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Analysis for Involute Spur Gears, the Bendings and Pittings Stress on Gears

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Abstract: There are some constraints which affect the design of involute designs such as scoring wear, interference, bending stress, strength, pitting resistance etc. The concentration is focused on spur gear sets which are used to transmit motion between parallel shafts. The method of using manual calculations applied by gear designers and manufacturers to determine the bending and pitting stress on gears is time consuming, inefficient and can easily generate errors. This work aims to design gear analytically using AGMA standard, determimining the bending and contact stresses on the gear teeth usingComputer Aided and Computer Engineering Softwares to make gear stress calculations. Parameters in the AGMA stress equations were determined numerically, with MATLAB and Visual Studio software which was used to create graphical user interfaces that allows the bending and pitting stress on gears to be easily and accurately calculated. Results from the test performed showed that the bending fatigue strength in both the asymmetric tooth form and optimized fillet form is higher than that of baseline designs. There is a significant increment in scuffing resistance in the asymmetric tooth form when compared with a conventional symmetric involute tooth design. A variety of bending and pitting stresses of spur and helical gears problem can be handled by the created software, which can be useful for the gear designers, educaitional institutes and likes in gear problems.

Keywords: Spur Gears, Bending Stress, Pitting Stress, Surface Durability, Stress Calculations

1. Introduction

Modal analysis is evolved as standard tool for structural dynamic problem analysis and design optimization. The research area is very dynamic with a focus on performance improvement, test, cost reduction and the development of new application areas. In experiment and computational analysis, validation of ten percent is allowed [1], additional tolerance for improvement of gear drives with unidirectional load cycles are available for direct gear design for asymmetric tooth profiles [2, 3]. Using this method, it appeared that the most affecting variable of changing the value of load transverse factor is helix angle, but, despite of this, the profile shift coefficients also affected the changing value of load transverse factor. It is noted that for any number of teeth and any gear ratio, this method achieves a value of 1 of the load transverse factor, which therefore corresponds to uniform load distribution [4-6], test results demonstrated higher bending fatigue strength for both the asymmetric tooth form and optimized fillet form compared to baseline designs. Scuffing resistance was significantly increased for the asymmetric tooth form compared to a conventional symmetric involute tooth design. Contact stress of existing gear train was calculated and compared with fatigue strengths of gear material. Effect of fatigue strength of gear material was analysed by doing the static analysis of the gear in order to find the Von-mises stress on the tooth of the gear in meshing, [7, 8].

The strength of these modified teeth was studied in comparison with the standard design. The analysis demonstrates that the novel design exhibit higher bending strength over the standard trochoidal root fillet gear. The result reveals that the circular root fillet design is particularly suitable for lesser number of teeth in pinion and where as the trochoidal root fillet gear is more opt for higher number of teeth, [9, 10]. Vikash showed that experimental calculation results from uniform load distribution model are different to the estimation from the non-uniform load distribution on gear teeth. But American Gear Manufacturing Association (AGMA) and International Standard Organization (ISO) assumptions based on uniform model of load distribution which is not accurate. In order to balance the discrepancies, some modifications factors were applied in derived mathematical relation for gears; such as minimum elastic potential energy method. Computers usage [12, 13], in this scenario is better approaches for the prediction of effect of whatever studied. To optimize the gear gear design, some procedures may be useful such as extraction of all the figure from the various graphs, using the curve fittings on the graph to obtain an equation with a MATLAB software combined with AGMA equation, with application of computer programming which has been tested successfully, [14-16]. The model and the solution methods, however, must be chosen carefully to ensure that the results are accurate and that the computational time is reasonable.

Determination of the bending and pitting stresses of any spur and helical gears with their corresponding factor of safety can be drtermined using AGMA methodology, [17, 18]. The Contact stress of existing gear train is calculated and compared with fatigue strengths of gear material. If this stress on gears are higher than fatigue strengths means gears are failed due to fatigue. To reduce the contact stress by increasing the module of gear. The contact stress are calculated by Hertz's Equation and Strain gauge is used for the experimental investigation of the stress field [19, 20].

The present work aimed at analytical design of girth gear by using AGMA standard. Bending and contact stresses on the gear teeth are calculated. The validation of the stresses were done by using ANSYS software. Validation of the package was achieved by comparing its operational principles and outputs with that of contemporary packages, and existing solutions of standard problems and set of standard literatures.

