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Design and Selection of Material for Plastic Gears

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Abstract

The use of plastic and polymer composite gears is increasing because of their low cost, lightweight and quiet operation compared to metal gears. Plastic gears find application in printers, cameras, timers, counters, *etc.* Four different combinations of materials for pinion and gear were selected in the present work, and a comparative study was done to investigate mechanical and thermal properties analytically. Analytical results were validated using SOLIDWORKS and ANSYS. Design 1 and Design 2 were found out to be preferable designs. Maximum principal and maximum shear stress generated were minimum for Design 1 (pinion of Polycarbonate and gear of Acetal copolymer). Simultaneously, the deformation and temperature rise were minimum for Design 2 (pinion of Nylon 66 and gear of Acetal copolymer).

Keywords

ANSYS; Plastic gear; Design; Pinion

1. INTRODUCTION

Plastic gears are an excellent alternative to metal gears because of their lightweight, low noise production, low cost, and high corrosion resistance [1] [2]. Historically, plastic gears were used mainly for light-duty applications such as printer, watches, toys, *etc.*, because of their low strength and thermal resistance compared to the metal gears [3] [4]. With the development of stronger and more consistent polymers, plastic gears are also being used for transmitting high power [5]. The most commonly used materials for plastic gears are Acetal Copolymer, Nylon 66, Polycarbonate, Polyester, *etc.* [5] [6]. Various reinforcements such as glass fibers, natural fibers, carbon nanotubes are also added to the above-mentioned polymer matrices to fabricate composite plastic gears [7] [8] [9] [10]. The properties of plastic gears, such as life and noise production, depend on pinion and gear material [11]. Gears are designed based on the types of failures that occur in the gear while in operation. Gear can fail by mechanical process, environmental and working condition [12]. Fatigue is the main reason for teeth breakage. Each time the tooth is engaged, it is subjected to varying load. Hence alternating bending stresses are developed at the root of the teeth. Various wear mechanisms are adhesive wear, abrasive wear, pitting, plastic flow, *etc.*

Four different combinations of materials for pinion and gear were selected in the present work, and a comparative study was done to investigate mechanical and thermal properties analytically. Analytical results were validated using SOLIDWORKS and ANSYS.

2. MATERIALS AND METHOD

Four different combinations of materials for pinion and gear were selected. Materials used for specific design are given in Table 1. Comparative study of the mechanical and thermal properties of mashing of pinion and gears for all the selected design was done analytically using data available in the literature [5] [13] [14] [15] [16]. Analytical results were verified using ANSYS (R15.0). Steps taken to analyse designs using ANSYS are shown in **Figure 1**.

Table 1 Design of gears

	Pinion	Gear
Design 1	Polycarbonate	Acetal copolymer
Design 2	Nylon 66	Acetal copolymer
Design 3	Acetal Copolymer + 30% Glass Fibre	ABS + 30% Glass Fibre
Design 4	ABS + 30% Glass Fibre	Nylon 66

