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Experimental and Theoretical Study of Radial Force Component Effects on Spur Gear Teeth Design to Resist Failure

Malik N. Hawas *, Doaa Fadhil Kareem

Department of Mechanical Power Engineering, Al Mussaib Technical Collage, Al Furat Al Awsat Technical University, Najaf, Iraq.

Keywords:

Lewis equation; Longitudinal vibration; Spur gear design; Tooth face width.

Highlights:

- *The radial component force is important if it has a significant value.*
- *The radial component force may cause longitudinal vibration along the spur gear teeth.*
- *The dynamic force caused longitudinal vibration due to the gear rotation.*

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**Corresponding author:*

Malik N. Hawas

Department of Mechanical Power Engineering, Al Mussaib Technical Collage, Al Furat Al Awsat Technical University, Najaf, Iraq.

Abstract: The Lewis equation in gear design is always used to determine spur gear tooth dimensions, i.e., face width tooth (b), based on the pushing force tangential component during engagement between gear and pinion. The radial pushing component force was neglected due to its insignificant value. The present study depends on the high radial component force along a tooth caused by the longitudinal vibration. The tooth was regarded as a cantilever. The Timoshenko equation was used to produce dynamic force. Then, the dynamic pushing component of tooth face width in the second case can be calculated. All the results were obtained from the Auto desk inventor program in the case of the Lewis equation depending on machine power. The Timoshenko equation for longitudinal vibration with different gear rotation speeds showed the same results as the theoretical ones. An inverse relationship was found between the speed increase and the tooth face width. The strain gauge technique, receiver, and transmitter (wireless) were experimentally used to record the dynamic strain readings and calculate (b). The importance of longitudinal vibration in tooth gear design in avoiding failure was highlighted.

دراسة تجريبية ونظرية لتأثيرات مكونات القوة الشعاعية على تصميم أسنان التروس المسننة لمقاومة الفشل

مالك نعمة حواس، دعاء فاضل كريم
قسم هندسة القوى الميكانيكية/ كلية المسيب التقنية/ جامعة الفرات الأوسط التقنية/ النجف – العراق.

الخلاصة
تعتمد معادلة لويس في تصميم التروس دائما في إنتاج قيم أبعاد سن الترس المهماز وخاصة عرض سن الوجه (b) وفقا للمكون العرضي لقوة الدفع أثناء الاشتباك بين الترس والترس مع إهمال مركبة قوة الدفع الشعاعي لقلة قيمتها. يعتمد البحث الحالي على مركبة القوة القطرية في حالة القيمة العالية بسبب الاهتزاز الطولي الذي كان على طول السن بالنسبة للناتئ. تم استخدام معادلة تيموشينكو لإنتاج قيمة القوة الديناميكية ومن ثم يمكن حساب مركبة الدفع الديناميكي لعرض وجه السن في الحالة الثانية. تمت محاكاة جميع النتائج بواسطة برنامج Auto Desk Inventor في حالة معادلة لويس بالاعتماد على قوة الآلة، وكذلك معادلة تيموشينكو للاهتزاز الطولي مع اختلاف سرعة دوران التروس، تعطي نفس اتجاه النتائج النظرية مع إظهار علاقة عكسية بين زيادة السرعة وعرض الوجه للترس. في العمل التجريبي استخدمت تقنية قياس الانفعال مع جهاز الاستقبال والارسال (لاسلكي) لتسجيل قراءات الانفعال الديناميكي ومن ثم حساب عرض الوجه عملياً. تمت الاشارة إلى أهمية الاهتزاز الطولي في تصميم التروس المسننة لتجنب فشلها.

الكلمات الدالة: معادلة لويس، الاهتزاز الطولي، تصميم الترس العدل، عرض وجه السن.

1.INTRODUCTION

The chatted design of mechanical elements, such as spur gear, is key due to the high request for efficiency, reliability, the capacity of load carrying, and service life. The goal of the present research is to review and collect practical examples of important points and guidelines of gear design, including dimensions and lubricant liquid. The supported tools to perform engaging two gears are important in gear design. The group of tooth profiles, pressure angles, suitable lubricant fluid, quality grade, and clearances is needed to get backlash [1-3]. The common faults in gearboxes are tooth surface wear, increasing fatigue force on the gearbox, and loss of its stable properties and service life. The gear mish characteristics highly vary with wear posted intensities. The sample of wear faults regarding tooth profile deformation, friction, and mish stiffness over time was considered to study the effect of vibration on tooth fatigue wear (TFW). The adjustment of friction coefficient and tooth profile were applied in (TFW). The simulation was conducted under multiple wear intensities, which became a single-state parallel to the gearbox according to the dynamic differential equation and under multiple torque conditions. The dynamic simulation's results converged with experimental results of [4-6]. The present research develops a process of two spur gears engaging with the involute tooth. The spur gears were selected from a lathe machine gearbox. Under the same conditions, the Lewis formula was used in stress analysis and the finite element approach. The stresses are combined as concentration stress, especially at the tooth's curvatures. The model and size of the spur gear tooth are important parameters due to their effect on stress analysis [7-9]. Spur gear design includes new predictions, such as shipwreck median, contact label as translating third body, the form factor of contact, a service load factor, the shipwreck median rule, the physical properties, and geometry properties of the contact label. The contact label is a body in transmission during a gear teeth clash. The factor of contact label was used to compare the influence of different pressure angle standards. The service load factor represents the effect of different agreement-altered load modifications. The analysis of gear design is discrete in design sizing and verification. The formula of design sizing and design verification is concocted and antedated in simplified forms for the Hertz contact and the Lewis root bending stresses [10, 11]. The research scientific methodology includes describing meshing procedures. Then, the procedures were applied to the spur gear model using stress analysis in the ABAQUS program. The finite element process was applied between the spur gear and pinion. The cross-section included a front spur gear tooth and tooth face width. The engaging process between the spur gear and pinion was performed using a CAD program. These processes save time and stress in spur design calculations. Also, these processes improve the design and perform it during operation [12, 13]. The common defects of gear transmission systems are tooth fractures and cracks. The fracture mechanics theory was used to investigate crack propagation paths. The software ADAMS was used to establish multibody dynamics in the gear system and simulate vibration singles of cracked gear with different torques. The root mean square method in statistics was used to limit the damage degree. Instantaneous energy was used to analyze the fault features of the gear under crack and tooth fracture [14-16].

