Applied Mechanics and Materials Vols. 465-466 (2014) pp 1347-1351

doi:10.4028/www.scientific.net/AMM.465-466.1347

Maximum Torque of Combinations threat for spur gear based on AGMA standard

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Keywords: Torque, Spur Gear, AGMA, Safety factor

Abstract. This study is an approach to investigate the transformations curve of gearing safety. Two types of failure can occur on a gear namely the bending stress and the surface pitting failure. There are many standards for gearing in used worldwide such as AGMA, JGMA, ISO, DIN, etc. but the focus of this study will be on AGMA standard. The main data for this study is the torque value applied which can be distinguished into causing bending strength or surface pitting failure. The Autodesk Inventor spur gear component accelerator was used as a tool for data acquisition based on the AGMA standard of calculations. Two gear materials with high value of allowable contact stress compared to its allowable bending stress was chosen for the study as they are predicted to have a transformation curve from surface durability to bending strength when its torque values are plotted against number of teeth. This is then repeated on various gear modules for both materials to obtain a series of combination curves useful in determining the maximum torque that can be applied on the spur gear before failures occur either by bending stress or surface pitting depending on the curve.

Introduction

Gears are defined as toothed wheels or multi-lobed cams which transmit power and motion from one shaft to another by means of successive engagement of teeth [1]. They are many types of gears and spur gear, among other types such as helical gear, internal gear and Herringbone gear are categorized as the parallel axes type of gear. Just like gear, there are also many standards on gearing in used worldwide but this study will focus on the AGMA standard of gearing. AGMA or The American Gear Manufacturers Association is accredited by the American National Standards Institute (ANSI) to write all U.S standards on gearing. A series of meshing gear or rather a gear drive system is most commonly used as a transmission component in various types of machinery or automotive applications to either increase/decrease the desired torque or speed. In developing countries like Malaysia, the gear industry focuses more on gear fabrication/manufacturing rather than on the design of the gear itself. Modern gear design is very much influenced and based on the manufacturing processes of gear teeth such as the generating and forming process.

The purpose of this study is to determine the maximum allowable torque within defined safety limits applied on spur gears before failing due to occurring threats such as bending or contact stress. It is important because safety is among the parameter that include in sustainable product. The prerequisite for manufacturers to survive in the competitive market is the ability to cope with the needs of sustainable development [2]. Tooth breakage can be the result of a fatigue mechanism or an overload which exceeds the gear tooth fracture strength Destructive fatigue pitting is a result of repeated stress cycling of the tooth surface beyond the material's endurance limit [3]. The result of the study will be presented in an easy-to-understand manner via relevant charts and graphs to provide an overall view for the user.

Methodology

The threat that will cost the failure for spur gear consists of bending stress and contact stress. The calculations for these stresses will be based on ANSI/AGMA 2101-D04:2005 -

Fundamental rating factors and calculation methods for involute spur and helical gear teeth (Metric Edition). The basic equations are: [4]

$\sigma = W^t K_o K_v K_s \frac{1}{Fm_t} \frac{K_m K_B}{J}$ (AGMA gear bending stress equation in S.I unit)

$\sigma_{allowable} = \frac{S_t}{S_F} \frac{Y_N}{Y_\theta Y_Z}$ (AGMA gear bending endurance strength equation in S.I unit)

$S_F = \frac{\frac{S_t Y_N}{Y_\theta Y_Z}}{\sigma}$ (AGMA bending factor of safety)

$\sigma_c = C_p \sqrt{\left(W^t K_o K_v K_s \frac{K_m}{d_p F} \frac{C_f}{I} \right)}$ (AGMA gear contact stress/pitting equation in S.I unit)

$\sigma_{c,allowable} = \frac{S_c}{S_H} \frac{Z_N Z_W}{Y_\theta Y_Z}$ (AGMA gear contact endurance strength equation)

$S_H = \frac{\frac{S_c Z_N Z_W}{Y_\theta Y_Z}}{\sigma_c}$ (AGMA wear factor of safety)