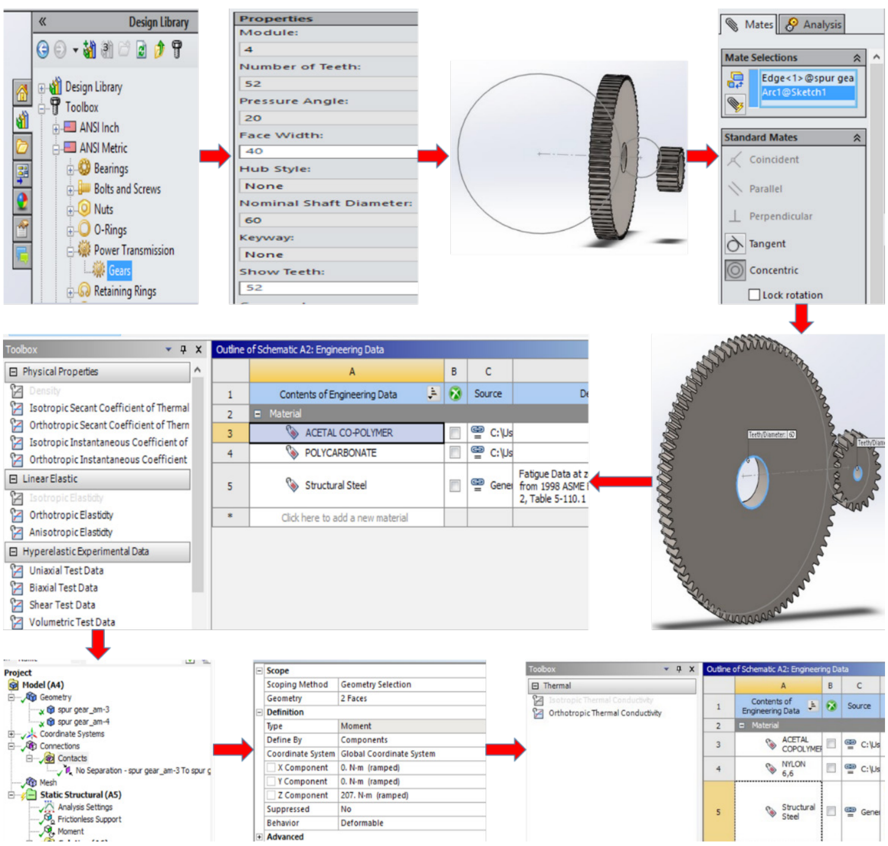

Figure 1 Step taken to analyse designs using ANSYS

3. SYMBOLS AND FORMULAE USED

3.1 Symbols

a = Centre distance between shafts (cm)
i = Gear ratio
Z_1 = Number of teeth on pinion
Z_2 = Number of teeth on gear
M = Standard module (mm)
d = Pitch circle diameter (mm)
p_c = Circular pitch (mm)
b = Face width (cm)
α = Pressure angle
ψ = b/a
ψ_m = b/m
M_t = Torque transmitted by pinion (Ncm)
D_b = Diameter of base circle (mm)
E_1 = Young's modulus of the pinion
E_2 = Young's modulus of the gear
E = Equivalent Young's modulus
σ_e = Endurance strength
F_t = Tangential force
d_1 = Diameter of pinion
N_1 = Speed of pinion (rpm)
K_{cl} = Life factor
C_R = Factor depending on surface hardness
N_2 = Speed of gear (rpm)
K_{ext} = Wear factor
φ = Pressure angle
P = Power (kW)
S_c = Surface stress intensity
θ_1 = Surface temperature of pinion
θ_2 = Surface temperature of gear
θ_a = Ambient temperature (30℃)
f = Friction factor (0.2-0.25)
U = Transmission ratio (Z_2/Z_1)
K_1, K_2 = VDI 2545 factors (2.5)
K_3 = Housing factor (0)
A = Surface area of housing
$\chi_{1,2}$ = Index of material of pinion and gear

3.2 Formulae

Formulae used for the design of plastic gears are given below [13] [14] [15] [16].

$$M_t = \frac{60 * P}{2\pi N_1} \tag{1}$$

$$[M_t] = M_t * K * K_d \tag{2}$$

$$E = \frac{E_1 * E_2}{E_1 + E_2} \tag{3}$$

$$a \geqslant (i+1)\sqrt[3]{\frac{(0.74)^2 * [M_t] * E}{(\sigma_c)^2 * i * \psi}} \tag{4}$$

$$m = 1.26\sqrt[3]{\frac{[M_t]}{Y * \sigma_b * \psi * m * Z_1}} \tag{5}$$

$$Z_1 = \frac{2a}{(i+1) * m} \tag{6}$$

$$a_{actual} = \frac{d_1 + d_2}{2} \tag{7}$$

$$b = \psi * a = \psi * m \tag{8}$$

$$V = \frac{\pi d_1 N}{60} \tag{9}$$

$$\sigma_c = C_R * HRB * K_{cl} \tag{10}$$

$$F_t = \frac{60P}{\pi d_1 N_1} \tag{11}$$

$$K_{ext} = \frac{F_t}{d_1 * b}\left(1 + \frac{N_1}{N_2}\right) \tag{12}$$

$$S_c = \sqrt{\frac{0.7 * K_{ext}}{\left(\frac{1}{E_1} + \frac{1}{E_2}\right) * cos\varphi * sin\varphi}} \tag{13}$$

$$Unit\,load = \frac{F_t}{m*b} \tag{14}$$

$$\theta_{1,?2} = \theta_a + 136Pf\frac{U+1}{Z_2+5}\left\{\frac{17100}{b*Z_{1,?2}}\left(\frac{K_2}{(Vm)^{\chi_{1,?2}}}\right)+6.3\frac{K_3}{A}\right\} \tag{15}$$

3.3 Pre-requisite for Design

Pre-requisite (i.e., physical properties, mechanical properties, and design data) for the design of plastic gears are given in **Table 2**, **Table 3**, and **Table 4** [13] [14] [15] [16].