2. Methodology

The procedure for this research involved five major tasks:

(i) Determination of the methods of evaluating the numerical value (s) of all the parameters in the AGMA equations stated earlier.

(ii) Identification of the parameters to be supplied by the end user of the proposed MATLAB and VISUAL STUDIO software program.

(iii) Converting the equation for the bending and contact stresses and the procedure for determining the operational and allowable stresses on the gear tooth into suitable programmable algorithm.

(iv) Development of the program algorithm into MATLAB software codes and converting it into C++ for use in VISUAL STUDIO software with the use of a compiler.

(v) Development of an interactive Graphical User Interface which accepts inputs from the user and outputs results in numerical formats.

Determination of Parameters in the AGMA stress Equations

Tangential Transmitted Load, W_t: determined from the power rating of the machine in Watts, the pinion speed in revolutions per minute (N_p), and the pitch circle diameter of the pinion (d_p) in millimeters.

$$W_t = \frac{60 * P}{\pi d_p N_p} \quad (1)$$

The Overload or Application Factor (K_o): is intended to modify the calculated stress according to the type of service the gear will be subjected to. Some of the pertinent application influences include type of load, type of prime mover, acceleration rates, vibration, shock, and momentary overloads. Application factors are established after considerable field experience has been gained with a particular type of application. The designer should establish the application factors based on past experience with actual designs that have run successfully.

The Dynamic Factor (K_v): dynamic factors are used to account for the inaccuracies in the manufacturing and meshing of gear teeth in action. Some of the effects which produce transmission errors are: vibrations of the tooth during meshing due to the tooth stiffness, inaccuracies produced in the generation of the tooth profile, magnitude of the pitch line velocity of the gear, dynamic unbalance of the rotating members, wear and permanent deformation of contacting portions of the teeth, gear shaft misalignment and the linear and angular deflection of the shaft, and tooth friction. In order to gain some control over these effects, AGMA has defined a set of quality control numbers. The AGMA transmission accuracy level number Q_v can be taken as the same as quality number. The following equations for the dynamic factor are based on these quality numbers and are sourced from AGMA 2001-D04 standard.

$$K_v = \left(\frac{C + \sqrt{200Vt}}{C}\right)^B \quad (2)$$

Where,

$$C = 50 + 56\ (1.0 - B) \text{ for } 5\ Qv \leq 11 \quad (3)$$

$$B = 0.25\ (12 - Qv)\ 0.667$$

V_{tmax}: the maximum pitch line velocity at operating pitch diameter. The maximum recommended pitch line velocity for a given Q_v is determined from the equation below:

$$V_{t\,max} = \frac{[C + (Q_v - 3)]^2}{200} \quad (4)$$

The quality number is assigned based on gear manufacturing techniques and precision level, this often reflects on the area of application in which the gear is to be used.

The Size Factor (K_S): the effects of the influence of non-homogenous materials. The size factor K_S corrects the stress calculation to account for the known fact that larger parts are liable to fail, since it is expected that a large section of material to be weaker than a small section due to the probability of the presence of a "weak link". According to AGMA 2101-D04; the size factor reflects non-uniformity of material properties. It depends primarily on: tooth size, diameter of parts, ratio of tooth size to diameter of part, face width, area of stress pattern, ratio of case depth to tooth size, and hardenability and heat treatment of materials. Standard size factors for gear teeth have not yet been established for cases where there is a detrimental size effect. In such cases, some size factor greater than unity should be used. The size factor may be taken as unity for most gears, provided proper choice steel is made for the size of the part and its heat treatment and hardening process. For the purpose of this study, the size factor will be assumed to possess a value of unity.

The Load Distribution Factor (K_H): intended to account for the effects of possible misalignments in the gear that will cause uneven loading and a magnification of the stress above the uniform case. According to the second rule of thumb which stated that the gear face width should be kept between three and four times the circumferential pitch, a factor larger than 2 is applicable for gears mounted with less than full face contact. The AGMA standards should be consulted if more precision is required.

