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RE: Week 9 Lab Assignment – Gear Stress Lab

Introduction:

This lab compares hand calculations from Shigley's Mechanical Engineering Design Textbook and Fusion 360 Finite Element Analysis (FEA) to evaluate contact pressure and root stresses in meshing spur gears. Hand calculations use simplified formulas based on mechanics and empirical data, while FEA provides a detailed numerical model for higher resolution (depending on mesh size) stress predictions. The comparison aims to assess the reliability and accuracy of these two methods in gear design. Understanding their differences helps determine the best approach for stress analysis in gear systems.

Results:

FEA Analysis

The Gears were modeling and simulated in Fusion 360 as per the instructions in the "Gear lab.pdf" (Doc 1) and "Week 8 – Using Fusion to find Stress in Gear Teeth.pdf" (Doc 2) documents provided on Canvas. The gears were modeled in Fusion 360 with the spur gear add-in per the parameters defined in Doc 2. The gears materials were defined as Steel AISI 1020 Annealed with a yield strength of 30 kpsi. The completed CAD model is shown below.

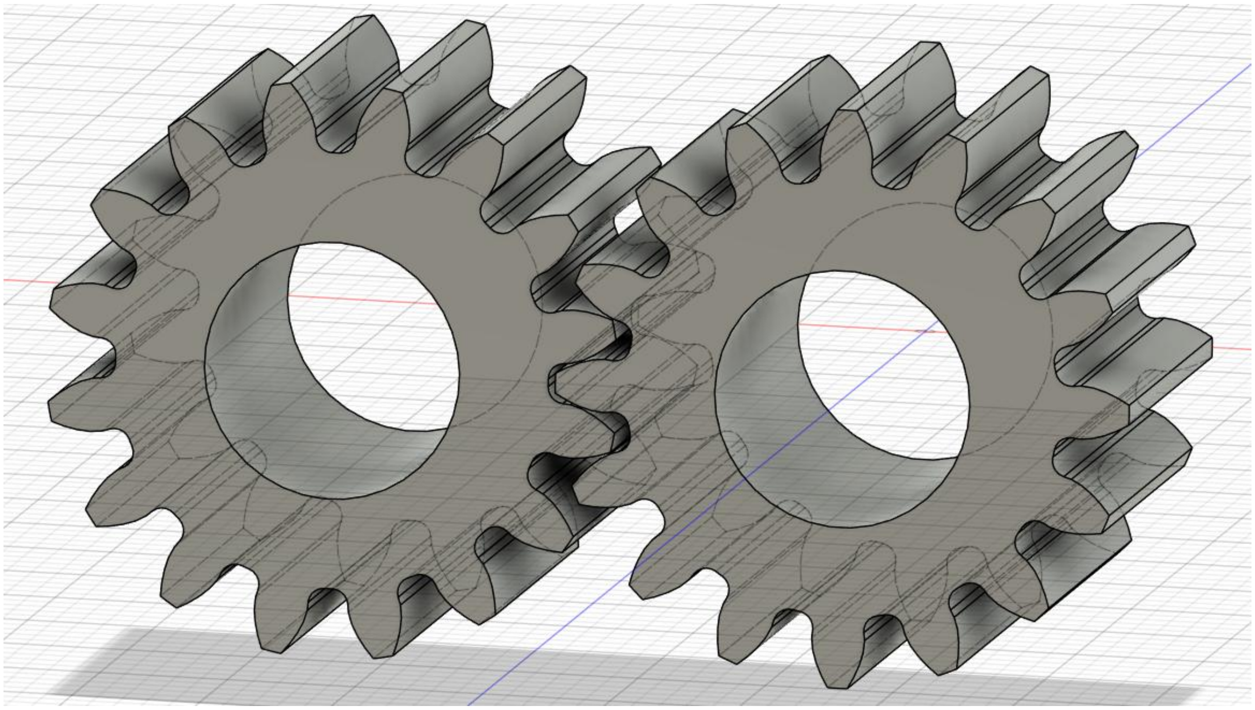

Figure 1. Fusion 360 Model Gear Pair

To create the Fusion 360 FEA static study, we followed the instruction in Doc 2. The tangential load on the gear is 365 lbf and found in Doc 1. Below is the FEA with everything prior to running the simulation.

