User Interface (GUI) window in Figure 4. On the same GUI window, select the type of material to be used for the gear, the "condition" and the "minimum tensile strength" required. Consequently, the software displays the required parameters for checking the design for adequate strength ("strength check" bar). By clicking on "compute" in Figure 4, a "message" window for checking the design appears as shown in Figure 5. These message windows are used to check for the appropriate quality number and face width which is the criterion for checking for the strength of gear tooth to ensure that the design is satisfactory.

If the quality number is not satisfactory, the software was designed to suggest alternative range of module different from the ones earlier selected and then the window switches back to the helical gear geometric parameter screen which allows the module to be re-selected and the parameters re-calculated. Alternatively, the area of application can be selected in accordance with the corresponding pitch line velocity computed in the helical gear detailed design window as recommended by the AGMA quality number versus pitch line speed and area of application as given in Table 3.

To satisfy the face width condition, the computed face width must fall between a given range, otherwise it is not satisfactory, hence the application would suggest alternative material with reference to the minimum tensile strength of the material. Note here that the minimum tensile strength of any material is inversely proportional to the face width, hence the

smaller the minimum tensile strength of the material selected, the higher the face width and vice versa. After the face width have been found satisfactory load the gear error constant, to conduct wear check, by clicking the “load error” tab which brings out the output screen for error constant as shown in Figure 6. Select appropriate input parameters as required and then click “compute”, then click “check” and then check the remark box to ensure the design is

satisfactory based on the condition $F_w \geq F_d$ i.e maximum load for wear is more than the dynamic load on the tooth. Otherwise the application suggests an alternative option of increasing the Brinell hardness. Provided all the above is satisfactory, and then click on the “output” at the bottom left button of the Figure 6, the resulting gear design parameter and the 2D as well as the 3D drawing is then displayed.

Figure 3: Output Screen for the helical gear geometric parameters

Figure4: Output Screen for helical gear detailed design



Figure 5: Message Windows

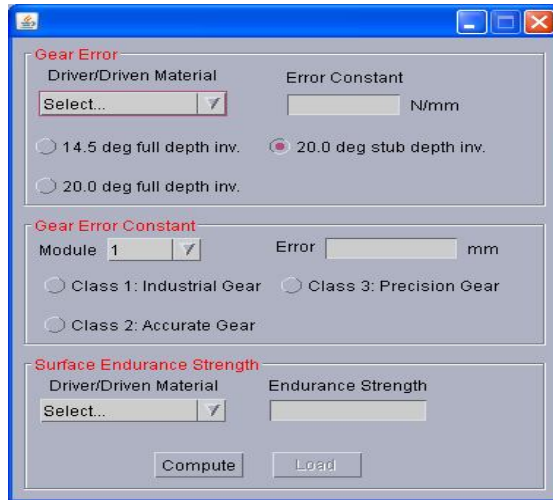


Figure 6: Output Screen for Error Constant

Software Validation Using A Case Study

Case study of a sample problem from standard text materials were considered to test the software performance. The result obtained from the software is compared with that obtained from the manually generated solution for validation. The case study used is as presented herein:

A pair of helical gears which is to be part of the drive for an automotive transmission engine requires 15kW. The teeth are 20^0 stub in diametral plane and have helix angle of 45^0 the pinion runs at 10,000 rpm and with 32 teeth of a reduction ratio of 4:1. Design the gear from the static strength consideration and check for wear.

