Optimization of Gear Tooth Geometry for Compact Design and Enhanced Beam Strength

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ABSTRACT
A critical piece of configuration is settling on OK and ideal boundary decisions, as cog wheels with not many tooth have inadequate conveying limit and unfortunate cross section execution. With an end goal to resolve this issue, we inferred the pressure condition for gear matches with not many tooth and found that pitting on the reaching surface is the transcendent disappointment system of cog wheels with not many tooth. Strength testing likewise affirmed the discoveries of the hypothetical review. The creators presented a clever boundary streamlining approach that limits contact pressure by expecting that the pinion and stuff tooth roots have an indistinguishable bowing weariness life. The hereditary calculation effectively tackled the issue, yielding plan boundaries that are inside decent cutoff points. Using limited component examination, we had the option to verify that the proposed approach essentially further developed the lattice execution and conveying limit of stuff matches with somewhat couple of tooth. Research introduced in this article major areas of strength for offers for drives to additionally promote gear transmissions with a couple of tooth.

Keywords: Gear, toothed gear, strength test, parameter optimization, failure analysis

1. INRODUCTION

Two to six tooth are ordinary for gears with not many tooth. Because of scaling down, weight decrease, and assembling innovation propels, gear matches with a couple of tooth are progressively utilized in air space, accuracy instruments, and other transmission gadgets with huge transmission proportions and unique spatial position prerequisites. Because of their diminished strength, solidness, and cross section execution, gears with a couple of tooth are generally utilized for movement transmission instead of force transmission[1]. To increment gear transmission power with a couple of tooth, tooth configuration should be tackled to increment conveying limit.

I Akira and Y Hidehiro explored gear pair outspread change and found that unreasonably high coefficients lead to pointed tooth, low tooth level, and little contact proportion, which lessen conveying limit[2]. Chien-Fa and Chung-Biau proposed the plan and creation of pinion wheels with a couple of tooth that have no undermined, great contact proportion, and high strength by consolidating outspread stuff change with rack shaper tooth profile change[3]. In Jun, digressive change in view of spiral alteration adjusts rack shaper tooth thickness to alter machined gear tooth thickness and level, supporting stuff pair bearing limit. There was no extra examination or evaluation. Yuehai et al. explored unrelated and outspread changes and offered a plan procedure of equivalent twisting weakness solidarity to settle the low strength of involute pinion wheels with not many tooth. Since gears with not many tooth have unfortunate transmission execution, undercut, and different hardships, Container Feng et al. inspected outspread alteration coefficients and helix points and tracked down reasonable plan decisions[4]. These exploration settled some stuff transmission tooth issues. Be that as it may, the significant disappointment system of stuff pair with a couple of tooth and its conveying limit boundaries still up in the air. Because of the absence of involute tooth profile in gears with a couple of tooth, a few researchers proposed applying curve, LogiX, and S-molded tooth profiles, which can change the lattice structure and further develop gear pair bearing limit, however complex tooth profiles increment gear producing intricacy. Spitas and Spitas10 propose a non-standard, ideal option involute stuff plan with similar pitting obstruction as ordinary cog wheels however most extreme twisting opposition[5]. Results show ideal plans might diminish filet pressure by 8.5%. Scientists have analyzed topsy-turvy gear and other outrageous plans. Kapelevich's mathematical hypothesis of cog wheels with uneven tooth takes into consideration expanded load limit, decreased weight, size, and vibration. Cavdar et al. fostered a PC helped approach for bowing strength investigation of hilter kilter involute prod gears. These enhancement hypothesis and PC supported strategies further develop gear strength and might be utilized in gears with not many tooth[6]. This paper inspected the significant disappointment system of stuff transmission with a couple of tooth, tooth profile plan, and stuff pair really look at hypothesis in view of hypothetical examination and trial confirmation[7]. A limited non direct improvement numerical model in light of equivalent tooth root bowing weakness life of pinion wheels with two way changes was made to build the bearing limit of stuff matches with a couple of tooth and limit contact pressure[8]. The model was changed into an unconstrained improvement issue with discipline using an external discipline ability. The developmental calculation streamlined the reenactment and created the tooth profile plan boundaries for gear matches with 4 to 60 tooth. At long last, limited component examination was performed to analyze and decipher discoveries. After streamlining, gear matches with a couple of tooth have better bearing limit and lattice execution[9].