2.METHODOLOGY

2.1.The Force Analysis of Spur Gear Tooth

The spur gear tooth is considered a beam of free limb and fixing from another limb. The tooth strength can be found using the Lewis equation, which yields suitable ability tooth loading results. The loading transfers from one tooth to another; therefore, the analysis of forces and stresses was applied to one tooth. The load was the same on the other teeth in the spur gear. When the contact occurred between the big gear and pinion, the load was assumed to be at the end of the driven gear tooth. The load was at the end of the driving gear tooth at contact stoppings. The tooth was considered a cantilever beam, and the natural force subjected to it, as shown in Fig. 1. This force is analyzed into two components: tangential force (W_T) and radial force (W_R), acting vertically and parallel to the center line of the tooth, respectively [17].

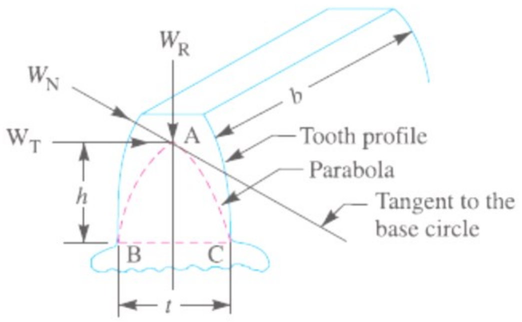


Fig. 1 Analysis of Gear Tooth.

The tangential component produces bending stress, breaking the tooth. This component has a high value and is a key variable in the Lewis equation. The radial component produces compression stress on the tooth, iterating every full revolution of the spur gear. The radial force is relatively small; however, it leads to a longitudinal vibration on the tooth root and generates compression stress at the root [18]. The tooth of the spur gear at each revolution is subjected to a radial force. This force is small; however, longitudinal vibration produces stress on the tooth root. The tooth root cross-section area that impacts the longitudinal vibration equation for Timoshenko is (b×t), where b and t are constants. The static stress for gear material is also constant. The stress transmission from the effect point through engaging to the root is (x), representing the stress path through tooth height (t) while changing the rotation velocity to a linear velocity by active pitch circle diameter parameter for the gear. This parameter is important due to its effect on increasing the radial force. The static strain is a gear property of gear material and stress transmission velocity from the tip to root (BC) of the tooth, as illustrated in the following equations [19,20]:

$$F+(\delta F/\delta x)dx=F+\rho.A.dx(\delta^2u/\delta t^2) \quad \textbf{(1)}$$

$$F = A.\sigma=A.E.\varepsilon =A.E. (\delta u/\delta x) \quad \textbf{(2)}$$

The radial component of the active force on the spur gear tooth is due to longitudinal vibration that acts from the free tip to the tooth root, causing stresses and strains that translate to the tooth root with functions of (x) and (t). Eqs. from 3 to 24 to derive the dynamic load equation used to spur gear design.

$$(\delta^2u/\delta x^2) = (1/a^2).(\delta^2u/\delta t^2) \quad \textbf{(3)}$$

$$U(x,t)=\sum_{i=1,2,3,4..}^{\infty} U_i(t).T_i(t) \quad \textbf{(4)}$$

$$T_i(t) = A_i \cos w_i t + B_i \sin w_i t \quad \textbf{(5)}$$

$$U(x,t)=\sum_{i=1,2,3}^{\infty} U_i(x)\left[A_i \cos w_i t + B_i \sin w_i t\right] \quad \textbf{(6)}$$

$$U(x)= (-wi2\ /a2\ ) .Ui(x) \quad \textbf{(7)}$$

$$U_i(x) = C_i \cos{(w_i/a)x}+ D_i \sin{(w_i/a)x} \quad \textbf{(8)}$$

$$U(x,t)=i=\sum_{i=1,2,3}^{\infty}\left[A_i \cos w_i t + B_i \sin w_i t\right].\left[C_i \cos(\frac{w_i}{a}) + D_i \sin(\frac{w_i}{a})\right].x \quad \textbf{(9)}$$