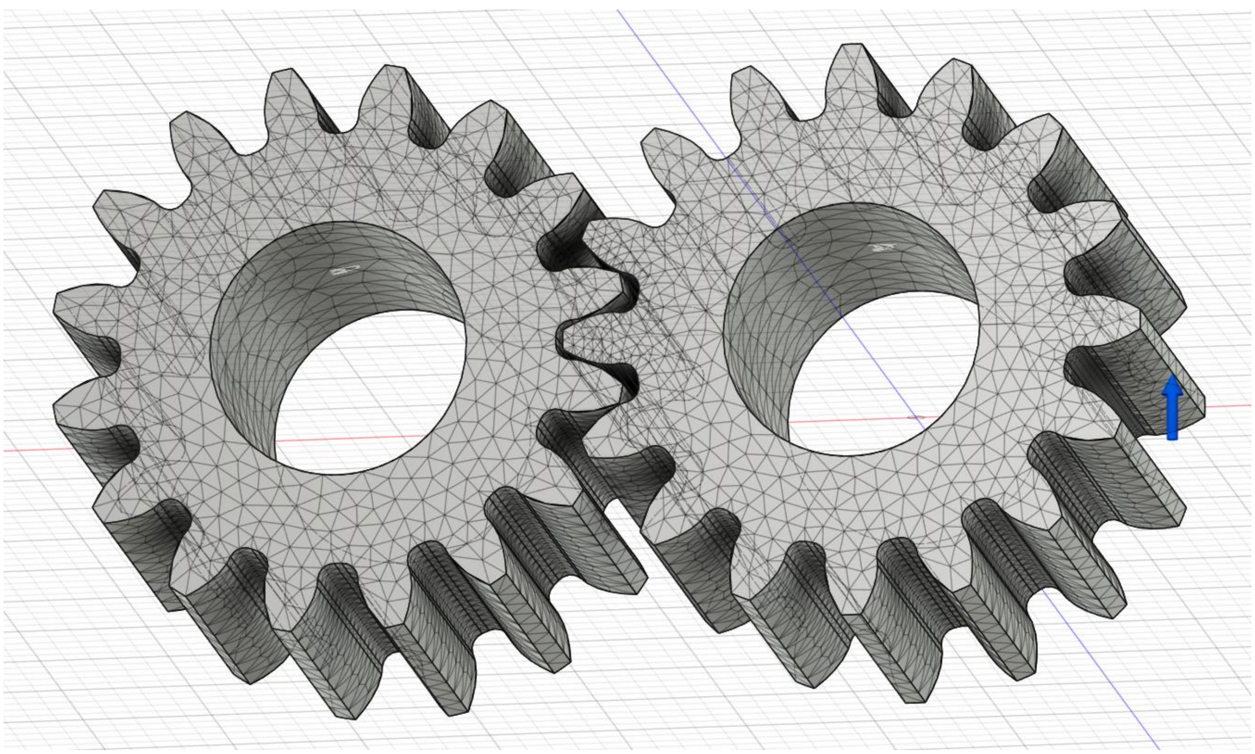

Figure 2. Fusion 360 FEA Gear Pair with Constraints, Contacts, Load, Global and Local Mesh

The FEA was then run. The critical results are shown below.