2. Analyzing The Strength Of A Gear Pair With Limited Tooth

Principal gear transmission disappointment components remember pitting for reaching surface and tooth weariness. Computing contact and bowing weariness strength for gears with not many tooth is basic. Gear matches with a couple of tooth have pointed pinion tooth and lattice past pitch point[10]. Hence, the old strength computation approach is at this point not substantial, and the condition should be examined. Helical cog wheels with a couple of tooth give persistent transmission. Hence, this article utilizes involute barrel shaped helical stuff with insignificant tooth.

A. Estimation of contact pressure

Pitting occurs along the pitch line on one side of the tooth root, hence for traditional involute material strength testing, the pitch feature register contact tension is used. Whatever the situation may be, the pitch point is not appropriate for calculating contact pressure as the cross area line is within the gear pitch circle and the contact extent of stuff pairs with two or three tooth is usually less than 1[11]. Consequently, the transmission fitting point with most noteworthy contact pressure should be found. The cross section point with the most brief total shape span has the most noteworthy contact pressure. The condition of thorough curve range is where r1 and r2 are the lattice point ebb and flow radii of pinion and stuff, individually; r0 1 and r0 2 are their pitch radii; a0 is the cross section point; and s is their distance.

$$\frac{1}{\rho_\Sigma} = \frac{1}{\rho_1} + \frac{1}{\rho_2} = \frac{1}{r_1' \sin\alpha' + s} + \frac{1}{r_2' \sin\alpha' - s} \quad (1)$$

Each involute profile point has a particular shape and contact pressure all through the functioning tooth profile. Utilizing condition (1), the cross section line's finished arch was determined (Figure 1). The cross section point B2 with the greatest complete arch range ought to be utilized to register contact pressure. Hertz condition works out tooth contact pressure[12].

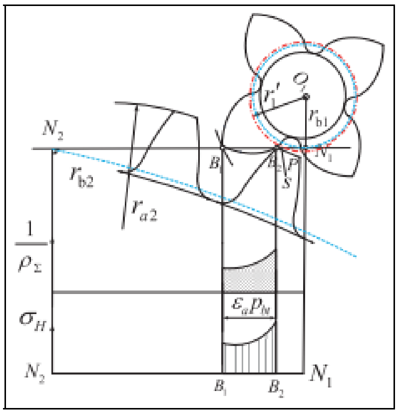


Figure 1. All out ebb and flow of the convergence point

$$\sigma_H = \sqrt{\frac{1}{\left(\frac{1-\mu_1^2}{E_1} + \frac{1-\mu_2^2}{E_2}\right)} \frac{F_n}{L\pi\rho_\Sigma}} \quad (2)$$

The ordinary ascertaining load Fn = KT1=(rv cos at cos bb), The pinion width (b), the storage factor (K), the overall length of the contact line on the tooth (L) (bea=cos bb), the sweeping bend range of the meshi (rS), and a similar estimate of the pinion under force T (rn) are all defined.

$$\varepsilon_\alpha = \frac{z_1(\tan\alpha_{a1} - \tan\alpha') + z_2(\tan\alpha_{a2} - \tan\alpha')}{2\pi} \quad (3)$$

Honed gear with a couple of tooth has a more modest addendum circle than the customary equation works out. Condition (4) applies to addendum circle pressure point aa1.

$$\operatorname{inv}\alpha_{a1} = \frac{1}{z_1}\left(\frac{\pi}{2} + 2x_{r1}\tan\alpha_t + x_{t1}\right) + \operatorname{inv}\alpha_t \quad (4)$$

The whole curve range of B2 should be determined. The lattice point B2 arch radii on pinion and stuff are determined utilizing Euler-Savary condition.

$$\rho_2 = \overline{B_2N_2} = \sqrt{r_{a2}^2 - r_{b2}^2} \quad (5)$$

$$\rho_1 = (r_{b1} + r_{b2})\tan\alpha' - \sqrt{r_{a2}^2 - r_{b2}^2} \quad (6)$$

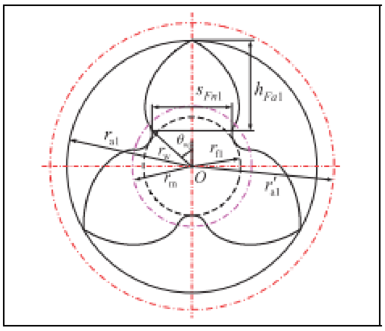


Figure 2. Perilous area around pinion tooth root

$$\rho_\Sigma = \frac{\sqrt{{r_{a2}}^2 - {r_{b2}}^2}\left[(r_{b1} + r_{b2})\tan\alpha' - \sqrt{{r_{a2}}^2 - {r_{b2}}^2}\right]\cos\beta_b}{(r_{b1} + r_{b2})\tan\alpha'} \quad (7)$$