The Rim Thickness Factor (K_B): The rim thickness factor K_B adjusts the estimated bending stress for the thin-rimmed gear; it's a function of the back ratio m_B (the ratio of the rim thickness below the tooth root, t_R, as compared to the tooth whole depth h_t).

$$m_B = \frac{t_R}{h_t} \quad (5)$$

Where,

$$K_B=1.6\ \mathrm{In}\frac{2.242}{m_B} m_B < 1.2 \quad (6)$$

$$K_B = 1\ m_B \geq 1.2 \quad (7)$$

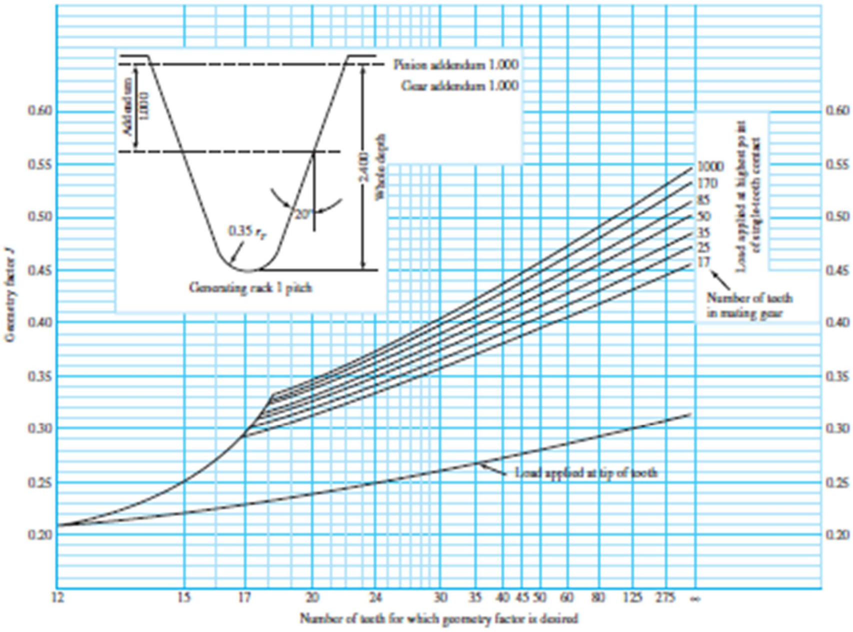


Figure 1. *Geometry factor chart, [24].*

The Geometry Factor (Y_J for bending stress): generated layout of the tooth profile in the normal plane and it is based on the highest of single tooth contact. Most text available on the subject matter including [22, 23] are silent on any formula that might be possibly used to generate Y, which the scholar is expected to determine the geometry factor somehow as a function of the number of teeth on the meshing 20° spur gears as illustrated Geometry factor chart in Figure 2.

The Face Width (b): the tooth measured parallel to the axis of the gear in millimeters (mm).

The Gear Module (m_t): this is the number in millimeters of the pitch circle diameter per tooth. For metric gears (as adopted by most countries of the world), the gear proportions are based on the module.

$$mt=\frac{\text{pitch circle diameter (mm)}}{\text{Number of teeth on gear}} \quad (8)$$

The preferred module values in millimeters are: 0.5, 0.8, 1, 1.25, 1.5, 2.5, 3, 4, 5, 6, 8, 10, 12, 16, 20, 25, 32, 40 and 50. For the purpose of the study, the formula below relating center distance (C), module (m), pinion speed (N_P) and gear speed (N_G).

$$C=\frac{(N_G + N_P)m}{2} \quad (9)$$

The derived module will be compared with available standard modules and the most suitable will be selected and utilized.

Surface Conditioning Factor (ZR): The surface condition factor can be taken as unity (i.e. ZR=1) provided the appropriate surface condition is achieved.

Pitch Diameter of the Pinion (d_{wl}): this is the operating pitch diameter of the pinion in millimeters (mm).