Table 2 Properties of polymers used

Properties	Acetal Copolymer	Poly- Carbonate	Nylon 6,6	ABS + 30% Glass Fibre	Acetal copolymer + 30% Glass Fibre
Density (g/cc)	1.41	1.2	1.15	1.28	1.6
Tensile strength (MPa)	65.5	60	82	40	91.6
Young's modulus (GPa)	2.76	2.35	3.4	5.66	12.2
Flexural strength (MPa)	82.7	89.6	117.2	182.75	206.84
Flexural modulus (GPa)	2.76	2.41	2.49	6.9	7.79
Compressive strength (MPa)	103	79.3	92	42	100
Hardness (Rockwell)	55, R120	M75, R126	M88	R108	M87, R99
Co-eff. of Linear thermal exp.	$5.4*10^{-5}$	$7*10^{-5}$	$9.5*10^{-5}$	$4*10^{-5}$	$2.7*10^{-5}$
Thermal conductivity (W/m-K)	0.237	0.187	0.28	0.2	0.23
Poisson's ratio	0.36	0.37	0.41	0.30	0.32

Table 3 Experimental value K_2 (VDI 2545 factor)

	Acetal Copolymer	Nylon 6,6	Poly-carbonate	ABS (+30%Glass Fibre)	Acetal Copolymer (+30% Glass Fibre)
Acetal Copolymer	2.5	2.5	2.2	-	-
Nylon 6,6	2.5	2.4	-	2.7	-
Poly-carbonate	2.2	-	-	-	-
ABS (+30% Glass Fibre)	-	2.7	-	-	2.9
Acetal copolymer (+30% Glass Fibre)	-	-	-	2.9	-

Experimental value χ (index)

For Acetal co-polymer – 0.4 For Nylon 66 – 0.75

For Polycarbonate – 0.55

For ABS (+30% glass fibre) – 0.5

For Acetal co-polymer (+30% glass fibre) – 0.52

Experimental value K3 (Housing factor)

K3 = 0, for open gear with free air access.

K3 = 0.04-0.13, partially enclosed gearbox in which air cannot circulate freely.

K3 = 0.172, totally enclosed gear box.

Table 4 Coefficient of friction for material combination in temperature model

	Acetal copolymer	Nylon 6,6	Polycarbonate	ABS (+30%glass)	Acetal copolymer (+30% glass)
Acetal copolymer	0.28	0.18	0.2	-	-
Nylon 6,6	0.18	0.2	-	0.19	-
Polycarbonate	0.2	-	-	-	-
ABS (+30% glass)	-	0.19	-	-	0.18
Acetal copolymer					
(+30% glass)	-	-	-	0.18	-

4. CALCULATIONS FOR DESIGN OF PARAMETERS

For Design 1,

E = 2.09 GPa, Power (P) = 5 kW, N= 300 rpm, i = 4, $K*K_d$ = 1.3, ψ = 0.3, ψ_m = 10, Z_1 = 20, Hardness = 55 HRB, C_R = 2100, K_{cl} = 0.6177

Using equation (1),

$$M_t = \frac{60*5000}{2\pi*300} = 159.35\,Nm$$

Using equation (2),

$$\left[M_t\right] = 159.35*1.3 = 207.006\,Nm$$

Using equation (10),

$$\left(\sigma_c\right)_{Pinion} = 2100*55*0.6177 = 105317.85\,N/cm^2$$

Using equation (4),

$$a \geqslant (4+1)\sqrt[3]{\frac{(0.74)^2*20700.6*2.09*10^7}{105317.85^2*4*0.3}} = 13.04\,cm$$

Using equation (5),

$$m = 1.26\sqrt[3]{\frac{207.006}{0.32*10*20*89.59*10^6}} = 4.06\,mm$$

Using equation 6,

$$Z_1 = \frac{2*13.04}{5*0.406} = 13.04 \sim 13$$

$$Z_2 = 13*4 = 52$$

$$d_1 = 13*4 = 52\,mm$$

$$d_2 = 52*4 = 208\,mm$$

Using equation 7,

$$a_{actual} = \frac{52+208}{2} = 130\,mm$$

Using equation 8,

$$b = 0.3*130 = 39\,mm\,or\,b = 10*4 = 40\,mm$$

So, b is taken as 40 mm.

$$\psi = \frac{b}{a} = \frac{40}{130} = 0.3077$$

$$\psi_m = b/m = 40/52 = 0.7692$$

$$addendum = 1\,module = 4\,mm$$

$$dedundum = 1.154m = 4.616\,mm$$

Using equation 9,

$$V = \frac{\pi*52*300}{60} = 816\,mm/s$$

Similarly, calculations for other designs were done and the results obtained are given in Table 5.