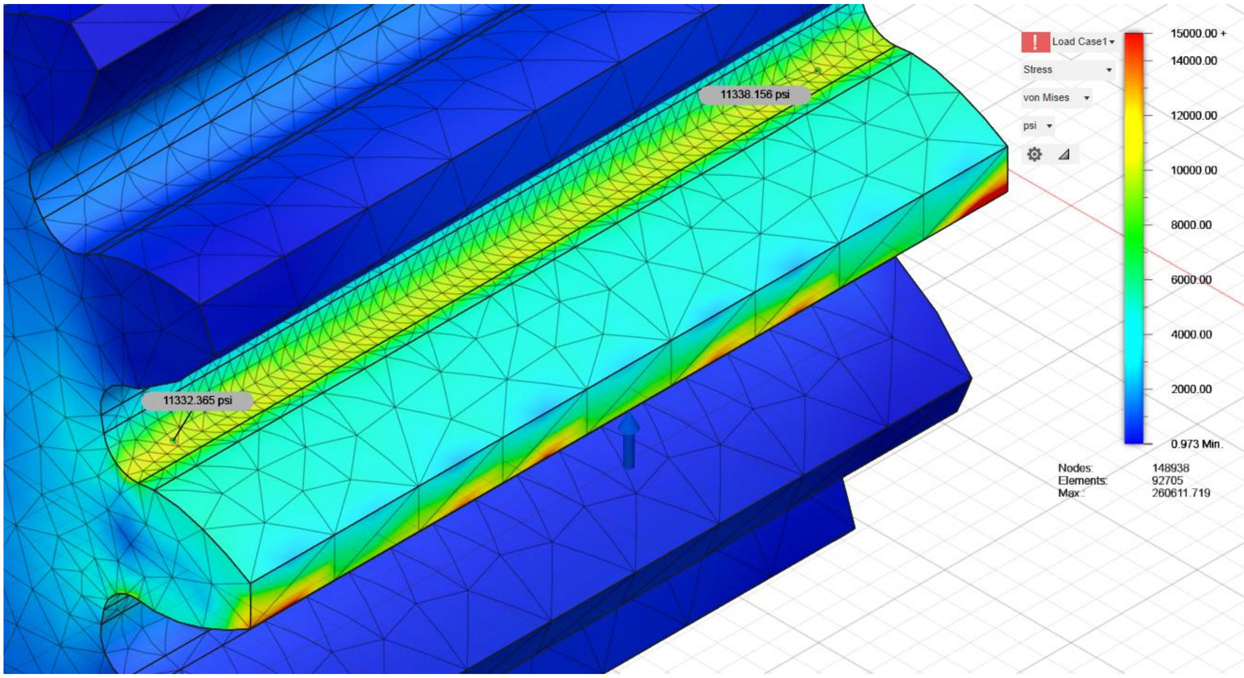

Figure 3. Fusion 360 FEA Gear Tangentially Loaded Tooth Root Stress

The maximum stress in the root of the tangentially loaded (365 lbf) gear tooth is 11.34 kpsi.