Manually Solved Solution to Case Study

The best combination of materials to give satisfactory design are given as

Material	Condition	Brinell hardness	Minimum Tensile Strength/ m^2
Carbon steel (0.55% carbon)	Hardened+Tempered	223 min	700
Carbon steel (0.55% carbon)	Cast	223 min	700

Determination of the Number of Teeth

Number of teeth of pinion = 32
Since the speed ratio is 4:1

Let $N_p = 32$,

$$N_g = 32 \times 4 = 128$$

Determination of the Formative Number of Teeth for the Pinion

$$N_{PF} = \frac{N}{\cos^3 \phi} = \frac{32}{\cos^3 45} = 90.50$$

Determination of the Formative Number of Teeth for the Gear

$$N_{GF} = \frac{128}{\cos^3 45} = 362.03$$

It is necessary to determine which one is weaker between pinion and gear,

Determination of the Lewis form factor y

Tooth form factor for the pinion for 20^0 stub teeth from Equation (14)

$$y'_p = 0.175 - \frac{0.841}{N_{PF}} = 0.175 - \frac{0.841}{90.50} = 0.165$$

Tooth form factor for the gear for 20^0 stub teeth,

$$y'_g = 0.175 - \frac{0.841}{N_{GF}} = 0.175 - \frac{0.841}{362.03} = 0.1726$$

$$\delta' y'_p = 700 \times 0.1657 = 115.99$$

$$\delta' y'_g = 700 \times 0.1726 = 120.82$$

Since the load carrying capacity of the tooth is a function of δy product and $\delta' y'_p < \delta' y'_g$ the pinion is weaker and it form basis for the design.

Selection of Module m

Let module $m=1$

$$d_p = m N_p = 1 \times 32 = 32 \text{ mm}$$

$$d_g = m N_g = 1 \times 128 = 128 \text{ mm}$$

Determination of the Pitch Line Velocity v

From Equation (10)

$$V = \frac{\pi D n}{60} = \frac{\pi \times 32 \times 10000}{60 \times 1000} = 16.75 \text{ m/s}$$

Determination of the Velocity Factor C_v

Since the velocity is less than 20m/s, Equation (3) gives

$$C_v = \frac{15}{15 + v} = \frac{15}{15 + 16.5} = 0.476$$

Determination of the Tangential Force F_t

Assuming continuous operation for 3hrs/day, Table 2 gives the service factor for light shock load as $C_s=0.8$. Therefore, from Equation (11)

$$F_t = \frac{1000 \times P \times C_s}{V} = \frac{1000 \times 15 \times 0.8}{1675} = 716.4 \text{ N}$$

Determination of the Face width, b

From Table 1, using a factor of safety of 3 which represent a steady load on a single pair, Equation (1) gives

$$b = \frac{7164}{\left(\left(\frac{700}{3}\right) \times 0.476\right) \times 1 \times 0.1656 \times \pi} = 12.4 \text{ mm}$$

Since the face width for helical gear lie between ranges $\frac{1.15\pi m}{\tan \phi}$ to $20m$, the required face width range is 3.61mm – 20mm. Therefore the calculated face width is okay.

Determination of the Dynamic load

Here the design is checked in respect of the dynamic load and the wear effect

From Equation (16)

$$F_d = F_t + \frac{21v(cb \cos^2 \phi + F_t) \cos \phi}{21v + \sqrt{(cb \cos^2 \phi + F_t)}}$$

For $m=1$, the error in action for class 2 accurate gear is 0.0254. Hence, the value of C by interpolation is

$$\frac{0.050 - 0.025}{0.0254 - 0.025} = \frac{60040 - 30020}{C - 30020} \quad C = 30500$$

$$\begin{aligned} F_d &= F_t + \frac{21v(cb \cos^2 \phi + F_t) \cos \phi}{21v + \sqrt{(cb \cos^2 \phi + F_t)}} \\ &= F_t + \frac{21 \times 16.75(305 \times 12.4 \times \cos^2 45 + 716.4) \cos 45}{21 \times 16.75 + \sqrt{(12.4 \times 305 \times \cos^2 45 + 716.4)}} \\ F_d &= 2279 \text{ N} \end{aligned}$$