By subbing condition (7) into condition (2), the greatest contact pressure might be registered.

$$\sigma_{Hmax} = \sqrt{\frac{KT_1}{\pi r_v b\varepsilon_\alpha \cos\alpha_t\left(\frac{1-{\mu_1}^2}{E_1} + \frac{1-{\mu_2}^2}{E_2}\right)}\frac{1}{\rho_\Sigma}} \quad (8)$$

B. Calculation of bending stress

Standard design and calculation formulas should be revised to account for the bending pressure of an involute material pair with two tooth, pointed tooth, and extraneous changes on the pinion[13]. Yuehai et al.4's restricted component method identified the potentially hazardous region on the pinion root (Figure 2). To get the distance from the site of force application to the hazardous area hFa1 and the thickness of the hazardous area on the tooth root sFn1, we use formulas[14].

$$h_{Fa1} = r_{a1} - r_w\cos\theta_{wj} \quad (9)$$

$$s_{Fn1} = 2r_w\sin\theta_{wj} \quad (10)$$

With rw being the length of the hazardous section, ra1 being the additional circle scope of the pinion, and rm being the circle clear where the change twist and involute partner, the equation uwj = ½arccos (rb=rm)+(xt1 + 2p)=z1 = 2 finds the circle point of the dangerous area[15].

$$r_m = \sqrt{(r_1 - h_a^* m_t + x_{r1} m_t)^2 + \left[\frac{(h_a^* m_t - x_{r1} m_t)}{\tan\alpha}\right]^2}$$

The tooth twisting weariness stress condition depends on the cantilever pillar stress thought in Mechanics of Materials.

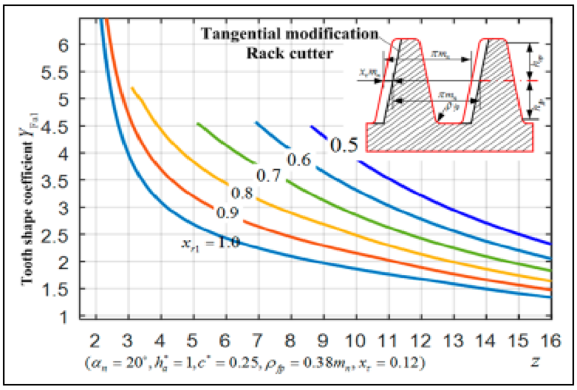


Figure 3. Shape of the tooth It is YFa1.

$$\sigma_{F1} = \frac{KT_1}{r_v b m_n} Y_{Fa1} Y_{Sa1} Y_\varepsilon Y_\beta \qquad (11)$$

$$Y_{Fa1} = \frac{6\cos\alpha_{a1}(h_{Fa1}/m_n)}{\cos\alpha_t (s_{Fn1}/m_n)^2} \qquad (12)$$

where Stuff Manual records Ysa, Ye, and Yb as stress adjustment, contact proportion, and helix point factors.

To develop strength checking of stuff coordinates with several tooth, The YFa1 component of the tooth structure is subject to negligible variation and is not fixed according to condition (12) (Figure 3). YFa1 is dimensionless and solely associated with tooth number, change variable, and strain point[17].

Figure 4 shows the stuff twisting weariness stress condition.

$$\sigma_{F2} = \frac{KT_1}{r_v b m_n} Y_{Fa2} Y_{Sa2} Y_\varepsilon Y_\beta \qquad (13)$$

The unrelated change coefficient is added to the situation to decide sFn2 and hFa2 utilizing the 308 Digression Strategy. Subtleties are not given since it is straightforward. Figure 5 shows gear tooth structure factor with digressive change[18].

C. Strength analysis of gear pair with a few tooth

The significant disappointment system of stuff pair with a couple of tooth decides boundary determination involving the condition for contact pressure or twisting pressure; extra examination will be finished here.