Temperature Factor (Y_θ): for oil or gear-blank temperatures (T) up to 120°C use Y_θ=1. For higher temperatures this factor should be greater than unity, [23], but the procedure for its selection was not stated, hence this study is restricted to gear blank temperatures below 120°C.

$$T \leq 120°C \text{ use } Y_\theta=1 \quad (10)$$

$$T \geq 120°C \text{ use } Y_\theta>1 \quad (11)$$

The Elastic Co-efficient (ZE): according to AGMA 2101-D04; the elastic coefficient, ZE, is defined by the following equation:

$$Z_E = \sqrt{\frac{1}{\pi\left[\left(\frac{1-\upsilon_P^2}{E_P}\right)+\left(\frac{1-\upsilon_G^2}{E_G}\right)\right]}} \quad (12)$$

Geometry Factor for Pitting Resistance (Z_I): pitting resistance can be obtained from the relations given below according to [23, 25]:

$$\frac{\text{Cos}\emptyset\text{tSin }\emptyset\text{t } m_G}{2m_{N*}(m_G+1)} \text{ (for external gears)} \quad (13)$$

$$m_G=\frac{N_P}{N_G}=\frac{d_G}{d_P} \quad (14)$$

Reliability Factor (Y_Z): is used to adjust for desired reliability levels either less or greater than 99 percent, which is the level for the allowable bending strength. If statistical data on the strength distribution of the gear material are in hand, a suitable reliability factor can be selected. In lieu of this, use the values.

AGMA Factor of Safety ($S_{F,H}$): states that ANSI/AGMA standard 2001-C95 and 2101- C95 have introduced safety factor guarding against bending fatigue failure S_F and pitting fatigue failure S_H, [23, 26]. For the purpose of this study, the AGMA factor of safety is assigned a value of 2.5.

Hardness Ratio Factor for Pitting Resistance (Z_W): If the pinion is harder than the gear, uniform surface strength can be obtained by making the pinion harder than the gear [22, 24] or when a surface hardened pinion is mated with athorough-hardened gear. The hardness ratio Z_W has the purpose of adjusting the surface strength for this effect. The values Z_W of are obtained**.** From the equation given below:

$$Z_w=1.0 + A' (mG -1.0) \quad (16)$$

$$\text{Where: } A'=0.00898\left(\frac{H_{BP}}{H_{BG}}\right) - 0.00829 \quad (17)$$

and

$$1.2 \leq \left(\frac{H_{BP}}{H_{BG}}\right) \geq 1.7 \quad (18)$$

for

$$\left(\frac{H_{BP}}{H_{BG}}\right) <1.2, \quad (19)$$

$$\left(\frac{H_{BP}}{H_{BG}}\right) > 1.7, A'=0.00698 \quad (20)$$

The terms H_{BP} and H_{BG} are the Brinell hardness (10mm ball at 3000kg load) of the pinion and gear respectively. It should be noted that while [22, 23, 26]; provides an alternative relation used in determining Zw based on the Rockwell hardness test, this paper have decided to stick with the equations given above.

Allowable Bending Stress (σ_{FP}): For the purpose on this study Allowable Bending Stresses will be examined for only steel gears as unavailability of data on Stress Cycle Factors for other types of material greatly limits the study. The following data on allowable bending stress for steel gears is sourced from the AGMA 2101- D04 standard.

Allowable Contact Stress (σ_{HP}): for the purpose on this study Allowable Contact stresses will be examined for only steel gears as unavailability of data on stress Cycle Factors for other types of material greatly limits the study. The following data on allowable contact stress for steel gears is sourced from the AGMA 2101-D04 standard.

Parameters to be supplied by user (s) of the program: users of the program will be required to supply some useful information. Some of these data include: pitch diameter of the gear and pinion, face width, gear module, power rating in the machine to which the gear is operated, pressure angle of the gear and helix angle (for helical gears) and pinion speed. The parameters listed above are the basic information the user is expected to possess and supply to the computer program. However these parameters are by no means sufficient in themselves in determining the stresses operating at the gear teeth when utilizing AGMA standards. A lot of the parameters in the AGMA equation still do not have standard procedures established yet for determining them, therefore, following the advice in [22, 24]; and various other texts, this work has decided to adopt a value of unity for size factors (K_s) and temperature factors (Y_θ). Dynamic factor (K_v) will be obtained using the equation stated. The user is expected to possess data on gear manufacturing technique and the program will be automated to select a quality number based on the purpose and precision level used to manufacture the gear. Information about the type of material to manufacture of the gear and pinion must be specific and précis as supplied by the user of the program.

3. Results and Discussions

The bending and pitting stress calculator determines the bending and pitting stresses of spur and helical gears. The softwares have been designed and created using both the MATLAB and Visual Studio software development environment also know as GUI.