Determination of the Wear Load

The wear load is estimated From Equation (17) as,

$$F_w = \frac{D_p b Q K}{\cos^2 \phi}$$

$$Q = \frac{2D_g}{D_p + D_g} = \frac{2 \times 128}{32 + 128} = 1.6$$

$$K = \frac{(\sigma_{es})^2 \sin \phi_N}{1.4} \left(\frac{1}{E_p} + \frac{1}{E_g} \right) \text{ Note: normal}$$

pressure angle (ϕ_N), from equation 3.5,

$$\phi_N = \tan^{-1}(\tan \phi \cos \phi) = \tan^{-1}(\tan 20 \cos 45) = 14.4^\circ$$

$$Q = \frac{2D_g}{D_p + D_g} = \frac{2 \times 288}{72 + 288} = 1.6$$

$$K = \frac{(\sigma_{es})^2 \sin \phi_N}{1.4} \left(\frac{1}{E_p} + \frac{1}{E_g} \right)$$

Average brinell hardness for 0.55% carbon steel (Hardened and tempered) and 0.55% carbon steel (cast)

$$= \frac{223 + 223}{2} = 223$$

Hence, the value of σ_{es} for carbon steel can be interpolated from table as,

$$\frac{223 - 200}{250 - 200} = \frac{\sigma_{es} - 480}{618 - 480}$$

$$\sigma_{es} = 543.48 \text{ MN/mm}^2.$$

Also, E_p and E_g are equal and was obtained from

table as $205 \times 10^3 \text{ N/mm}^2$, and

$$\begin{aligned} \therefore K &= \frac{(543.48)^2}{1.4} \sin 14.4^\circ \left(\frac{1}{205 \times 10^3} + \frac{1}{205 \times 10^3} \right) \\ &= 0.512 \text{ N/mm}^2 \end{aligned}$$

Maximum limiting load for wear

$$\begin{aligned} F_w &= \frac{D_p b Q K}{\cos^2 \theta} = \frac{80 \times 32 \times 1.6 \times 0.512}{\cos^2 45} \\ &= 665.1 \text{ N} \end{aligned}$$

Since F_w is less than F_d , the design is not satisfactory from the stand point of wear. To ensure satisfactory design, we equate $F_w = F_d$. To design against wear, F_w must be at least being equal to dynamic load. We then have to get new stress factor K.

$$K = \frac{F_w \cos^2 \phi}{D_p b Q} = \frac{32279 \times \cos^2 45}{32 \times 12.4 \times 1.6} = 1.79$$

We then calculate new endurance strength

$$\begin{aligned} (\sigma_{es})^2 &= \frac{K}{\sin \phi_N \left(\frac{1}{E_p} + \frac{1}{E_g} \right)} \\ &= \frac{1.79}{\sin 14.4^\circ \left(\frac{1}{205 \times 10^3} + \frac{1}{205 \times 10^3} \right)} = 10163.03 \text{ MN/m}^2 \end{aligned}$$

$$\begin{aligned} \text{Hence, } 1016.303 &= 2.75 \text{ BHN} - 70 \\ \text{BHN} &= 395 \end{aligned}$$

Software Application to the Case Study

The appropriate input used to manually obtain the design parameter in the design of helical gear conducted in the preceding section was supply to the software and the result of the software analysis is as given in Figure 7.