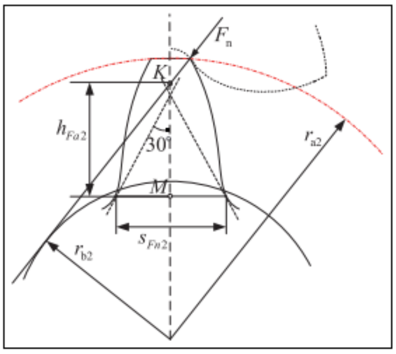


Figure 4. Risky area on gear tooth root.

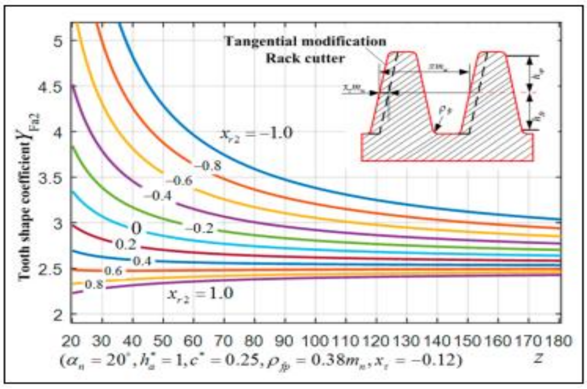


Figure 5. Shape of the tooth YFa2

The changing guideline of contact pressure and twisting pressure of stuff pair with 2 to 5 tooth with module is found from the situations (Figure 6)[19]. The transmission force T is 1 N m, the strain point an is 208, the helix point b is 258, the stuff tooth number z2 is 60, and the pinion takes the least change coefficient under no subverting. Together, these parameters provide a complete picture. When the number of gear tooth is between fifty and one hundred, a comparison rule comes into play[20].

The above mathematical review yields the accompanying ends:

1. Contact weakness is the essential bearing limit element and head disappointment system for gear matches with a couple of tooth.
2. Gear modules beneath 0.5 see critical expansions in tooth contact and twisting pressure.
3. The twisting pressure of pinion is higher than that of stuff for module values somewhere in the range of 0.5 and 2.5.

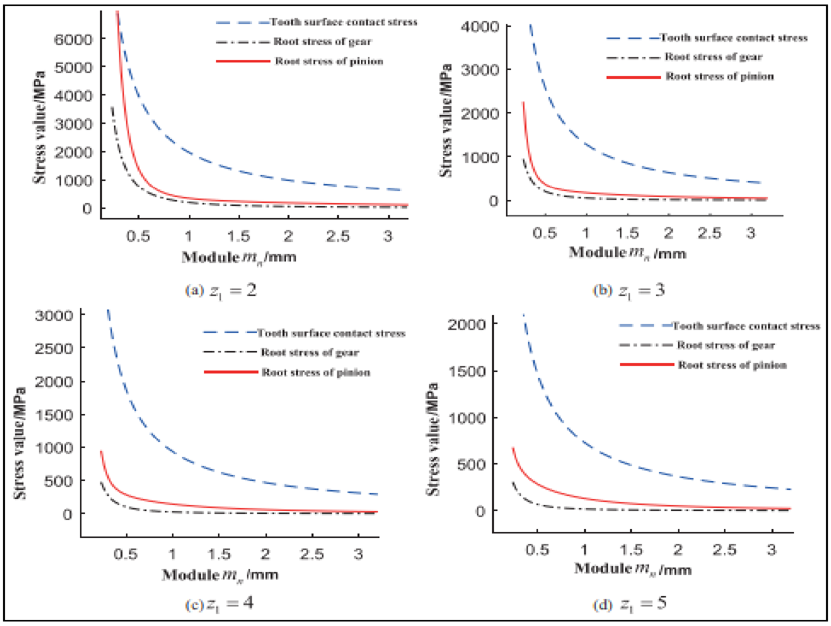


Figure 6. Strength assessment of multi-module gears with few tooth.

Table 1. Test device boundaries.

Division	**Number of Tooth**	**Module (mm)**	**Angle of Helix (°)**	**Perpendicular to Normal Pressure**	**Normal-Plane Radial Modification Coefficient**
Division 1	2	1.75	28	23.7	0.8
	60	1.75	28	23.7	-0.4754
Division 2	4	1.75	25	23.7	0.7
	60	1.75	25	23.7	-0.5787

To approve the hypothetical computations, a couple of tooth gear pair with Table 1 qualities is strength tried. In Figures 7 and 8, we can see the outcomes and the evidence. After 16.6 hours at 50 N m and 1000 r/min, Figure 8(a) reveals that the gear tooth has developed small-scale pitting. Under the same testing circumstances, the 2-tooth pinion shaft failed after 31.2 hours, as shown in Figure 8(b). Figure 8(c) shows that after 28.5 hours of operation at 60 N m and 1000 r/min, the gear tooth exhibited wear and slight pitting[21].
After the stuff pair test, the pinion tooth surface had pitting and a little measure of holding, and the shaft with two tooth tumbled off. Gear tooth had plastic twisting and holding districts and clear wear.