Table 5 Calculations

	Design 1	Design 2	Design 3	Design 4
Centre line distance (a) (in mm)	130	220	140	130
No. of teeth on pinion (Z1)	13	22	14	13
No. of teeth on gear (Z2)	52	88	56	52
Diameter of pinion (D1)	52	88	56	52
Diameter of gear (D2)	208	352	224	208
Addendum (in mm)	4	4	4	4
Dedundum (in mm)	4.616	4.616	4.616	4.616
Face width (b) (in mm)	39	66	42	39
Velocity (mm/sec)	816	1382.3	879.6	816.8

5. ANALYTICAL RESULTS

5.1 Design According to Failure

For Design 1,

Using equation 11,

$$F_t = \frac{5000}{0.816} = 6.127\, kN$$

Using equation 12,

$$K_{ext} = \frac{6127}{52 * 40}(1+4) = 14.728\, N/mm^2$$

Using equation 13,

$$S_c = \sqrt{\frac{0.7 * 14.728}{\left(\frac{1}{2.76}+\frac{1}{2.35}\right) * cos 20 * sin 20}} = 12.68$$

Using equation 14,

$$Unit\, load = \frac{6127}{4 * 40} = 38.29\, N/mm^2$$

Similarly, calculations for other designs were done and the results obtained are given in **Table 6**. Surface stress intensity and unit load are minimum for Design 2. On increasing the surface stress intensity, wear of tooth increases. On the other hand, on increasing the unit load, the chances of tooth breakage increases. So, according to failure criteria, Design 2 is safer.

Table 6 Calculated values for the designs according to failure

	Design 1	Design 2	Design 3	Design 4
Tangential force (in kN)	6.127	3.617	5.684	6.121
Wear factor (N/mm^2)	14.728	3.114	12.084	14.71
Surface stress intensity	12.68	2.419	5.41	4.21
Unit load (N/mm^2)	38.29	13.7	33.83	38.256

5.2 Temperature Rise

For Design 1,

P = 5 kW, f = 0.2, U = 4, Z_2 = 52, b = 40 mm, Z_1 = 13, K_2 = 2.2, V = 816 mm/sec, m = 4 mm, χ_1 = 0.55, χ_2 = 0.4

Using equation 15,

$$\theta_1 = 30 + 136 * 5 * 0.2 \frac{5}{57}\left\{\frac{17100}{40 * 13}\left(\frac{2.2}{(816 * 4)^{0.55}}\right)\right\} = 40.081℃$$

So,

$$\text{Temperature rise in pinion} = 40.081 - 30 = 10.081^{\circ}\text{C}$$

Using equation 15,

$$\theta_2 = 30 + 136 * 5 * 0.2 \frac{5}{57} \left\{ \frac{17100}{40 * 52} \left(\frac{2.2}{(816 * 4)^{0.4}} \right) \right\} = 38.42^{\circ}\text{C}$$

So,

$$\text{Temperature rise in gear} = 38.42 - 30 = 8.42^{\circ}\text{C}$$

Similarly, calculations for other designs were done and the results obtained are given in **Table 7**. Temperature rise in pinion is minimum for Design 2. At the same time, temperature rise in gear is the minimum for Design 4. The temperature rise in gear for design 2 is also not very high. So, from a thermal failure point of view also, Design 2 is safer.

Table 7 Values of temperature rise in pinion and gear

	Design 1	Design 2	Design 3	Design 4
Temperature rise in pinion (℃)	10.081	0.495	14.616	17.6
Temperature rise in gear (℃)	8.42	2.538	3.83	0.58

5.3 ANSYS Results

Figure 2, **Figure 3**, **Figure 4**, and **Figure 5** show maximum principal stress, maximum shear stress, total deformation, and temperature rise in various designs. ANSYS results are given in **Table 8**.