Where ; $W_t = \frac{60000H}{\pi dn}$ (transmitted load, KN), H = power, KW,
V = pitch line velocity, mm/s
K_o = Overload factor, K_v = Dynamic factor, K_s = Size factor, m = Tranverse metric module
F = Facewidth, mm, K_m = load distribution factor, K_B = Rim Thickness factor
J = Geometry factor for bending strength, Y_θ = Temperature factor
S_t = Gear bending strength, MPa(value depends on gear materials)
Y_N = Stress cycle factor for bending stress, Y_Z = Reliability factor
S_F = AGMA bending factor of safety (normally 1.5 minimum), C_p = elastic coefficient, $\sqrt{N/mm^2}$
C_f = Surface condition factor, used $C_f = 1$, d_p = Pitch diameter of pinion, mm
I = Geometry factor for pitting resistance, Z_N = Stress cycle life factor
S_c = contact fatique strength, Mpa(value depends on gear material)
Z_W = Hardness ratio factors for pitting resistance, S_H = AGMA factor of safety for wear (normally 1.5)

If the torque acting on the gear exceed a certain value (allowable torque), it will cost the gear to fail either by its bending strength or surface durability depending on which threat having a lower value of allowable torque. This allowable torque value in Nm is obtained with the aid of the **Autodesk Inventor Spur gear component accelerator** design tool software for the purpose of this study utilizing the same AGMA standard and equations stated above. The software is able to calculate dimensions and check strength of external and internal gearing with straight and helical teeth.

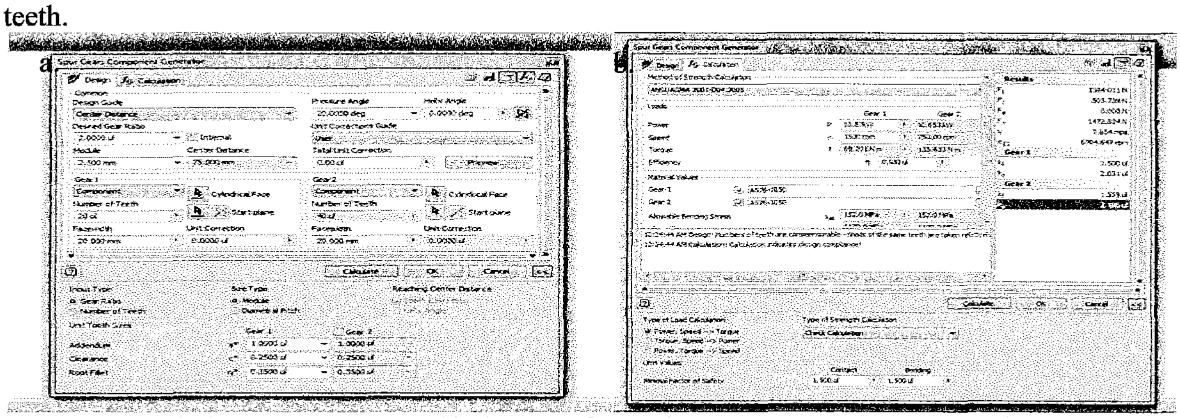

Figure 1: (a) Spur gear generator (design interface), (b) Spur gear generator (Result interface)

The value of allowable torque obtained from the software is recorded and then recalculated after adjusting several gear design parameters which are the variables for this study. The variable parameters are gear modules, gear ratio and no of teeth. The fixed parameters which are set in the software for this study are rotation speed of gear (rpm = 1500), safety factor for bending and surface durability (1.5), pressure angle for the gear teeth (20°) and gear materials (A576-1050 carbon structured steel and A322-5135 alloy structured steel). Charts are then plotted based on these allowable torque values versus the relevant parameters before analyzed.