Applying the displacement conditions yields:

$$U(0,t)=\sum_{i=1,2,3}^{\infty}(A_i \cos w_i t + B_i \sin w_i t).C_i = 0 \quad \textbf{(10)}$$

$$U(x,t)=\sum_{i=1,2,3}^{\infty}(A_i^- \cos w_i t + B_i^- \sin w_i t).D.\sin(w_i / a)x \quad \textbf{(11)}$$

$$\frac{\partial u(x,t)}{\partial x}=\sum_{i=1,3,5}^{\infty}(\frac{w_i}{a})\left[\bar{A}_i \cos w_i t + \bar{B} \sin w_i t\right].\cos(\frac{w_i}{a}\ t) \quad \textbf{(12)}$$

$$\cos(\frac{w_i}{a}.\ell) = 0 \quad \textbf{(13)}$$

$$\frac{u}{x} = \frac{\varepsilon_\circ}{\ell}.\ell = \varepsilon_\circ \quad \textbf{(14)}$$

$$u(x,0) = \varepsilon_0 \cdot x \quad \textbf{(15)}$$

$$\dot{u} = V_0 \tag{16}$$

$$u(x,0) = \sum_{i=1,3,5}^{\infty} \bar{A}\sin(\frac{i\Pi}{2\ell}.x) = \in_0 .x \tag{17}$$

$$A_i = \frac{8\in_0 \ell}{\Pi^2\ell^2}(-1)^{\frac{i-1}{2}} \tag{18}$$

$$\frac{\partial u(x,0)}{\partial t} = \sum_{i=1,3,5}^{\infty}(\frac{i\Pi a}{2\ell})\bar{B}_i\int_0^{\ell}\sin(\frac{i\Pi x}{2\ell}).\sin(\frac{j\Pi x}{2\ell})dx. \tag{19}$$

$$\bar{B}_i = V_{0.}\frac{8\ell}{\Pi^2\ell^2 a} \tag{20}$$

$$U(x,t) = \sum_{i=1,3,5}^{\infty}\sin(\frac{i\Pi x}{2\ell}).\frac{8\ell}{\Pi^2 i^2}\left[\in_0 (-1)^{\frac{i-1}{2}}\cos(\frac{i\Pi a}{2\ell})t + \frac{V_0}{a}\sin(\frac{i\Pi a}{2\ell})t\right] \tag{21}$$

$$\frac{U(x,t)}{U_0} = 8\sum_{i=1,3,5}^{\infty}(-1)^{\frac{i-1}{2}}\frac{1}{\Pi^2 i^2}\left[(-1)^{\frac{i-1}{2}}\cos(\frac{i\Pi a}{2\ell})t + (\frac{V_0}{\in_0 a})\sin(\frac{i\Pi a}{2\ell})t\right] \tag{22}$$

Hook's law can be written as the ratio of dynamic stress to static stress:

$$\frac{\sigma_d}{\sigma_{ult.}} = 4\sum_{i=1,3,5}^{\infty}\frac{1}{i\Pi}\left[(-1)^{\frac{i-1}{2}}\cos(\frac{i\Pi a}{2\ell})t + \frac{V_0}{\in_0 a}\sin(\frac{i\Pi a}{2\ell})t\right] \tag{23}$$

$$W_d = 4 * A_{root} * \sigma_{o\ ult.}\sum_{i=1,3,5}^{\infty} cos\frac{i\pi x}{2l}.\frac{1}{i\pi}[(-1)^{\frac{i-1}{2}}.\cos\left(\frac{i\pi a}{2l}\right)t + \frac{V_o}{\varepsilon_o.a}.\sin\left(\frac{i\pi a}{2l}\right)t] \tag{24}$$

The pinion's mechanical properties are known, such as density and modulus of elasticity; therefore, the stress speed along the height of the gear tooth can be calculated as($a = \sqrt{\frac{E}{\rho}} = \sqrt{\frac{105\frac{GN}{m^2}}{780\frac{Kg}{m^3}}}$) for a static strain of (0.004) at pinion different rotation speeds of (600, 1500, and 3000 rpm) and circular pitch diameter of (120mm). Therefore, the liner speed for the tooth movement becomes (6.28, 9.2, and 11.3 m/sec). The Quick Basic program was used with counter (I = 99), tooth height (t, h = 50mm), tooth root width (t = 60mm), and tooth face width (b = 100mm). The results are shown in Figs. 2, 3, and 4.