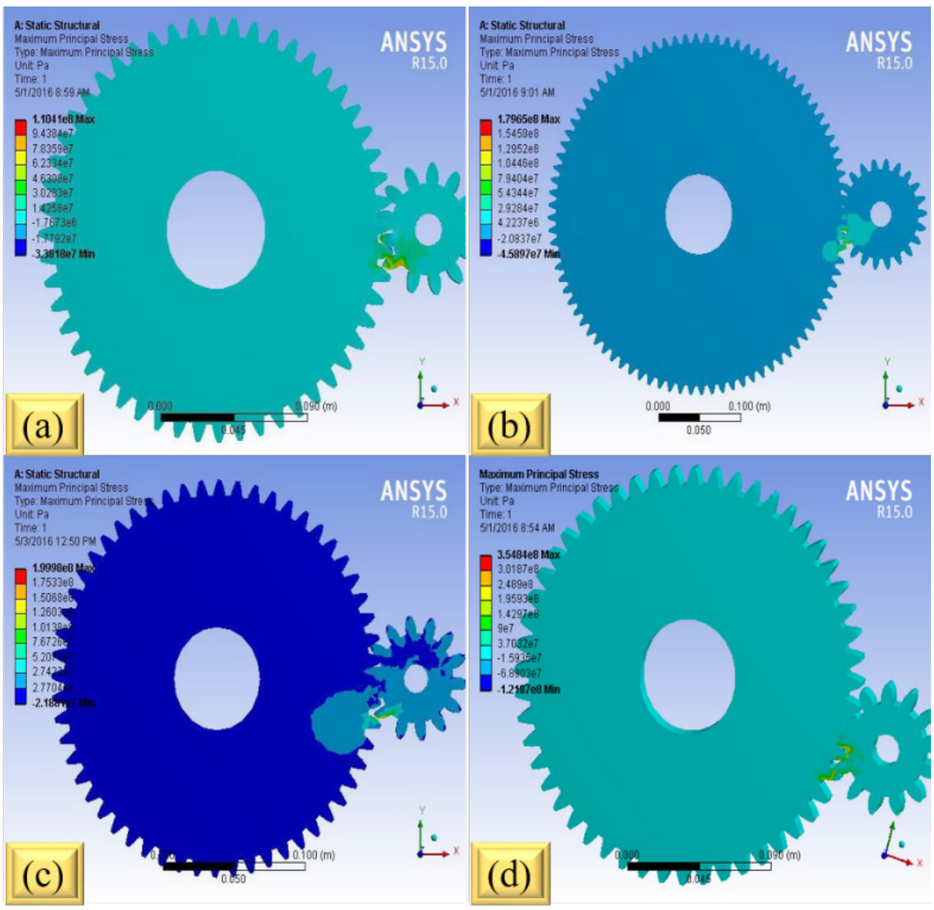

Figure 2 Principal stress generated in (a) Design 1, (b) Design 2, (c) Design 3, and (d) Design 4

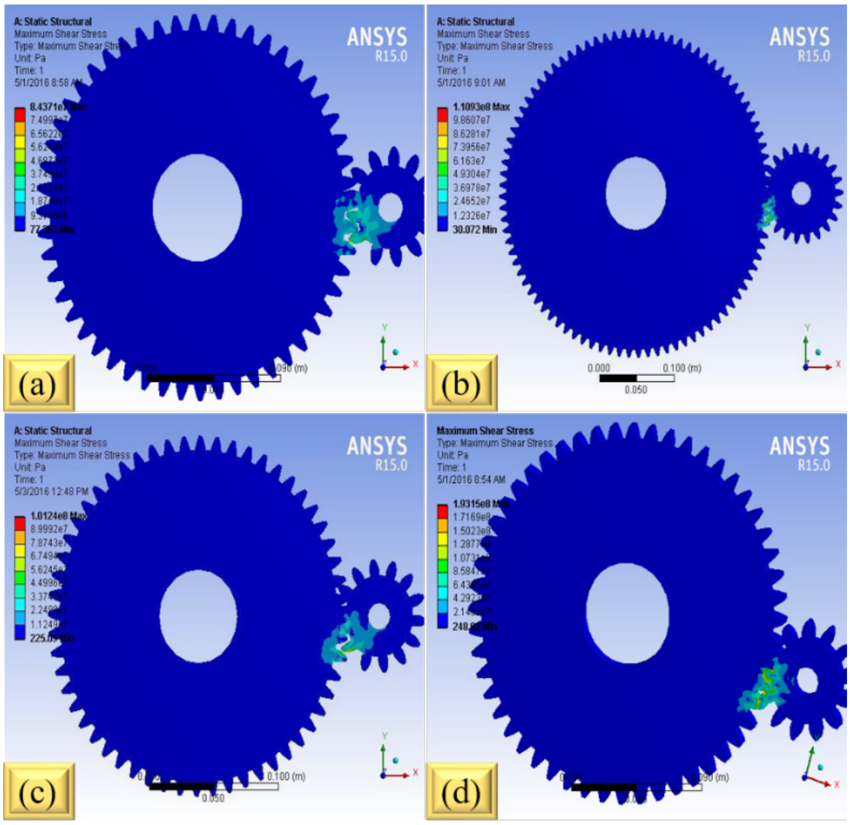

Figure 3 Shear stress generated in (a) Design 1, (b) Design 2, (c) Design 3, and (d) Design 4