MATLAB Program Development: Button controls, edit text list boxes are used as basic controls for the interface. Series of scripts and function were bound to the user ininterface controls in order to be able to solve all the parameters that will be used to determine the bending and pitting stresses. VISUAL Studio Program Development**:** is a high-level language program and an interactive environment that can be used for numerical computation, visualization, and programming. With Visual Studio, data can be analyzed, develop algorithms, and create models and applications. The language, tools, and built-in math functions enable users to explore multiple approaches and reach a solution faster than with spreadsheets or traditional programming languages. The minimum input data required on both computer programs are: Power (Watts), Pinion speed (rev/min), Face width (m) and Pinion diameter (m), and a grade number that specifies the AGMA stress constants to be used. The software displays the stress parameters for the gear which comprise the bending and pitting stress; and pitting stress of the pinion.

The efficiency and capability of the software were put into test as follows:

Test 1: (Source: AGMA Gear Bending Stress Solution Manual, 2001)

The software was used to calculate the AGMA bending and contact stresses of a spur pinion with 80 mm diameter and a module of 3.5 mm transmits 16 kW to a gear. The pinion speed is 1300 rev/min, number of teeth of pinion and gear are 28 and 84 respectively and the gears have a 95 mm face width, through-hardened steel at 200 Brinell, uncrowned, manufactured to a No. 10 quality standard, and considered to be of open gearing quality installation. Assume gear constants to be 1.

The test 1 input is as shown in Plate 1a and the result output to this test being displayed by the software as in Plate 1b.

Test 2: (Source: AGMA Gear Bending Stress Solution Manual, 2001)

The AGMA bending and contact stresses of a steel helical gear of a pinion with a module of 1.9mm; diameter of 45 mm. it moves at a speed of 764rev/min. if it has a face width of 50 mm. It transmits power at 7kW. The Gear is through-hardened steel at 200 Brinell, uncrowned, manufactured to a No. 10 quality standard. Assume gear constants to be 1.

The test 2 input is as shown in Plate 2a and the result output to this test being displayed by the software as in Plate 2b.

Test 3: (Source: AGMA Gear Bending Stress Solution Manual, 2001)

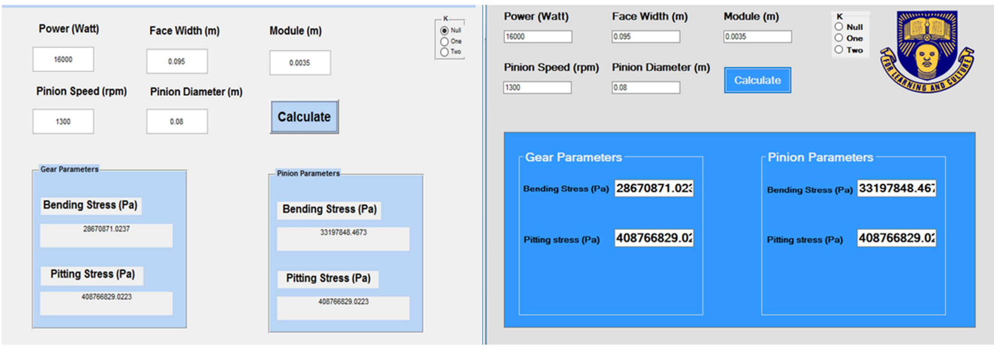
A spur pinion has a speed of 662 rev/min. The number of teeth of pinion and gear are 28 and 84 respectively and the gears have a 33.9mm face width and are through-hardened steel. The pinion has a 90 mm diameter a module of 7.89 mm transmits 7000W to a 80 mm diameter gear. The pinion is a No. 10 quality standard gear, and considered to be of open gearing quality installation. Assume gear constants to be 1

The created software displayed the input test in Plate 3a and the result output in Plate 3b which was used to find the AGMA bending and contact stresses in test3.