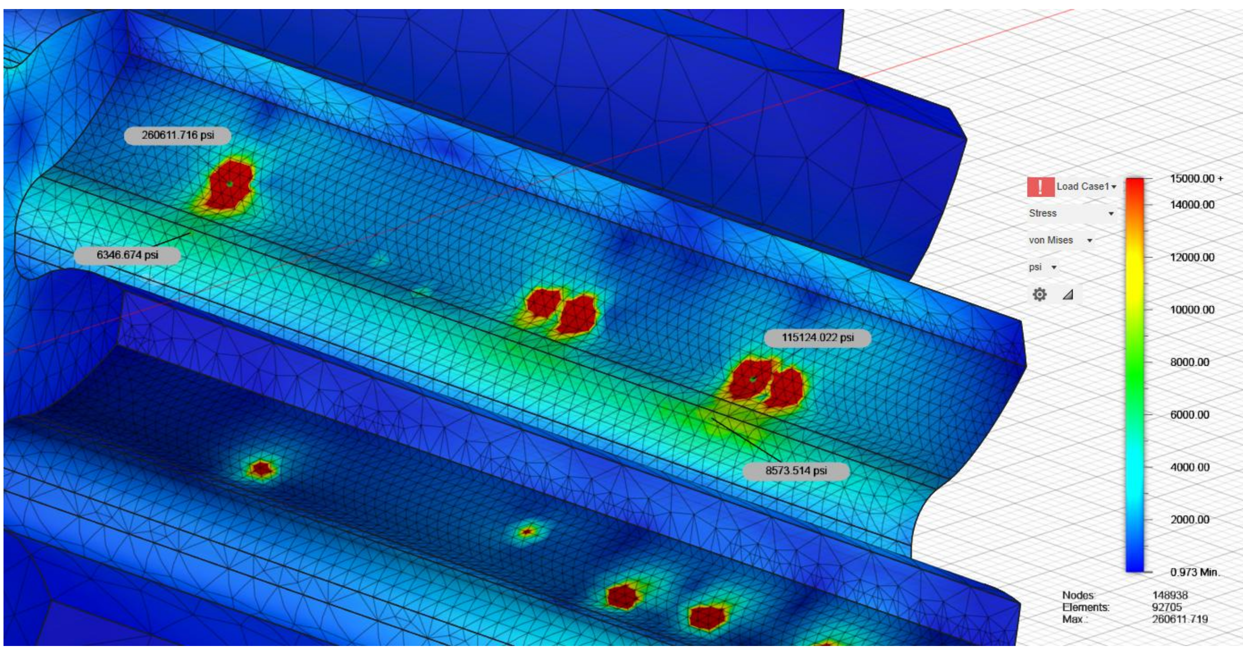

Figure 4. Fusion 360 FEA Gear Tooth in Mesh Root and Contact Stress

The maximum contact stress between the two gears is 260.61 kpsi and the maximum root stress is 8.57 kpsi. It is worth noting that this stress only occurs if the tangential load is applied while the other gear remains fixed as this is how the simulation was set up.

Hand Calculations Using Microsoft Excel

Given the original specifications of the gears shown in Figure 5 and assuming a safety factor, $n = 3$, the tangential load, W_t, and the output power, H, can be calculated for the given gears.

INPUTS	**Units:**	English
Problem Givens		
Diametrial Pitch, P (N/d)	8	teeth/in
Face Length, F	1.5	in
# teeth, N	16	teeth
Pressure Angle, ϕ	20	deg
Safety Factor, n	3	-
Rotational Speed, n	1200	rev/min
Rotational Speed, ω	125.66371	rad/s
Material Properties		
Material	AISI 1020 steel as-rolled	
Manufacturing Process (for K_v)	Cut or Milled Profile	
Tensile Strength, S_{ut}	55.000E+03	psi
	55	kpsi
Yield Strength, S_y	30E+03	psi
	30	kpsi

Figure 5. Gear Specifications and Input Parameters

Based on the input specifications in which both gears in mesh are the same dimensions Figure 6 shows the tangential load, W_t, and the output power, H, can be calculated.

OUTPUTS		
Output Power While Designing to Prevent Yeild		
Allowable Sress, $\sigma_{allowable}$	10.000E+03	psi
	10.00	kpsi
Lewis Form factor, Y (Table 14-2)	0.296	-
pitch diameter, d	2	in
Pitch Line Velocity, V	125.664	in/s
	628.319	ft/min
Velocity factor K_v	1.5236	ft/min
Tangent Load per Power Out, W_t/H	52.52113	lbf/HP
Output Power, H	6.93567	HP
Tangential Load, W_t	364.26913	lbf
	364,269.130	*10^-3 lbf

Figure 6. Calculated Tangential Load and Output Power

Based on the given safety factor of 3 and the yield stress of the material the allowable stress was calculated to be 10 kpsi. From the number of teeth, the Lewis form factor was found through Table 14-2 in "Shigley's Mechanical Engineering Design". The pitch diameter was calculated through the given diametral pitch and the number of teeth to find the pitch line velocity. The pitch line velocity was used to calculate the velocity factor, K. Finally, the tangential load was calculated in terms of the output power. Setting the Lewis bending stress equal to the allowable stress based on the given safety factor the expected output power was calculated to be $H = 6.94\ HP$. Recalculating tangential load based on this output power results in $W_t = 364.27\ lbf$.

The FEA run assumed the tangential loading was $W_t = 365\ lbf$ therefore recalculating the Lewis Bending Stress based on the given loading instead of the given safety factor results in Figure 7.

