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Design and Analysis of Planetary Gearbox For All-Terrain Vehicle

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Abstract: Planetary Gear Trains are extensively used for power transmission and are the most critical component. This report focuses on the designing and analysis of a planetary gearbox for an SAE BAJA All-Terrain Vehicle (ATV). The conventional two-stage speed reduction gearbox is bulkier and has a high volume to power ratio. A planetary gearbox gives the best balance between weight and power to be transmitted. For the design procedure standard method is utilized. Based on conventional equations gear calculations are performed and subsequently, CAD modeling for various parts is done. The analysis for each component is performed and checked for its stresses. The transmission shafts and bearings are designed using the standard force equations. In addition, the design and analysis of gearbox casing is performed and the complete assembly is checked for interferences. This report also presents the advantages and limitations of planetary gear train over other transmission systems. The comparison of planetary gear system with compound spur gear system is done on the basis of volume and weight, ease in manufacturing, assembly and disassembly and aesthetical viewpoint.
Keywords: Planetary Gearbox, All-Terrain Vehicle (ATV), Volume to power ratio, Designing, CAD modeling, Analysis, Gearbox casing, Ease in manufacturing, Aesthetic

I. INTRODUCTION

Planetary gear Train also referred to as epicyclic gear Train consists of three elements sun gear, planet gear, and ring gear. Sun gear is located at the centre that transmits torque to planet gears orbiting around the sun gear. Both systems are located inside the ring gear. In the toothed formation, sun and planet gears are in external meshing while planet gears and ring gear are internally meshed. The planetary gear train is found in many variations and arrangements to meet a broad range of speed-ratio in the design requirements. Now, industrial applications demand high torque in a compact (a high torque/volume) and light (a high torque/weight ratio) package. In a planetary gear train, torque density can be increased by adding more planets through multiple gear mesh points. This means a planetary gear with say three planets can transfer three times the torque of a similar-sized fixed axis standard spur gear system. The applied load to planetary gears is distributed onto multiple gear mesh points means the load is supported by N contacts (where N = the number of planet gears) increasing the torsional stiffness of the gear train by factor N. Hence it lowers the lost motion compared to similar size standard gear trains. Planetary gears with deep groove ball bearings are frequently used in applications in which high radial loads occur. Hence, they are used to develop a hub drive system for an automobile. Automated Guided Vehicles and Mobile Satellite Receivers are other applications of a planetary gearbox.

II. DESIGN OF GEAR BOX

A. Determination of Speed Reduction Ratio For Planetary Gearbox

For proposed application, a four-stroke cycle, air-cooled engine would serve the purpose. This segment has considered Briggs & Stratton (B&S) 10 HP overhead valve engine (OHV) intake Engine as a prime mover/Powertrain. This engine develops a maximum torque of 18.98 N-m at 2800rpm and a peak power of 10 HP at 3800 rpm.

The main objective of the transmission is to provide to the drive more than enough torque to the wheels from the engine. Enough torque means torque required to pull the driving wheels against the road loads.

To choose the transmission capable of producing enough torque to propel the All-Terrain Vehicle (ATV), it is necessary to determine the total tractive effort (TTE) requirement of the vehicle.

Following are the design parameters selected for the vehicle

Gross vehicle weight (car + driver) 170+70 = 240 kg

Weight distribution ratio =0.6

Maximum engine torque 18.98 N-m

Engine power 10 Hp = 7.5 kW
Wheel diameter 0.534 m = 21 inches
CVT low reduction ratio =3.7
Power transmission efficiency =90%
Co-efficient of static friction 0.8 (sand) 0.9 (concrete)
Co-efficient of rolling resistance (Fr) =0.05 (sand)and 0.014 (concrete)
Air density =1.122 kg/m3
Frontal area of ATV =1.055 m2
Drag co-efficient= 0.44
Vehicle velocity (initially assumed)= 20 kmph
Total Tractive Effort (TTE) = RR+AR+GR
Where,
RR=Rolling Resistance= Gross Vehicle Weight x coefficient of rolling resistance
AR=Aerodynamic Resistance= $0.5\ x\ \rho\ x\ A\ x\ V^2\ x\ C_d$
Where
ρ = air density
A = frontal area of car
v = vehicle velocity
Cd = air drag coefficient
GR=Grade Resistance= Vehicle weight x sin (α)
Where α = inclination angle of plane
Now,
TEMax = μ x 0.6 x GVW x cos α
Where,
μ = Static Friction Coefficient
0.6 = Rear Weight Distribution
Torque required at wheels is,
Tw = TE_{max} x r
To calculate speed reduction ratio.
Net reduction ratio = output torque input torque x η
Where, η = power transmission efficiency
Our power transmission layout includes the use of a continuous variable transmission or CVT to further provide speed reduction and torque multiplication.
This CVT has fixed specifications as per given by the manufacturer and we use a Just In Time CVT that has a speed reduction ratio of 3.7
Therefore,
Speed reduction ratio, i = net reduction ratio/3.7
By performing above calculations, we get,
Rolling Resistance =117.72 N (for sand)
Aerodynamic Resistance =8.038 N
Grade Resistance =995.012 N
Total Tractive Effort =1120.77 N
Maximum Tractive Effort =1194.934 N
Torque at the wheels =320 N-m
Net reduction ratio =18.7
Final Speed reduction ratio= 5

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B. Design Of Planetary Gear Train Material Selection