Test 4: (Source: AGMA Gear Bending Stress Solution Manual, 2001)

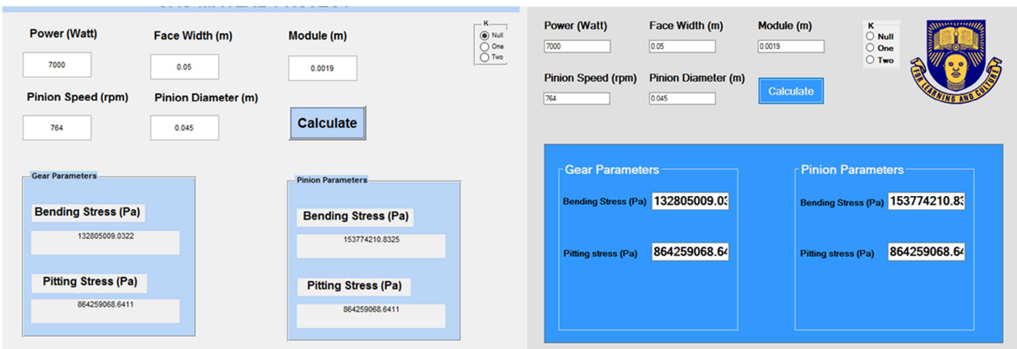
Find the AGMA bending and contact stresses of a steel helical gear having pinion with a module of 3.5mm; diameter of 78 mm. it moves at a speed of 1200rev/min. if it has a face width of 75 mm. It transmits power at 1.2kW. The Gear is through-hardened steel at 200 Brinell, uncrowned, manufactured to a No. 10 quality standard. Assume gear constants to be 1.

The test 4input is as shown in Plate 4a and the result output to this test being displayed by the software as in Plate 4b.



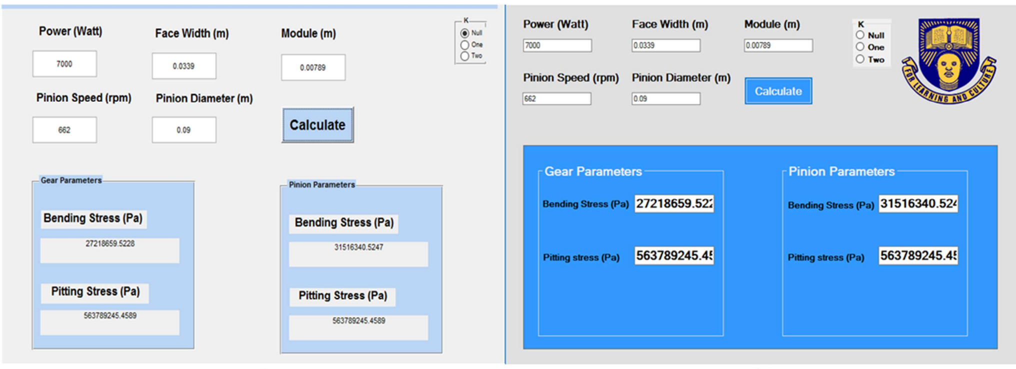
(a) (b)

Figure 2. *Test 1input and Result output from bending and pitting stress calculations on MATLAB / Visual Studio.*



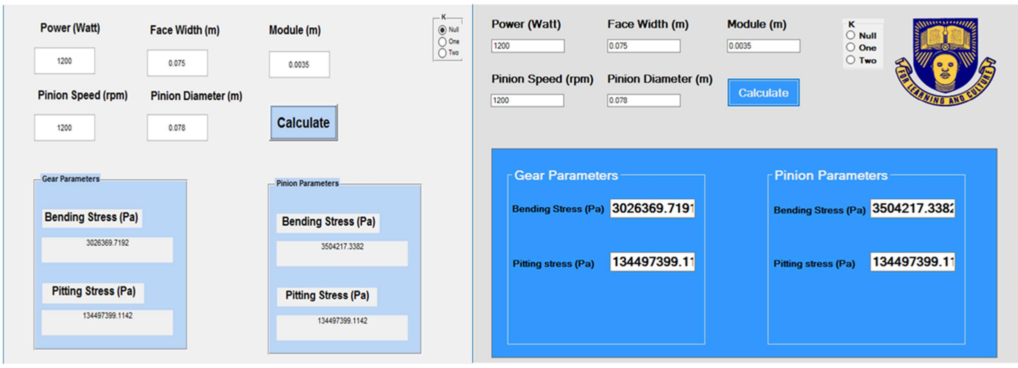
(a) (b)

Figure 3. *Test 2 input and Result output from bending and pitting stress calculations on MATLAB/ Visual Studio.*



(a) (b)

Figure 4. *Test 3 input and Result output from bending and pitting stress calculations on MATLAB/ Visual Studio.*