Lewis Stress Given Tangential Loading W_t		
Tangential Load, W_t	365.00000	lbf
	365,000.000	*10^-3 lbf
Output Power, H	6.94958	HP
Lewis Bending Stress, σ_{bend}	10,020.064	psi
	10.020	kpsi
Lewis Bending Stress, σ_{bend} w/ SCF	12.826E+03	psi
	12.826	kpsi
Calculating Stress Concentration Factor (SCF)		
fillet radius r	0.063	in
Circular pitch, p	0.39270	in
x	0.0555	in
l	0.28125	in
t	0.249874969	in
r/d ~r/t =	0.252126095	-
D/d ~p/t =	1.571582314	-
K_t	1.4	-
q	0.7	-
SCF K_f	1.28	-

Figure 7. Lewis Bending Stress Calculation for Given Loading

The Lewis Bending Stress for a tangential load $W_t = 365\ lbf$ is $\sigma_{bend} = 10.02\ kpsi$.

Using A-15-6 to find K_t, shown in Figure 8, and 6-26 to find q, shown in Figure 9 both graphs from "Shigley's Mechanical Engineering Design", the stress concentration factor $K_f = 1.28$.

Figure A–15–6

Rectangular filleted bar in bending. $\sigma_0 = Mc/I$, where $c = d/2$, $I = td^3/12$, t is the thickness.