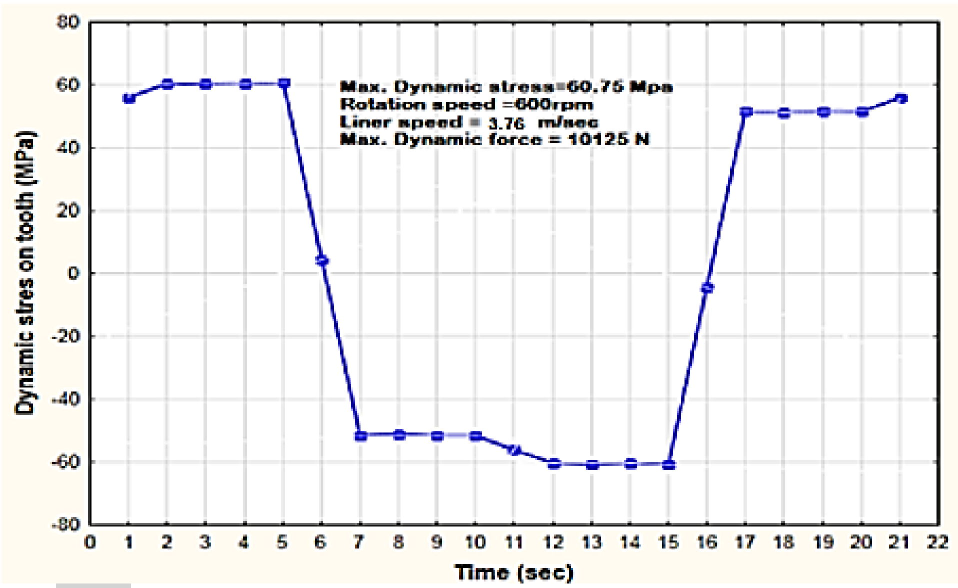


Fig. 2 One Resonance Cycle Rotation of Spur Gear Tooth at 3.76 m/sec.

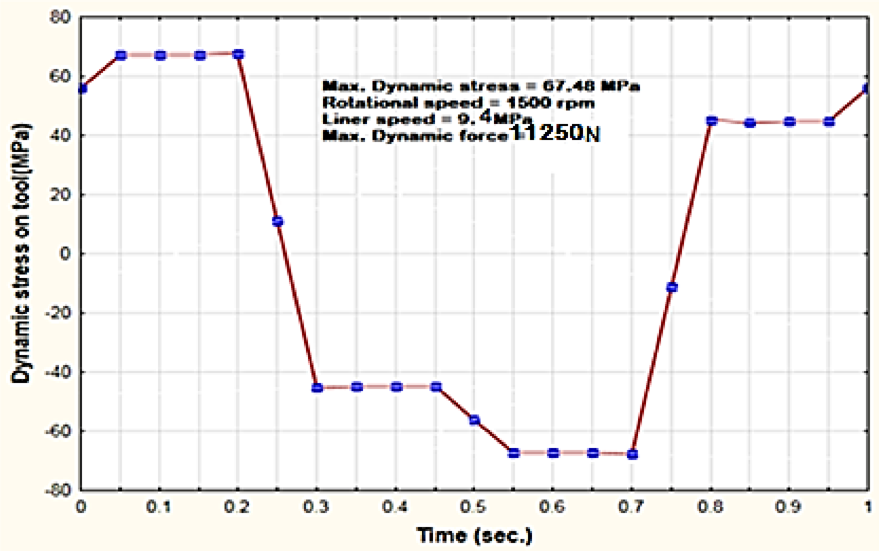


Fig. 3 One Resonance Cycle Rotation of Spur Gear Tooth at 9.4 m/sec.

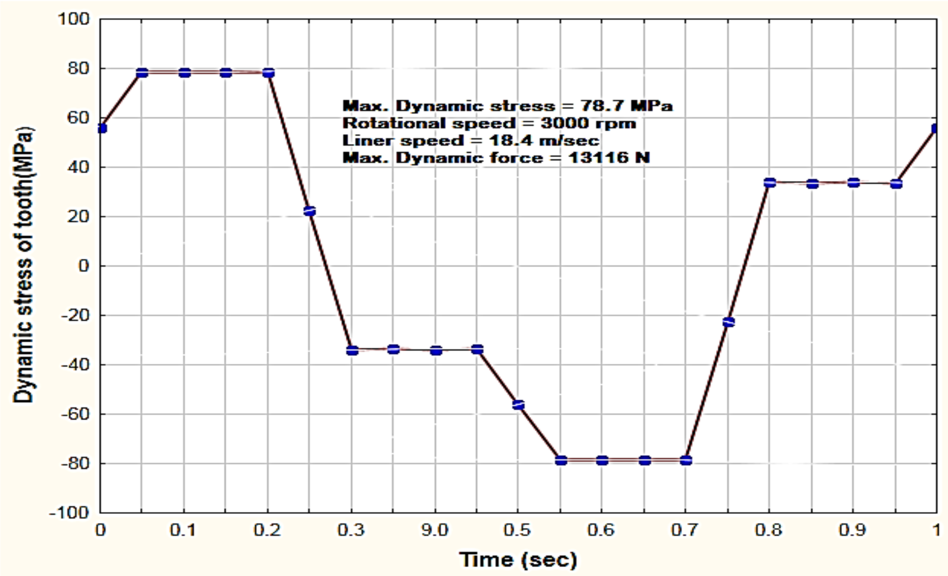


Fig. 4 One Resonance Cycle Rotation of Spur Gear Tooth at 18.4 m/sec.