(a) (b)

Figure 5. *Test 4 input and Result output from bending and pitting stress calculations on MATLAB/ Visual Studio.*

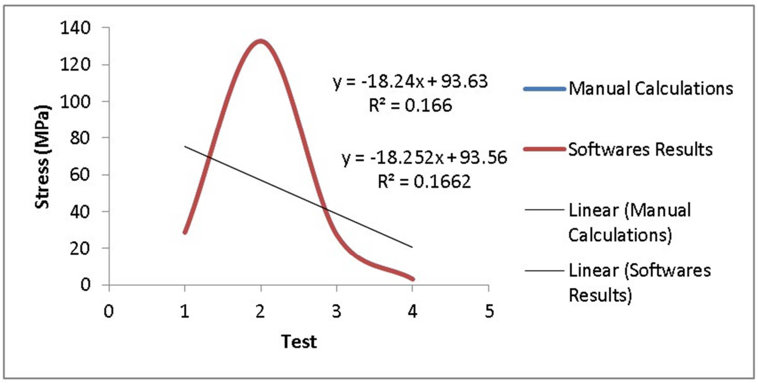


Figure 6. *Bending Stress of Gear using manual and software computed result.*

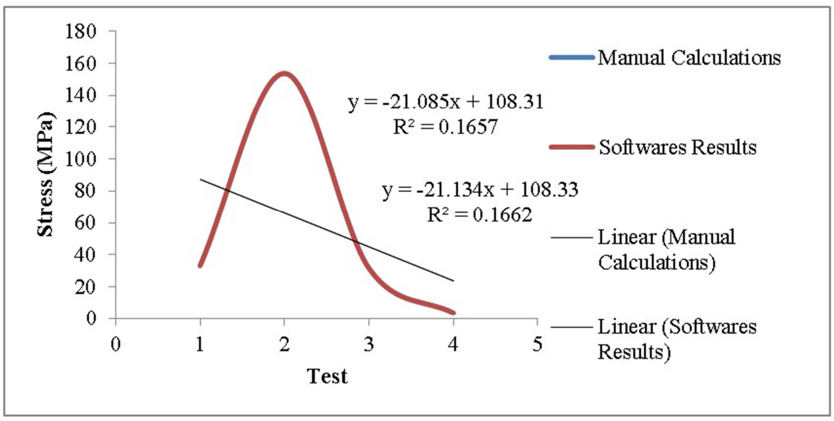


Figure 7. *Bending Stress of Pinion using manual and software computed result.*

Table 1. Results Obtained From Bending and Pitting Stress Calculations.

	Manual Calculations	Softwares Results
1. Bending Stress for Gear (MPa)	28.72	28.6708
Bending Stress for Pinion (MPa)	33.24	33.1978
Pitting Stress for Gear (MPa)	408.87	408.7668
Pitting Stress for Pinion (MPa)	408.88	408.7668
2. Bending Stress for Gear (MPa)	132.91	132.8050
Bending Stress for Pinion (MPa)	153.8	153.7742
Pitting Stress for Gear (MPa)	864.88	864.2590
Pitting Stress for Pinion (MPa)	864.9	864.2590
3. Bending Stress for Gear (MPa)	27.4	27.2186
Bending Stress for Pinion (MPa)	31.69	31.5163
Pitting Stress for Gear (MPa)	563.81	563.7892
Pitting Stress for Pinion (MPa)	564.01	563.7892
4. Bending Stress for Gear (MPa)	3.09	3.0263
Bending Stress for Pinion (MPa)	3.66	3.5042
Pitting Stress for Gear (MPa)	135.01	134.4973
Pitting Stress for Pinion (MPa)	135.03	134.4973

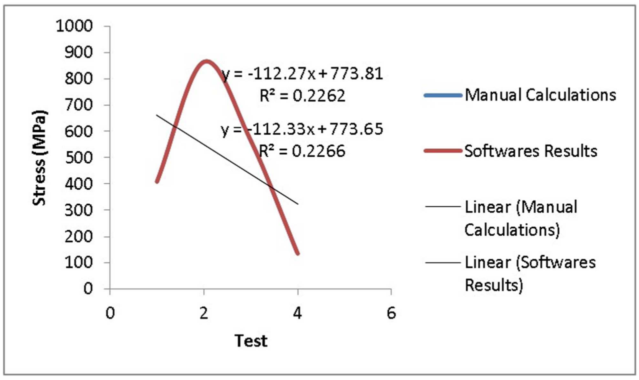


***Figure 8.** Pitting Stress of Gear using manual and software computed result.*

Note: Non-linear line indicates that the test data was sampled randomly without any linear relationship, and - From the graph, the difference between the manual and software computed result can be monitored from the equation of the trendline and its corresponding R-sqaure.