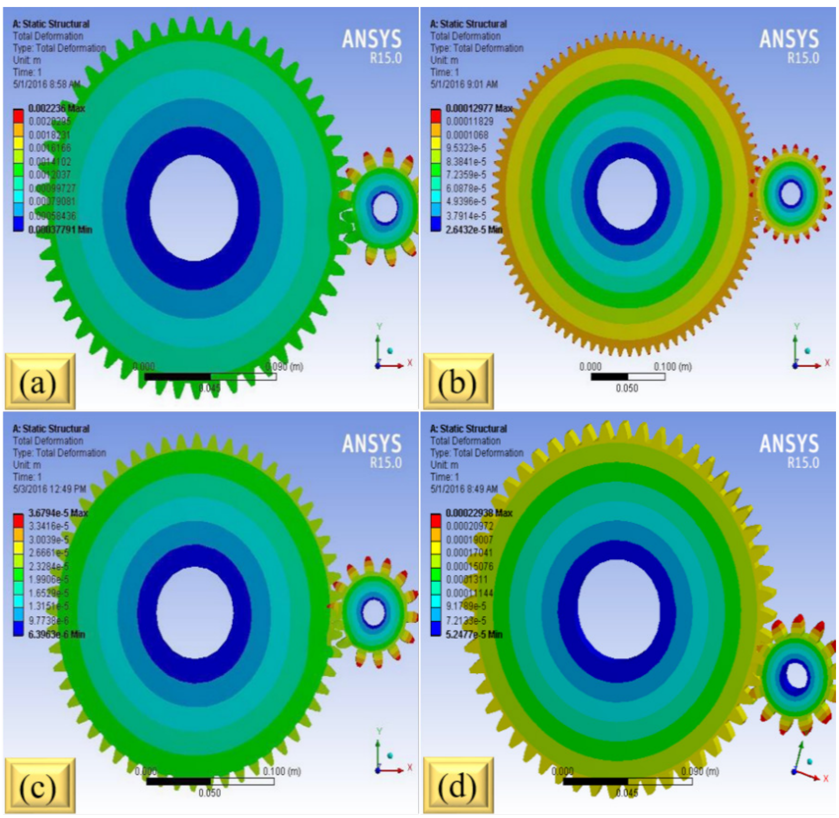

Figure 4 Total deformation in (a) Design 1, (b) Design 2, (c) Design 3, and (d) Design 4