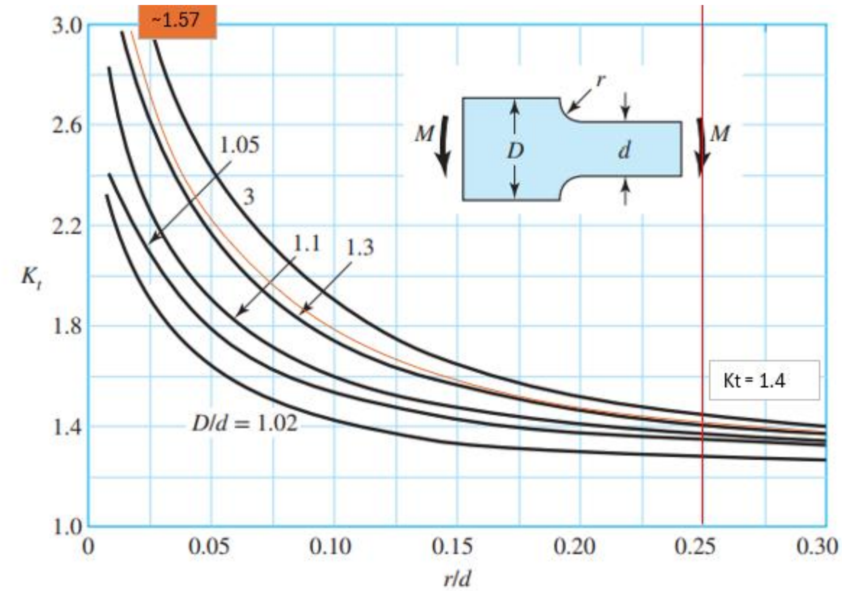

Figure 8. Finding Stress Concentration Factor K_t

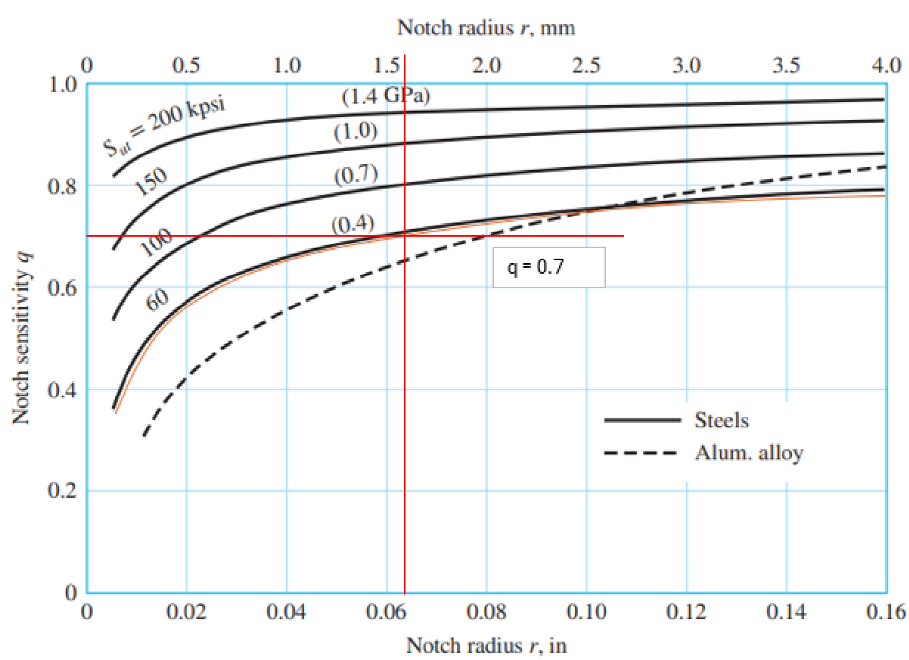

Figure 6–26

Notch-sensitivity charts for steels and UNS A92024-T wrought aluminum alloys subjected to reversed bending or reversed axial loads. For larger notch radii, use the values of q corresponding to the $r = 0.16$-in (4-mm) ordinate. Source: Sines, George and Waisman, J. L. (eds.), *Metal Fatigue,* McGraw-Hill, New York, 1969.

Figure 9. Finding Notch Sensitivity Factor q

Applying this stress concentration factor to the Lewis Bending Stress previously calculated results in a bending stress of $\sigma_{bend} = 12.826\ kpsi$.

Comparing both these bending stresses to the FEA's measured stress values in Figure 10.

COMPARING TO MEASURED FEA STRESS						
Bending Stress at Root based on Wt = 365 lbf without SCF				**Bending Stress at Root based on Wt = 365 lbf with SCF**		
Lewis Bending Stress, σ_{bend}	10,020.064	psi		Lewis Bending Stress, σ_{bend} w/ SCF	12,825.682	psi
	10.020	kpsi			12.826	kpsi
Root Stress Tangentially Loaded Gear	11,338.156	psi		Root Stress Tangentially Loaded Gear	11,338.156	psi
	11.338	kpsi			11.338	kpsi
% diff. Tangentially Loaded Gear	13.155%	-		% diff. Tangentially Loaded Gear	-11.598%	-
Root Stress Mesh Gear	8,573.514	psi		Root Stress Mesh Gear	8,573.514	psi
	8.574	kpsi			8.574	kpsi
% diff. Mesh Gear	-14.437%	-		% diff. Mesh Gear	-33.154%	-

Figure 10. Comparison of Hand Calculations and FEA Stress Results

The root stress of the tangentially loaded gear was 13.2% different and the root stress of the meshed gear was -14.4% different when compared to the Lewis bending stress without the stress concentration factor applied. The root stress of the tangentially loaded gear was -11.6% different and the root stress of the meshed gear was -33.2% different when compared to the Lewis bending stress with the stress concentration factor applied. Therefore, applying the stress concentration factor provides a more conservative estimate of the bending stress at the root assuming that the FEA is representative of what occurs in real gears in which a physical test would need to be performed to verify the FEA values.

Conclusion:

The comparison between hand calculations using Shigley's Mechanical Engineering Design textbook and Fusion 360 FEA highlights key differences in stress estimation for meshing spur gears. The Lewis bending stress calculation, when adjusted for stress concentration, provides a more conservative estimate than FEA with the percent differences between the two methods, ranging from -11.6% to -33.2% this suggests that while hand calculations offer a reliable first approximation, FEA provides more detailed insight into localized stress distributions. However, verifying FEA results with physical testing is crucial to ensuring accurate calculations in real-world applications. For example, in the aviation industry, the FAA requires physical testing to be performed to verify a FEA models accuracy for type certificated aircrafts.