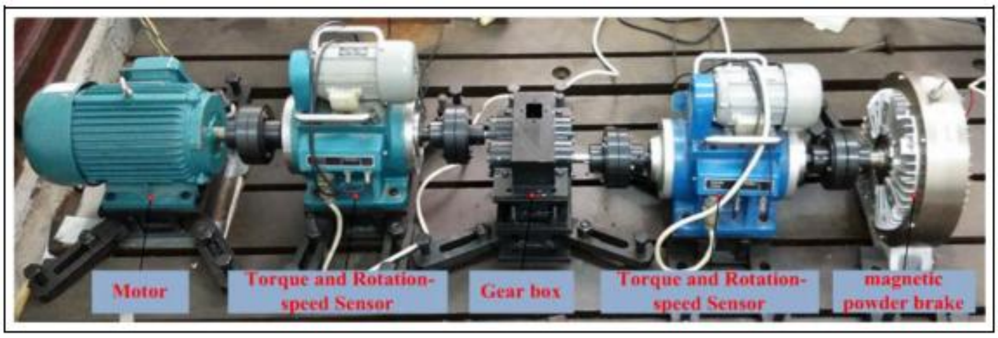


Figure 7. Mechanical testing platform

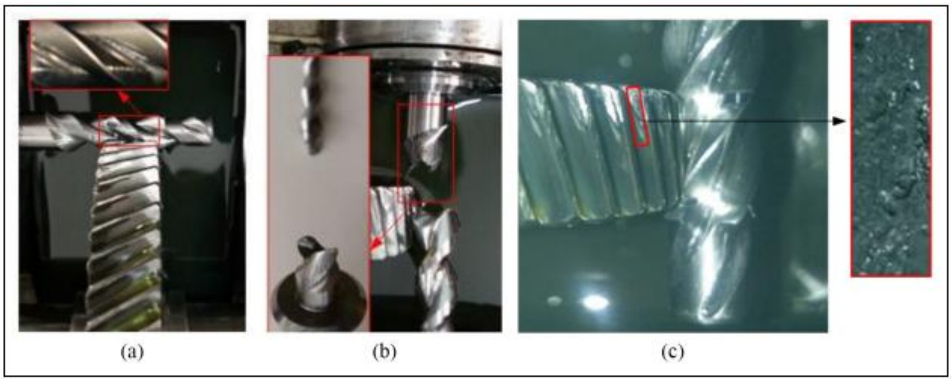


Figure 8. Strength test results for gear pair with gear extents of 2/60 and 4/60: (a) pitting appeared, (b) the pinion shaft was broken, and (c) tooth wear and pitting appeared.

The hypothetical examination and exploratory check exhibit that contact weariness pitting is the significant stuff transmission disappointment system with few tooth. Hence, the condition of contact weakness strength ought to be utilized to set boundaries for gear matches with a couple of tooth, and the condition of twisting exhaustion strength ought to confirm the discoveries. Because of their high transmission proportion, pinions go through a lot more pressure cycles than gears[22]. Accordingly, pinion is the stuff pair's failure point. Really look at the shaft of pinion's bowing and torsional solidarity to keep it from twisting or breaking.

3. Gear Tooth Profile Design

Teeth (T) = 20
Pitch Circle Diameter = D
Pressure Angle(ϕ) = 200
Take module (m) = 2.5

We know, $m = \frac{D}{T}$

$$2.5 = \frac{D}{20}$$

D = 50 mm

The breadth of the circle (D) is 50 mm, thusly.
With a round pitch (C.p) of 7.85, the equation Pc = ΠD/T = Π.50/20 = .
0.4 is the breadth pitch, or 20 partitioned by 50.
Aspects: M = 2.5 mm Addendum

The profundity of the dendrite is 1.25 meters, or 3.125 millimeters.
A tooth's thickness is 3.927 millimeters, or 1.5708meters.
With a filet span of 0.4 m/1 mm and a work profundity of 2 m/5 mm and a base all out profundity of 2.25 m/5.62 mm, the determined aspects are as per the following.
Addendum in addition to pitch circle breadth rises to 27.5 mm for top distance across.
The donut's pitch, estimated in breadth, is 21.875, which is equivalent to 25 short 3.125.
Sixty-6.25" is the freedom profundity, which is determined as all out profundity short addendum and dendum.