$$W_N = \frac{W_T}{cos\emptyset} \qquad (25)$$

$$W_T = \frac{P}{v} * C_s \qquad (26)$$

$$T = \frac{P * 60}{2 * \pi N_p} \qquad (27)$$

$$W_T = (\sigma_o \, . C_v) b . \pi . m . y \qquad (28)$$

$$W_r = W_N \, . \sin \emptyset \qquad (29)$$

$$W_d = W_T + W_I = W_T + \frac{21v\,(b.C + W_T)}{21v + \sqrt{(b.C + W_T}} \qquad (30)$$

The material of the two spur gears engaged was cast iron ordinary, which has allowable static stress (σ_o = 84 MPa). The speed factor depends on the linear velocity of the gear rotation based on machine power of(20kW). The pinion pitch circle diameter was (120 mm) and had (16 teeth), while the gear had (48 teeth). Therefore, the module of engaging was (8). The tangential force can be calculated using the Lewis equation, and tooth face width can be determined. All these steps were performed according to the previous equation, i.e., of tangential vibration in the tooth (Eq. 1) as pulse motion produces a dynamic force (Eq. 30) assuming (b = 100mm) and (C= 228) according to information in machine design reference by KHURMI [21]. Then, the tangential force was calculated, and the Lewis equation was applied to determine the tooth face width. These results are illustrated in Table 1.

Table 1 Theoretical Results of Power and Longitudinal Vibration Force.

Type of force	V=3.76m/sec		V= 9.4m/sec		V = 18.4m/sec	
	W_t (N)	b (mm)	W_t (N)	b (mm)	W_t (N)	b (mm)
Machine power equation	5319	60	2174	45	1087	30
Longitudinal vibration equation	6750	75.3	7275	52	7310	36

3.2.Simulation Work

The Static simulation is used (Auto desk inventor program), the forces with the same values of the power equation, and the longitudinal vibration equation were used to calculate the static stress work at the tooth root. Then, these values were applied in the Lewis equation with the same data to determine the tooth face width (b) and compare the results with previous values, as listed in Table 1. The simulation results of the stresses and strain are shown in Figs. 5-10 and Table 2.

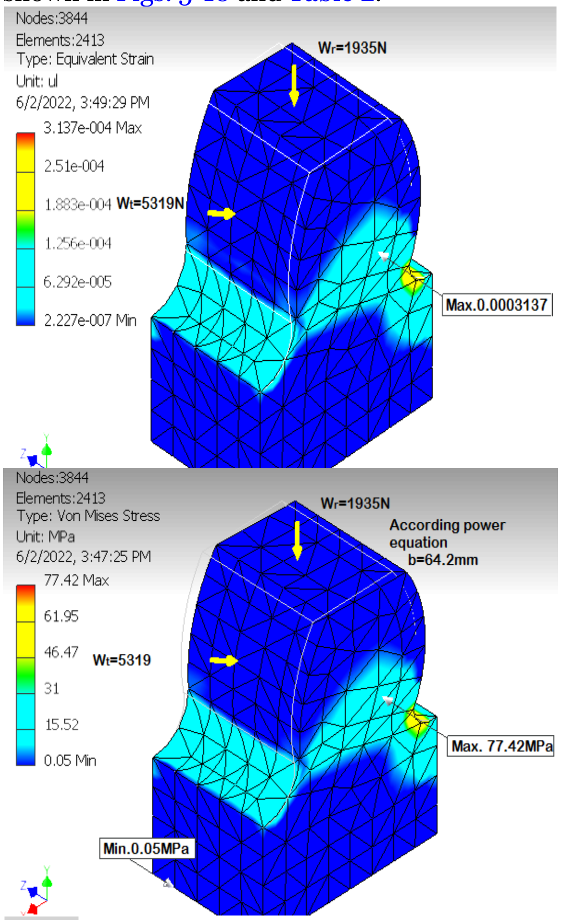
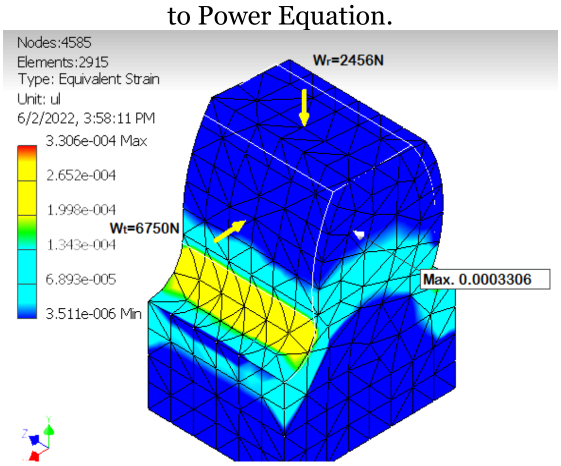


Fig. 5 Simulation Analysis of Stress and Strain on Spur Gear Tooth at V=3.76m/sec According to Power Equation.