Results and discussion

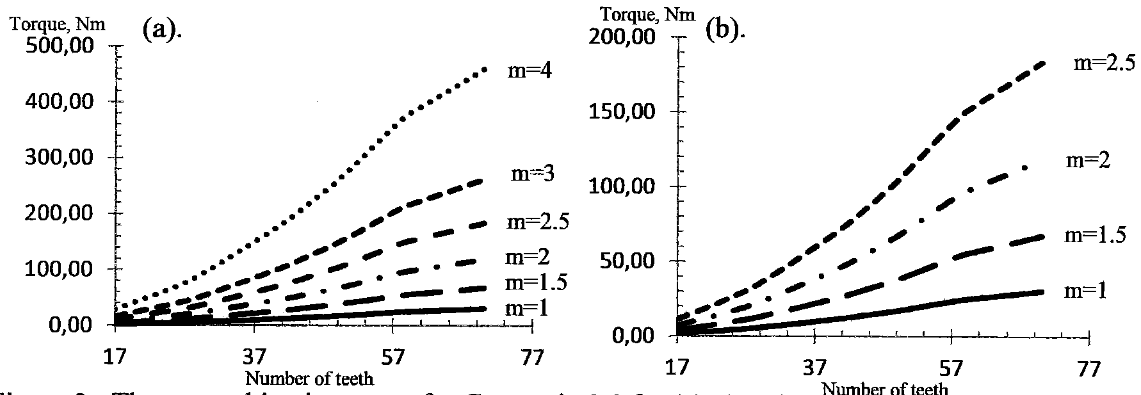

Figure 3: Threat combination curve for Gear ratio 2.0 for (a). A576-1050 Carbon Structured Steel, (b) A322-5135 Alloy Structured Steel

From **Figure 3(a)**, we can conclude that for **A576-3050 Carbon Structured Steel**, value of maximum torque increases with gear modules and number of teeth. Module is a measure of the gear tooth size. With larger module and the same number of teeth, the gear tooth is larger and thicker thus having better bending strength. The maximum torque for bending strength will increase linearly (straight line) as shown in **Figure 5** and this increment is affected by the strength geometry factor, J which an increase of this factor will reduce the bending stress. For surface durability, the increment in maximum torque is exponentially up. At lower number of teeth, the contact is more thus making its maximum torque value lower than bending strength. This will happen up to a point where the maximum torque for surface durability will overtake its value for bending strength. (**Figure 5(a)** shows that happening at no. of teeth 58) A maximum torque curve for threat combination can then be obtained by using the surface durability curve (with the lower torque value) up to the intersection point and combine it with the bending strength line (which has the lower torque value from the intersection point onwards) The result is shown in **Figure 7** (for module 2.5) and **Figure 3(a)** (for all modules).

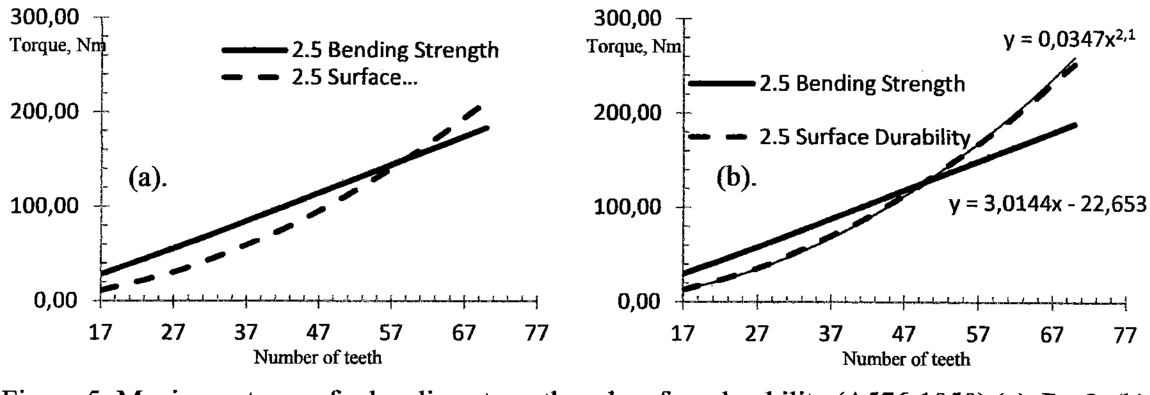

Figure 5: Maximum torque for bending strength and surface durability (A576-1050) (a). R= 2, (b). R=4