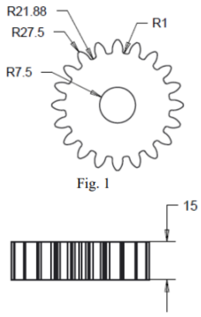


Figure 9

Sr.No.	Gear Terminology	Description
1	No. of Teeth	20
2	PCD	50
3	Module	2.5
4	Working depth	5
5	Tooth thickness	3.927
6	Min total depth	5.625
7	Pressure angle	20°

Figure 10

4. Creo Modeling

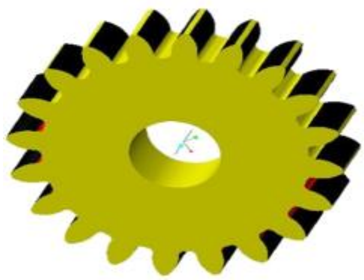


Figure 11. Designed machinery

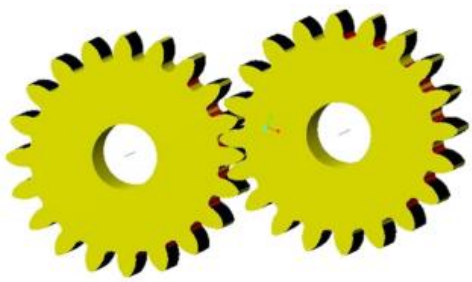


Figure 12. Gears meshing

Figure 13. Tooth profile

5. Determining Tooth Load Through Module Changes

A. Assuming that module(m) equals 2,

Then, at that point, b = 6m = 12

Take $\sigma_o = 12.6$ kgf/mm^2 = 126 N/ mm^2

$$C_v = \frac{3}{3+v}$$

$$C_v = \frac{3}{3+3} = 0.5\ (as\ v = 3m/s)$$

$$W_T = \sigma_W\, b\, \pi\, m\, y$$

$WT = 126 * 0.5 * 12 * 2 * 0.1084 * \pi = 514.88\ N$ Thus, extraneous tooth load = 514.88 N

B. A module with m = 3

Then b = 6m = 18

Take $\sigma_o = 12.6$ kgf/mm^2 = 126 N/ mm^2

$$C_v = \frac{3}{3+v} = \frac{3}{3+3} = 0.5\ (as\ v = 3m/s)$$

$$W_T = \sigma_W\, b\, \pi\, m\, y$$

$WT = 126 * 0.5 * 18 * 3 * 0.1084 * \pi = 1158.88\ N$ In this way, distracting tooth load = 1158.88 N

C. Assuming that module(m) equals 4,

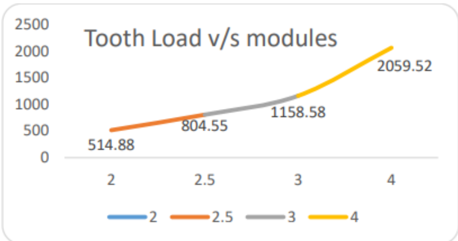
Then, at that point, b = 6m = 24

Take $\sigma_o = 12.6$ kgf/mm^2 = 126 N/ mm^2

$$C_v = \frac{3}{3+v} = \frac{3}{3+3} = 0.5\ (as\ v = 3m/s)$$

$$W_T = \sigma_W\, b\, \pi\, m\, y$$

$WT = 126 * 0.5 * 24 * 4 * 0.1084 * \pi = 2059.52$ N



6. Adjusting The Width Of The Tooth To Determine The Load

A. Length b = 8 m = 20 mm

Take σo = 12.6 kgf/mm2 = 126 N/mm2

$$C_v = \frac{3}{3+v} = \frac{3}{3+3} = 0.5\ (as\ v = 3m/s)$$

$$W_T = \sigma_W\, b\, \pi\, m\, y$$

$WT = 126 * 0.5 * 20 * 2.5 * 0.1084 * \pi = 1072.73\ N$, Thus, digressive tooth load = 1072.73 N