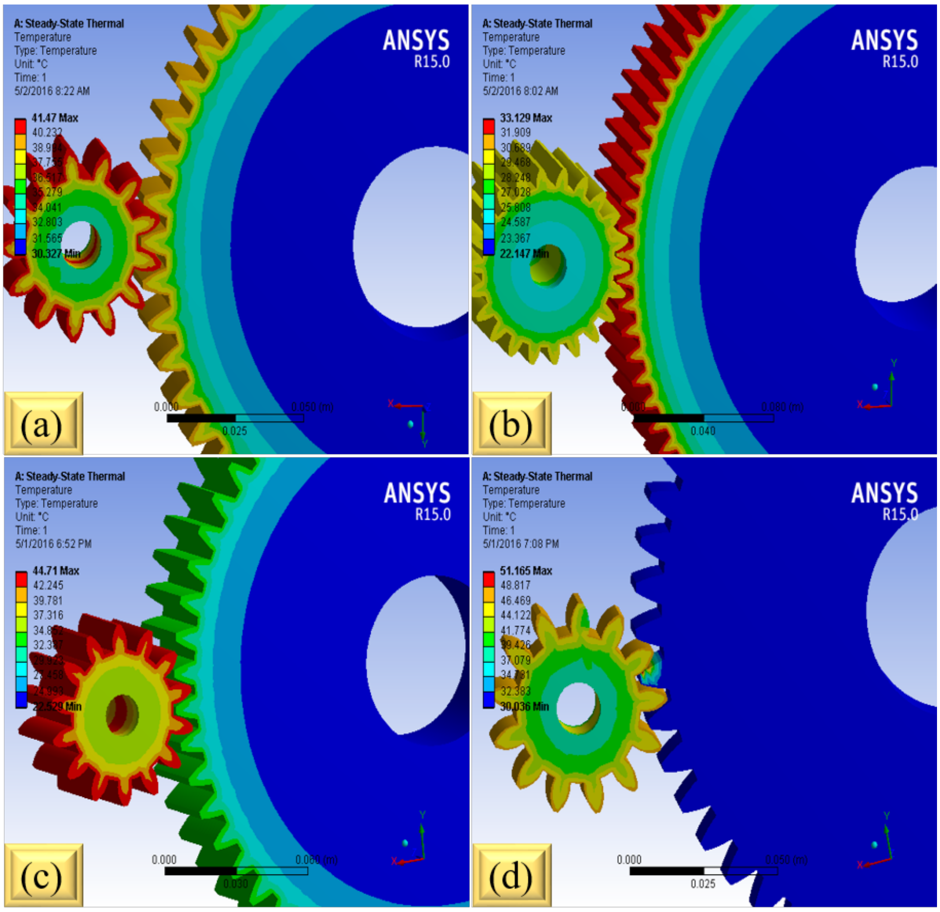

Figure 5 Temperature rise in (a) Design 1, (b) Design 2, (c) Design 3, and (d) Design 4

Table 8 ANSYS results

		Design 1	Design 2	Design 3	Design 4
Principal Stress (MPa)	min	14.25	4.22	76.726	37.03
	max	62.33	129.52	175.33	248.9
Shear Stress (MPa)	min	~0	~0	~0	~0
	max	28.12	73.95	89.99	193.15
Total Deformation (m)	min	0.000378	$2.64*10^{-5}$	$6.39*10^{-6}$	$5.24*10^{-5}$
	max	0.002236	0.000129	$3.68*10^{-5}$	0.0002294
Pinion Tooth Temperature (℃)	min	38.94	30	39.781	41.774
	max	41.47	31.909	44.71	48.817
Pinion Tooth Temperature (℃)	min	36.517	30	32.27	30
	max	40.23	33.129	34.852	46.47

6. DISCUSSION

The design was performed for gear mesh of polymer or reinforced polymer composite of a combination of Polycarbonate, ABS (+30% glass fibre), Acetal copolymer and Acetal copolymer (+30% glass fibre). All the designs were performed for power transmission of 5 kW @ 300 rpm and ambient condition of 300 C, and atmospheric pressure. Considering the strength of polymer combinations, the pinion and gear specifications were designed. The design specification includes the centre line distance, actual module, diameter of pinion and gear, and the required number of teeth on both wheels for each design.

It can be observed from Table 6 that the surface stress intensity and unit load are minimum for Design 2. On increasing the surface stress intensity, wear of tooth increases. On the other hand, on increasing the unit load, the chances of tooth breakage increases. So, according to the failure criteria, Design 2 is safer. Unit load and surface stress intensity are highest for Design 4, so Design 4 is more prone to failure. According to design according to failure criteria, the best combination of pinion and gear will be Nylon 66 and Acetal copolymer, respectively.

It can be seen from Table 7 that the temperature rise in pinion is also minimum for Design 2. At the same time, temperature rise in gear is minimum for Design 4. Temperature rise in gear for design 2 is also not very high (2.5℃). So, from a thermal failure point of view also, Design 2 is safer.

As per the parameters designed, the designs were analysed on the ANSYS for mechanical and thermal aspects. The maximum principal and maximum shear stress generated in the first design were the least for all designs. So, where the strength of the gear in terms of the stress is required for power transmission of 5 kW @ 300 rpm, then pinion can be made of Polycarbonate and gear of Acetal copolymer. The centre distance and size of the first design were also least, which shall require less space. The deformation of the second design was the least. So, where deformation is the constraint, then pinion can be made of Nylon 66 and gear of Acetal copolymer.

From thermal analysis, the surface temperature generation of the gear and pinion for the second design was the least. If the temperature is the constraint, then the pinion should be made of Nylon 66, and gear should be made of Acetal copolymer.

Suppose the operating temperature of gear is allowed up to 50-60℃. In that case, power transmission required is up to 5 kW @ 300 rpm, then Design 1 is the most feasible and economical design as the size and centre line distance is less; hence space requirement is least among the four designs. The Principal and shear stress generated in the tooth is the least. Design 1 shall be preferred where pinion is made of Polycarbonate and gear is made of Acetal copolymer.