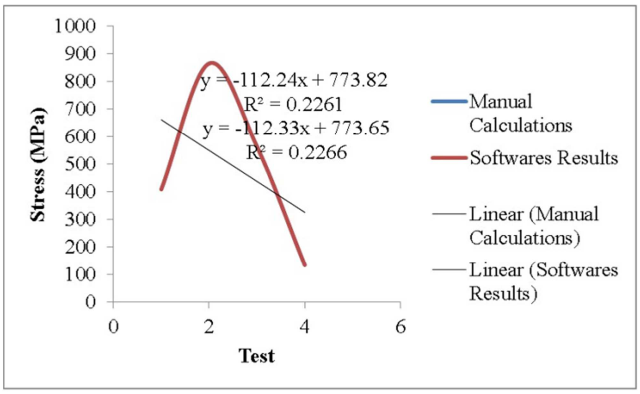


***Figure 9.** Pitting Stress of Gear using manual and software computed result.*

In Figures 2, 3, 4 and 5, non-linear line indicates that the test data was sampled randomly without any linear relationship. From the graphs (figures 6, 7, 8 and 9) the difference between the manual and software computed result can be monitored from the equation of the trendline and its corresponding R-sqaure.

Discussion of the stress calculator output result: from the output results shown in Table 1, both software (s) showed slight deviations in the values of the bending stress for the gears and the bending stress for the pinion and a high level of accuracy.

$$\frac{\sum \text{values from manual calculation}}{\sum \text{values from Software(s) Calculation}} = \frac{4359.9}{4356.3378} = 1.000817705$$

Assuming the data obtained from manual calculations are standard, then the accuracy of the value of the results calculated from both software (s)

$$=100 - 1.000817705 = 98.9991822948\%$$

From the above results, both the MATLAB and Visual Studio Programs showed no degree of deviation from the values of the pitting stress for both the gear and the pinion. Both results had a 100% level of accuracy as compared to the manual calculation for the pitting stress on both gear and pinion. However, the value of the bending stress for the gear and pinion calculated from both software (s) showed 98.9991822948% degree of accuracy and 1.000817705% deviation from the value obtained from manual calculations.

4. Conclusions

Accurate results were achieved when determining the bending and pitting stresses of spur and helical gears by the computer programs developed. This software will save machine designers a lot of time when calculating the stress on gears and the user do not need to know any or have any programming skills or knowledge of how to create a graphical user interface. These gear stress calculation software have been created and they have the capability of handling a variety of different gear problems. The results obtained from both the MATLAB and Visual Studio based virtual tool were very accurate. It is strongly recommended for gear designers, lecturers and the like to use this software whenever dealing with the bending and pitting stresses of spur and helical gears which are the basic gears in operation. Determination of the bending and pitting stresses of bevel and worm gears should be further developed on the application. Also similar applications and graphical user interfaces should be created on other computer software. The two applications were developed on the windows operating system but it can also be developed on other computer operating systems like *Linux, Mac, Os, and android* to make it more accessible to a wider range of users.

Conflict of Interest

The authors declare that they have no competing interests.

Nomenclature

t_R	rim thickness below the tooth, (mm),
h_t	whole depth, (mm),
Z_E	elastic coefficient, (N/mm)
μ_p and μ_G	poisson's ratio for pinion and gear,
Ep and E_G	modulus of elasticity for pinion and gear, N/mm^2.
$\emptyset_t$	transverse pressure angle, (deg),
N_G	speed of the gear, (rpm),
N_p	speed of the pinion, (rpm),
d_G	pitch circle diameter of the gear, (mm),
d_p	pitch circle diameter of the pinion, (mm),
mG	speed ratio, (Constant),
N	number of stress cycles, (Constant),
L	life,(hours),
ω	speed,(rpm),
q	number of contacts per revolution,(rpm).

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