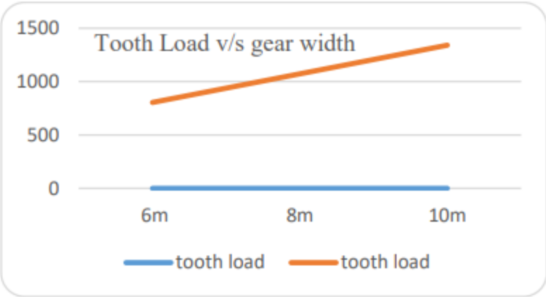
B. Length b = 10 m = 25 mm

Take σo = 12.6 kgf/mm2 = 126 N/mm2

$$C_v = \frac{3}{3+v} = \frac{3}{3+3} = 0.5\ (as\ v = 3m/s)$$

$$W_T = \sigma_W\, b\, \pi\, m\, y$$

$WT = 126 * 0.5 * 25 * 2.5 * 0.1084 * \pi = 1340.91$ N



7. Using Solid Works, Evaluate Gear Tooth

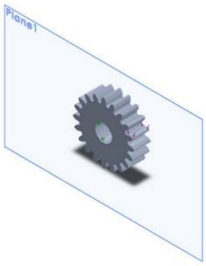


Figure 14. Information sent out from Solid Works

A. Properties:

- 0.0702584 kg mass
- 2.60216e-005 m^3 volume
- Thickness: 2,700 kg/m^3.
- Gauge: 0.688533 Newtons

B. Material properties:-

- Default disappointment metric: Max von Mises Tension; model sort: Direct Adaptable Isotropic; name: 1060 Composite
- The strength of the yield is 2.75742e+007 N/m^2.
- Rigidity: 6.89356e+007 N/m^2.

C. Loads and Fixtures:-

Fixed-1

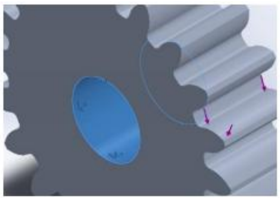


Figure. 15

Force-1

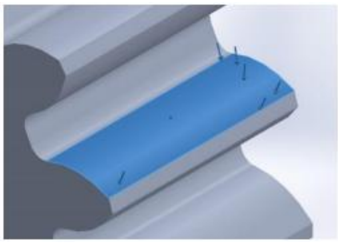


Figure. 16

D. Mesh Information:

- Absolute Hubs - 22392
- Absolute Components - 14031
- Component Size - 2.96436 mm
- Network Quality - High
- Jacobian focuses - 4 Focuses

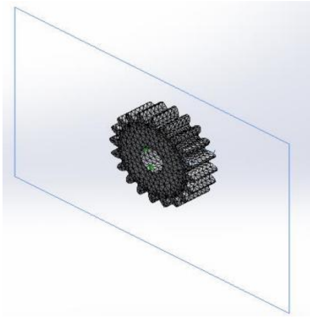


Figure. 17

E. Results:

- Stress Type-von Mises Pressure
- MIN. Esteem - 827.971 N/m^2
- At Hub: 17359
- Max. Esteem - 6.03174e+007 N/m^2
- Hub: 21007

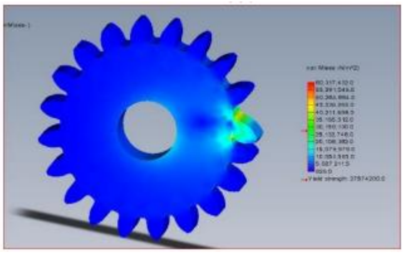


Figure. 18 Von-mises stress

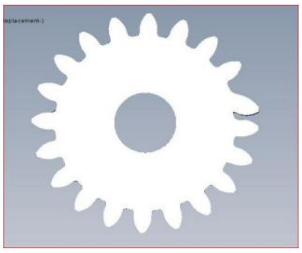


Figure. 19 Deformed Shape

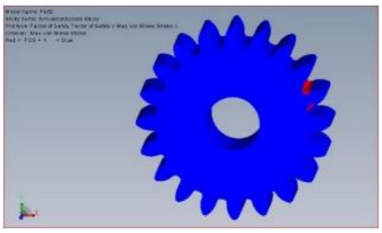


Figure. 20 Factor of safety

CONCLUSION

Hypothetical review demonstrates the way that a mathematical procedure can figure the most extreme contact pressure of stuff with a couple of tooth, and that contact weariness on tooth surface is the central

point impacting bearing limit and stuff pair disappointment component, which is tried. Laid out gear pair strength estimation procedure.
To accomplish a practically identical contorting shortcoming life, it is important to make a computational model of the tooth profile that limits the contact weight on the tooth surface.
In the model, LTCA is finished utilizing limited component procedure when streamlining of plan boundaries. Results uncover that the enhanced stuff pair with a couple of tooth works on bearing limit and cross section execution, offering a strategy to pick gear particulars.