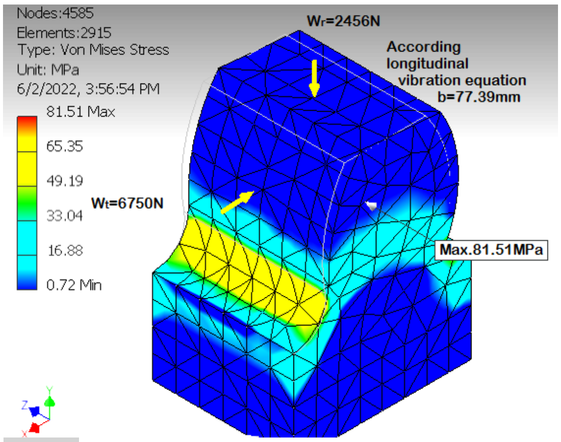


Fig. 6 Simulation Analysis of Stress and Strain on Spur Gear Tooth at V=3.76m/sec According to Longitudinal Vibration Equation.

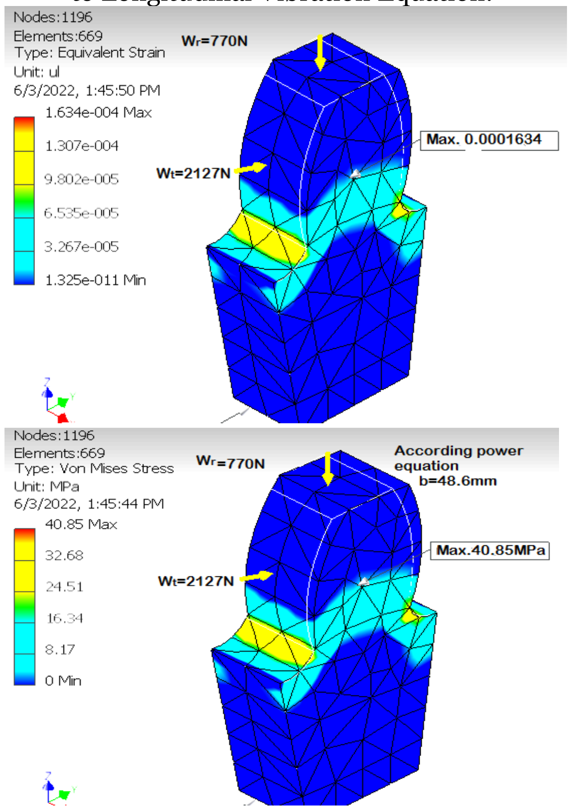
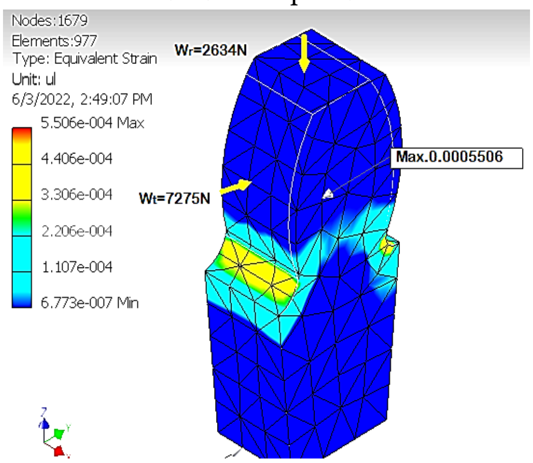


Fig. 7 Simulation Analysis of Stress and Strain on Spur Gear Tooth at V=9.4m/sec According to Power Equation.



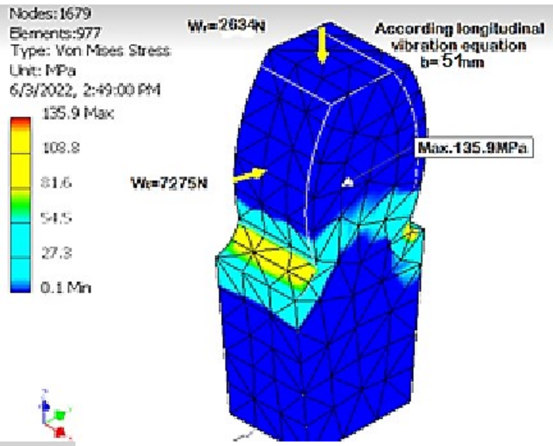


Fig. 8 Simulation Analysis of Stress and Strain on Spur Gear Tooth at V=9.4m/sec According to Longitudinal Vibration Equation.