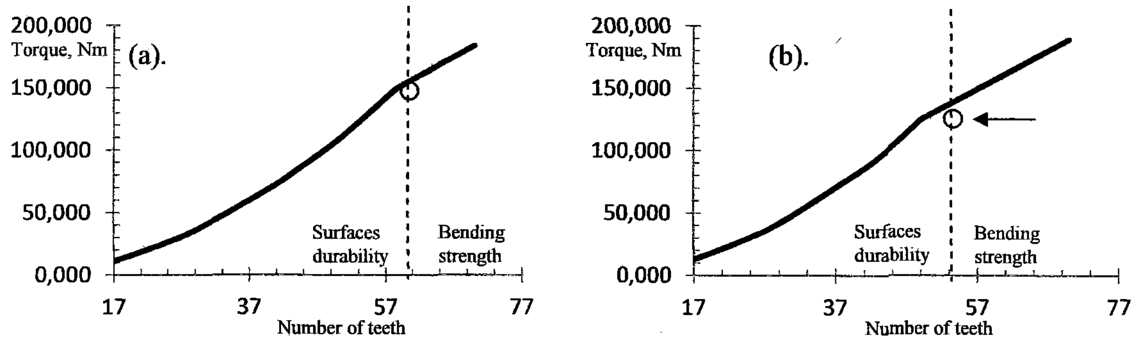

Figure 7: Maximum torque for module 2.5 when bending strength and surface durability are combined. (a). R = 2, (b). R = 4 (transformation at lower no. of teeth and higher maximum torque)

Figure 3(b) shows the same Threat combination curve but for **A322-5135 Alloy Structured Steel**. Both set of graphs (figure 3a & 3b) show similarities in terms of values and curves because the same phenomenon has occurred to A322-5135 regardless of material. It must also be mentioned here that both A576-1050 and A322-5135 steel chosen for this study has the same heat treatment which is tooth face hardened and the same allowable contact and bending stress value at 1170MPa and 152MPa respectively. A322-5135 Alloy steel has a Brinell hardness number, H_B ranging from 179-217 while A576-1050's Brinell hardness number, H_B is 197. Material hardness is proportional to its allowable contact stress. With the same allowable contact stress value, both materials are thought to have the same hardness for this study and thus having the same combination curve showing similar value for maximum torque. An increase in gear ratio will increase the value of maximum torque slightly with surface durability overtaking maximum torque value of bending strength at a lower no. of teeth as shown in **figure 5b**. (58 for gear ratio 2.0 and 49 for gear ratio 4.0 for both materials) **Figure 7b** shows the same threat combination curve of A576-1050 for module 2.5 but for gear ratio 4.0 with higher maximum torque and lower no. of teeth as mentioned. With the threat combination curves for gear ratio 2.0 and 4.0, number of teeth ranging from 17 to 70, module 1.0 to 4.0 for A576-1050 and module 1.0 to 2.5 for A322-5135 obtained for both materials, the safe torque applied on the gear within the mentioned parameter can be estimated before any failure occurs. For specific threat to the gear teeth, a series of equations can be derived from each set of graph for every module available as shown in **figure 5b**. By using these equations, we can also estimate the maximum torque of the gear by referring to either bending strength or surface durability.

Conclusion

It can be concluded from the findings that the value for maximum torque will increase as the number of teeth and gear modules increase for both materials. Increasing the gear ratio will further increase the value of maximum torque. Initially at lower number of teeth, the torque value for surface durability is lower than bending strength but will be higher as number of teeth increases. Because of this phenomenon, another set of graph was obtained by initially taking the lower value of maximum torque (surface durability) and then the bending strength (which has the lower value of torque) after the two lines intersect. Surface hardness is a main factor for this phenomenon and both materials is thought to have the same Brinell hardness number and allowable contact stress. One significant application from the graphs of this study is the estimation of maximum torque applied on a spur gear within the parameters as a preventive measure against its tooth failures.

References

[1]. Dr. S.S Wadhwa, Er. S.S.Jolly. Machine Design, A basic approach. New Delhi: Dhanpat Rai & Co, 2007, Page 546

[2]. Chee Kiong Sia, Loo Yee Lee, Siaw Hua Chong, Mohd Azwir Azlan & Nik Hisyamudin Muhd Nor. "Decision Making with the Analytical Hierarchy Process (AHP) for Material Selection in Screw Manufacturing for Minimizing Environmental Impacts." *Applied Mechanics and Materials* 315 (2013): 57-62.

[3]. Lynwander, Peter. Gear drive systems, Design and Application. New York : Marcel Dekker, Inc, 1983. Page 388 & 391. ISBN 0-8247-1896-8 .

[4]. Shigley, J.E Mischke, C.R & Budynas, R.G., (2010), "Mechanical Engineeering Design", Ninth Edition, McGraw Hill.