REFERENCES

[1] B.Venkatesh, V.Kamala2, A.M.K. Prasad3Parametric Approach to Analysis of Aluminum alloy Helical gear for High Speed Marine ApplicationsVol.1, Issue1, pp-173-178
[2] J.A. Wright, et al, Design, development and application of new, high-performance gear steels, Gear technology(2010),pp 46 –53[2]Rao, C.M., and Muthuveerappan G., Finite Element
[3] Modeling and Stress Analysis of Helical Gear, Tooth, Computers & structures, 49,pp.1095-1106, 1993. [4] Marappan, S. and Venkataramana, 2004,ANSYS Reference Guide., CAD CENTRE, India
[4] PSG, 2008.Design data, KalaikathirAchchagampublishers, Coimbatore, India
[5] Shigley, J.E. and Uicker,J.J,Theory of machines and mechanisms,McGraw Hill, 1986.
[6] R.S. KHURMI and J.K. GUPTA,Theory of machine , S. Chand publications, Edition 16 reprint (2008), pp.382-397.
[7] "Machine Design" by S.Md.Jalaludeen,AnuradhaPublications(2009).
[8] "Design Data Hand Book for Mechanical Engineers"ByK.Mahadevan&K.Balaveera Reddy
[9] Akira I and Hidehiro Y. Design and manufacturing processes and load carrying capacity of cylindrical gear pairs with 2 to 4 pinion tooth for high transmission ratio (1st report design and manufacture and surface durability of gears with 2 to 3 pinion tooth). T Jpn Soc Mech Eng 1981; 416: 507–515.
[10] Chien-Fa C and Chung-Biau T. Tooth profile design for the manufacture of helical gear sets with a few tooth. Int J Mach Tool Manu 2005; 45: 1531–1541.
[11] Jun J. Bilateral modification of cylindrical gear pair with a few tooth. Chinese Patent, ZL94208649P, 24 May 1995.
[12] Yuehai S, Yan-Feng L and Luxi D. Modification of involute gear pair with a few tooth under the condition of equal tooth bending strength. J Tianjin Univ 2014; 11: 1001–1007
[13] Pan-Feng Z, Xiu-Bing J and Cai-Fang M. Involute gearing with small tooth number. Chin Mech Eng Soc 2004; 17: 113–115.
[14] Akira I and Hidehiro Y. Design, manufacture and load carrying capacity of Novikov gears with 3–5 pinion tooth for high transmission ratio (1st report, design, manufacture and power transmission efficiency). T Jpn Soc Mech Eng 1983; 447: 2039–2047.
[15] Komori T, Ariga Y and Nagata S. A new gears profile having zero relative curvature at many contact points (LogiX tooth profile). J Mech Des: T ASME 1990; 112: 430–436.
[16] Qiang S and Yuehai S. Geometry modeling and stress analysis of S-type-profile gear with a few tooth. J Tianjin Univ 2016; 7: 702–708.
[17] Litvin FL. Gear geometry and applied theory. Englewood Cliffs, NJ: Prentice-Hall, 1994.
[18] Spitas V and Spitas C. Optimizing involute gear design for maximum bending strength and equivalent pitting resistance. Proc IMechE, Part C: J Mechanical Engineering Science 2007; 221: 479–488.
[19] Kapelevich A. Geometry and design of involute spur gears with asymmetric tooth. Mech Mach Theory 2000; 35: 117–130.
[20] Cavdar K, Karpat F and Babalik FC. Computer aided analysis of bending strength of involute spur gears with asymmetric profile. J Mech Design 2005; 127: 477–484.
[21] Gear Manual Editorial Board. Gear manual. Beijing, China: China Machine Press, 2000.
[22] Qiang S and Yuehai S. Calculation in bending and torsional deformation of gear with a few tooth. Mech Des Res 2015; 2: 46–50.