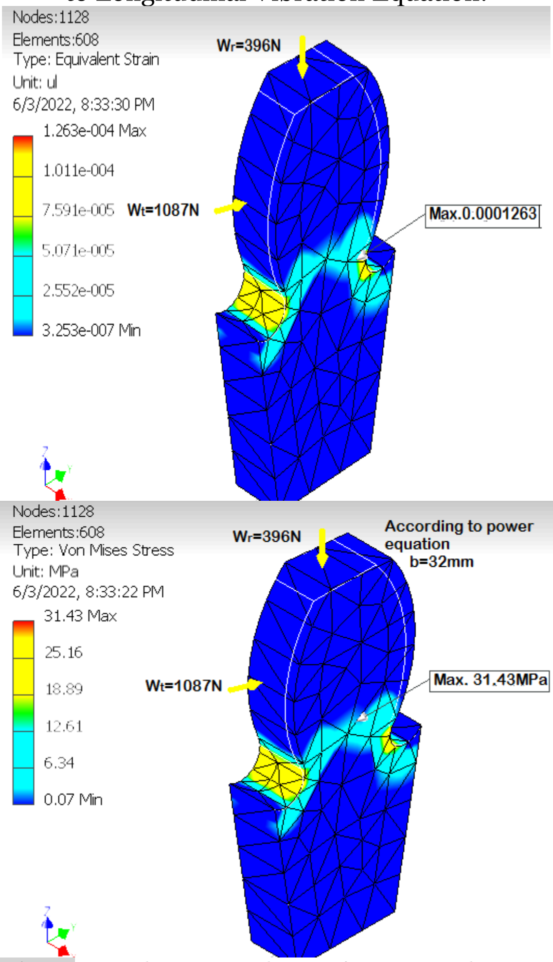


Fig. 9 Simulation Analysis of Stress and Strain on Spur Gear Tooth at V=18.4m/sec According to Power Equation

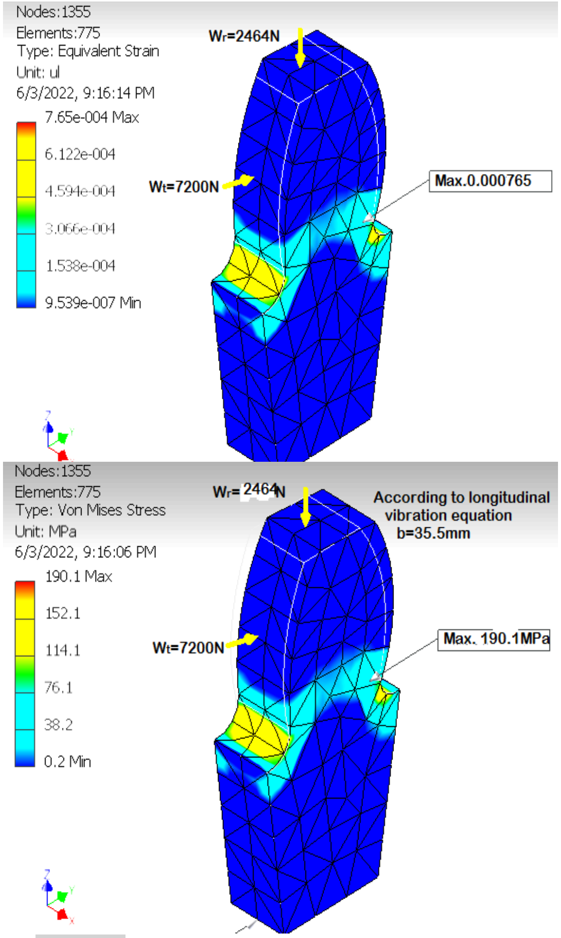


Fig. 10 Simulation Analysis of Stress and Strain on Spur Gear Tooth at V=18.4m/sec, According to Longitudinal Vibration Equation.

Table 2 Simulation Results of Spur Gear Face Width in Two Cases.

Calculation Type	(b) at v=3.76m/sec	(b) at v=9.4m/sec	(b) at v=18.4m/sec
Machine power equation (Lwis equation)	64.2	40.85	32
Longitudinal vibration equation	77.39	51	35.5

3.3.Experimental Work

The Timoshenko equations were used to determine the ratio of dynamic stress to static stress, representing the same ratio of dynamic strain to static strain for any point in the spur gear tooth. Then, the dynamic strain can be measured using a strain gauge, i.e., applying a close contact point between the two teeth of the drive and driven gears during engaging, dynamic strain reading device with transmitter and receiver technique. With the Static IBM program on the computer, as shown in Fig. 11. As previously mentioned, the dynamic strain readings can be recorded. The static strain (0.004) was divided for the spur gear metal. Finally, the equal ratio for stress can be determined, and (W_d) can be calculated at different practical speeds (3.76, 9.4, and 18.4

m/sec) and compared with the longitudinal vibration values for the spur gear tooth to attain the accuracy, as shown in Fig. 12.

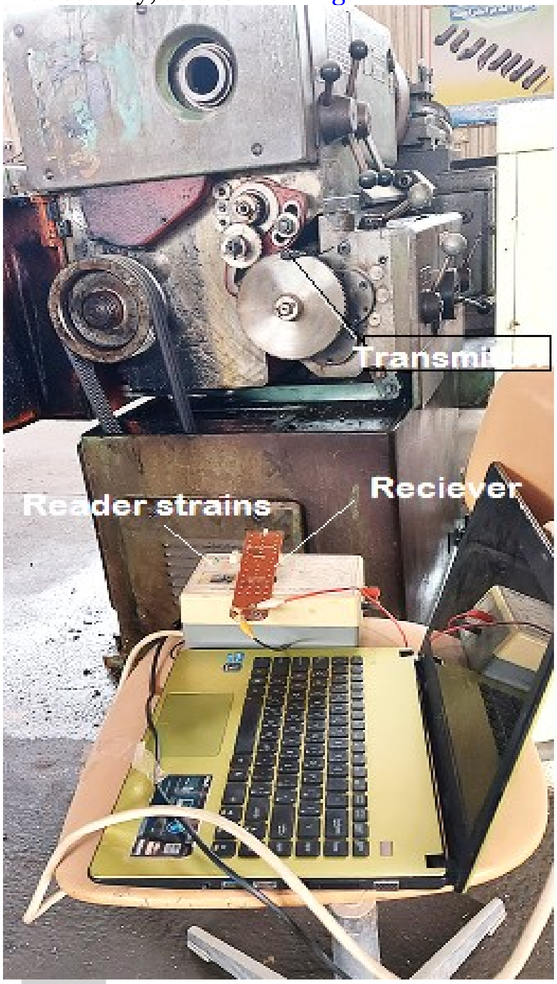


Fig. 11 Experimental Apparatus to Measure Strain.