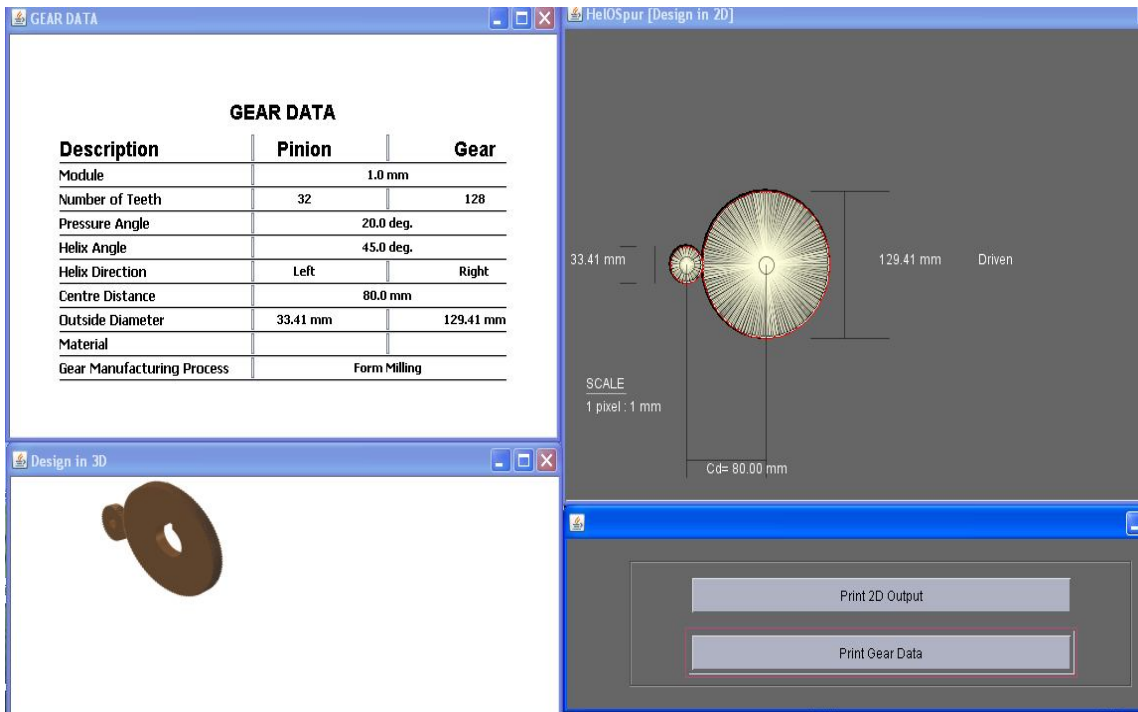


Figure7: Result of CADDgear gear design data, 2D and 3D drawing

Comparing the results in this Figure with the ones obtained from the manual calculation procedure shows that the result generated from the software is accurate (see Table 4). Therefore the software can be reliably used for the design of helical gears. Furthermore the 2 dimensional and 3 dimensional drawings of the designed gears are also accurately drafted by the software. While it takes an individual that did the manual calculation and drafting of the gears' details approximately 2 hrs 49 minutes to complete the job it took the Computer aided system (CADDgear) and its operator only 5 minutes. This revealed that the computer aided system have been able to increase the productivity of helical gear design and drafting by approximately thirty-four times

Table 4: Comparison of Manually Solved and Computer Generated Values on Helical Gear Design

Parameters	Manual calculated values	Computer generated parameters
Tangential Tooth Load(N)	716.4	716.197
Pitch Line Velocity(m/s)	16.75	16.755
Velocity Factor	0.476	0.472
Face Width(mm)	12.4	12.482
Dynamic Load(N)	2227.805	2333.379
Wear Load(N)	665.1	655.729
Brinell Hardness(min)	395	398.259

CONCLUSIONS

This study developed a software called CADDgear for the design of helical gears which is also extendible to spur gears design as well (when helix angle is zero). The result of the software was

found to be in good agreement with that obtained from manually calculated procedure. The proprietary software validates user input and relieve designer of the risk of errors in calculations that may results from design processes.

The results generated from "CADDgear" when compared with the manually calculated values of helical gear design from standard text materials shows no appreciable change in the calculations of face width (for strength check) and wear check. The developed software would facilitate the design of helical and spur gears based on given areas of application such as paper box making machine, cement kiln, steel mill drives, clothes washing machine, printing press, computing mechanism, automotive transmissions, radar antenna drive, marine propulsion drive and gyroscope. It would also facilitate the generation of reliable data for use in the manufacturing process and future design activities. The system which was designed to validate user inputs before calculating gear data could also generate accurate 2D and 3D drawings, which could be used as working drawings, for the designed gears. CADDgear is interactive, user friendly and runs on Windows operating system. The software would provide substantial saving in term of time and hence cost of production. The software was found to increase productivity by an estimated thirty-four times in term of the time required to do the job. However it is important to mention here that the software is suitable or applicable only to the design and drafting spur and helical gears alone.

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