If less heat generation is required, then Design 2 shall be preferred, which has the least total deformation for the same required power transmission of 5 kW @300 rpm. In this case, dimensions are larger than other designs as well centre distance is large, requiring more space than others. In this case, the pinion is made of Nylon 66, and the gear is made of Acetal copolymers.

So, Design 1 and Design 2 are preferable designs.

CONCLUSIONS

Following conclusions were drawn after analysing the designs analytically and using ANSYS:

- The size and centre line distance were least in the case of Design 1. Hence, when space requirement is a constrain, then pinion can be made of Polycarbonate and gear of Acetal copolymer.
- Surface stress intensity and unit load obtained were minimum for Design 2 (pinion of Nylon 66 and gear of Acetal copolymer). The best combination of material for pinion and gear is Nylon 66 and Acetal copolymer, respectively, to prevent tooth breakage and excessive wear.
- Temperature rise in pinion was minimum for Design 2 (pinion of Nylon 66), while the temperature rise in gear was minimum for Design 4 (gear of Nylon 66). So, Nylon 66 is the best material for making pinion and gear if only temperature rise is considered.
- The maximum principal and maximum shear stress generated in Design 1 were the least. So, where the strength of the gear in terms of the stress is the failure criteria, then pinion can be made of Polycarbonate and gear of Acetal copolymer.

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REFERENCES

[1] Z. Lu, H. Liu, R. Zhang, C. Zhu, Y. Shen and D. Xin, “The simulation and experiment research on contact fatigue performance of acetal gears,” *Mechanics of Materials*, vol. 154, p. 103719, 2021.

[2] P. K. Singh, K. Mausam and A. Islam, “Achieving better results for increasing strength and life time of gears in industries using various composite materials,” *Materials Today: Proceedings*, 2021.

[3] K. F. Wotodzo, K. A. Kassegne, D. Koffi, S. Tiem and A. Batako, “A numerical simulation of the influence of torque on the performance of composite gears,” *Materials Today: Proceedings*, 2021.

[4] B. H. Soudmand and K. S. Nezhad, “Experimental investigation on the durability and failure modes of polybutylene terephthalate/calcium carbonate nanocomposite gears,” *Engineering Failure Analysis*, vol. 120, p. 105113, 2021.

[5] A. N. Taywade and V. G. Arajpure, “Design and Development of Nylon 66 Plastic Helical Gears in Automobile Application,” *International Journal of Engineering Research & Technology*, vol. 3, no. 9, pp. 1330-1334, 2014.

[6] K. Gupta and S. Chatterjee, “Analysis of Design and Material Selection of a Spur gear pair for Solar Tracking Application,” *Materials Today: Proceedings*, vol. 5, pp. 789-795, 2018.

[7] A. K. Singh, Siddhartha and P. K. Singh, “Polymer spur gears behaviors under different loading conditions: A review,” *Journal of Engineering Tribology*, pp. 1-19, 2017.

[8] S. Senthilvelan and R. Gnanamoorthy, “Wear characteristics of injection-moulded unfilled and glass-filled nylon 6 spur gears,” *Journalof Engineering Tribology*, vol. 2018, pp. 495-502, 2004.

[9] A. K. Singh and Siddhartha, “Thermal and Wear Behavior of Glass Fiber-Filled Functionally Graded Material-Based Polyamide 66 Spur Gears Manufactured by a Novel Technique,” *Journal of Tribology*, vol. 140, pp. 1-16, 2018.

[10] N. Kumar, A. Bharti and M. K. Gupta, “Effect of Treatments on Thermo-mechanical Properties of Epoxy based Sisal Biocomposites,” *International Journal on Emerging Technologies*, vol. 11, no. 3, pp. 491-495, 2020.

[11] B. Cerne, M. Petkovsek, J. Duhovnik and J. Tavcar, “Thermo-mechanical modeling of polymer spur gears with experimental validation using high-speed infrared thermography,” *Mechanism and Machine Theory*, vol. 146, p. 103734, 2020.

[12] A. Yadav, “Different types Failure in gears-A Review,” International Journal of Science, *Engineering and Technology Research*, vol. 1, no. 5, pp. 86-92, 2012.

[13] Design Data (Data Book of Engineers), Coimbatore: Kalaikathir Achchangam.

[14] S. Beermann, “Estimation of lifetime for plastic gears,” Kiss Soft, Switzerland, 2007.

[15] R. Walter, "“Engineering Principles for Plastic Gears,” Gear Solution, 2004.

[16] "Design of Plastic Gears,“ KHK Stock Gears. [Online]. [Accessed 2 4 2021].