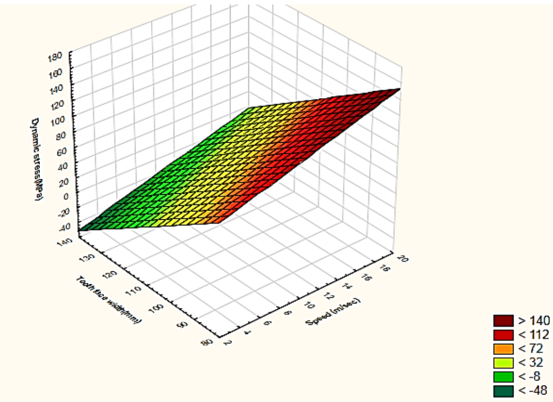


Fig. 12 The Experimental Results with Different Speeds of Spur Gear.

4.DISCUSSION

The design of spur gear and its resistance to dynamic pushing forces and wear are vital in manufacturing, factory production, and dynamic mechanical elements. The design equation (Lewis equation) is a classical equation. The tangential component is always considered in design calculation, while the radial and dynamic loads are neglected. Both loads cause longitudinal vibration along the tooth, representing a cantilever fixed in one end and free in the other. Eq. (1) represents the longitudinal vibration equation, exposed to the spur gear tooth during engaging, and is regarded as a periodical equation, where the tooth rotates one revolution (360°) and again belongs to engaging and expose vibration. The considered parameters were mechanical properties for tooth metal, rotation speed, tooth height, tooth root area, and static stress for the tooth metal. Based on the spur gear studied speeds (3.76, 9.4, and 18.4 m/sec) and machine power and using Eqs. (3, 4, 5, and 6), the quick basic program determined the dynamic loads (W_d) (10125, 11250, and 13116 N), as shown in Figs. (2, 3, and 4), respectively. Depending on the machine's power, the gear dimensions can be designed based on the Lewis equation. The values of (b) were (60, 45, and 30mm) for (3.76, 9.4, and 18.4 m/sec), respectively. The tangential force component (W_t) can be calculated from Eq. (8) and (W_D) values, which depend on the longitudinal vibration equation. Then, the Lewis equation can be applied to determine the values of (b), i.e., 75, 52, and 36mm, based on the studied velocities. This result reveals the importance of the tooth longitudinal vibration. The tooth face width should be increased to avoid failure caused by the tooth longitudinal vibration, as shown in Table 1. Figures (5, 7, and 9) show the static analysis results of (W_t and W_r), produced from the power equation. The values of (b) were (64.2, 40.85, and 32mm) for speeds of (3.76, 9.4, and 18.4 m/sec), respectively. Figures (6, 8, and 10) show the static analysis results of (W_t and W_r) produced from the longitudinal vibration equation. The values of (b) were (77.39, 51, and 35.5mm) for speeds of (3.76, 9.4, and 18.4 m/sec), respectively. The above two simulation analysis results agree with the longitudinal vibration design, i.e., increase the tooth face width to avoid failure. The experimental results showed the same values of (b), as shown in Fig. 12.

5.CONCLUSIONS

1- The longitudinal vibrations, subjected to spur gear tooth, are very dangerous and lead to tooth failure.
2- The design, according to the longitudinal vibration equation, shows an increase in the spur gear tooth face width to resist the produced stresses and avoid failure.
3- The tooth of the spur gear was subjected to falter iteration for stresses at each gear revolution, designed based on fatigue to avoid failure.
4- The simulation static analysis results of the auto desk inventor program effectively produce close theoretical and practical results.

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NOMENCLATURE

Vo	Linear speed	m/sec
N	Rotational speed	rpm
εo	Static strain of tooth material	-
a	Stress speed	m/sec
E	Modulus of elasticity	N/m^2
ρ	Material density	kg/m^3
h	Tooth height	mm
b	Tooth face width	mm
σo	Static stress of tooth material	N/m^2
Aroot	Tooth root area	mm^2
Wr	Radial force component	N
WN	Normal force of tooth	N
WT	Tangential force component	N
Dp	Circular pitch diameter	mm
T	Rotational torque	N·m
P	Rotational power of motor	W
Cv	Velocity factor	-
M	Module of spur gear	-
y	Lewis factor	-
ϕ	Pressure angle of tooth	22.5°
U	Displacement	-
C	Deformation factor	-
t	Time	sec
w	Angular velocity	rad/sec
ℓ	Tooth length	mm
x	Stress translate displacement	-
Cs	Service factor	-

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