20MnCr5 is Chrome-Manganese Carburizing Steel generally supplied annealed to HB 229(max). Carburized and heat-treated, it develops a hard wear resistant case of HRC 60-63 and a tough strong core with a typical tensile strength range of 900-1300 MPa. This material can be easily acquired by vendors and can undergo the required heat treatment for our application. It is low alloy steel capable of being hardened which provides a tough core and can be case hardened by carburizing or carbonitriding to achieve a hard case

We choose the following parameters for sun gear:

Pressure angle, $\emptyset = 20°$

Module, m= 2 mm

Number of teeth, Z_s =20 (as per AGMA standards)

For epicyclic gear train following criteria should be satisfied:

$m\ (Z_p+2) < m(Z_p + Zs)\ (\mathbf{sin}\ \boldsymbol{\pi}/n)$

Where,

n is Number of planet gear

Number of teeth on sun gear = Z_s

Number of teeth on planet gear = Z_p

Number of teeth on ring gear = Z_r

Therefore,

L.H.S: 2(30+2) = 64

R.H.S: 2(20+30) sin (π/2) = 100

L.H.S < R.H.S

Hence, above criteria is satisfied.

Since i=5 and Z_s=20

According to i = Z_s+Z_r/Z_s

Z_r = 80

$\boldsymbol{Z_p} = (\mathbf{Z_s}*\mathbf{i} - \mathbf{2}*\mathbf{Z_s})\ \mathbf{2} = \mathbf{30}$

m = PCD/z

By using above equation,

Pitch Circle Diameter of sun gear: 40 mm

Pitch Circle Diameter of planet gear: 60 mm

Pitch Circle Diameter of ring gear: 160 mm

For Calculation of Speeds,

α = Rotation of sun gear

β = Rotation of carrier

(+ and – signs are used in below table to distinguish between the direction of rotation)

Sr. No.	Description	Carrier	Sun Gear	Planet Gear	Ring Gear
1	Rotate sun gear by α, while holding carrier fixed	0	α	$-\frac{Z_S}{Z_P}*\alpha$	$-\frac{Z_S}{Z_R}*\alpha$
2	System is fixed and carrier is rotated by β	+β	+β	+β	+β
3	Final speed	β	α+β	$-\frac{Z_S}{Z_P}*\alpha+\beta$	$-\frac{Z_S}{Z_R}*\alpha+\beta$

In our case Ring gear is stationary hence, Its RPM is 0. …(1) – $(z_S/z_P)*\alpha+\beta=0$

We know the maximum engine speed hence, Speed at the sun gear,

Engine Speed$*\eta_{CVT}$ / CVT reduction = 3800*0.65/0.9 = 2744 rpm

Hence, α + β = 2744 rpm …(2)

Solving above two equation, we get Taking equations from above table and finding rpm for each component we get

Planet Speed = 915 rpm

Carrier speed = 548.8 rpm

Sun Speed = 2744 rpm

Sr No.	Type	Number of teeth	Diameter	RPM (max)
1	Sun	20	40	2744
2	Planet	30	60	915
3	Ring	80	160	0
4	Carrier	0	0	548.8

Tangential Load (F_t) is important because it determines the magnitude of torque and consequently the power, which is transmitted.
$Power\ (P) = Tangential\ Load\ (F_t\ )x\ pitch\ line\ velocity\ (v)$
$pitch\ line\ velocity\ (v) = \pi xDxN/60 = \pi$ x 40 x 2800 3.7/60 x 1000 = 1.58 m/s
Where,
D = pitch circle diameter of sun gear
N = RPM of sun gear at maximum torque condition
We know that, P = 7.5 kW
Therefore, we calculate, $F_t = P/v$ = 7.5 x 1000/1.58
F_t = 4734.42 N
Allowable Stresses
Step 1: Lewis Equation
$F_t = \sigma_w\ \ P_c x\ y$ x b
Where,
σ_w = Allowable Stress on Gear
P_c = Circular pitch = πm = 6.28 mm
y = Form Factor based on Tooth Profile and number of Teeth = 0.154 − 0.912 Zs = 0.1084
b = Face Width of Gear First,
we need to find the Allowable Stress by following the further mentioned steps:
σ_u = Ultimate strength of material = 1200 MPa
FOS = factor of safety = 2
σ_d = safe ultimate stress = σ_u/FOS = 1200/2 = 600 MPa
C_v = velocity factor = 6/(6 + v) = 0.8
$\sigma_w = \sigma_d$ x C_v = 480 MPa
We use the Lewis Equation to find the face width of the gear. b = 20 mm
Step 2: Beam Strength (F_s)
It is the maximum value of tangential force that the tooth can transmit without bending failure. It is determined by using the Lewis Equation which assumes the tooth as a cantilever beam. After finding face width (b), we can calculate the beam strength of the gear by considering the gear tooth as a cantilever beam under tangential load.
$F_s = \sigma_d$ x P_c x y x b = 600 x 6.283 x 0.1084 x 20
$\boldsymbol{F_s}$ = 8169 N
Step 3: Dynamic Load (F_d)
The gears rotate at an appreciable speed and it becomes necessary to consider the dynamic force resulting from the impact between mating teeth. It is induced due to the following factors:- •

- Inaccuracies of tooth profile
- Errors in tooth spacing
- Misalignment between bearings

It is the sum of tangential load (F_t) and dynamic induced load (F_i).
To calculate Fi;
$F_i = [21v \text{ x } (bCe+Ft)] / [21v + \sqrt{(bCe+Ft)}]$
Where, C = deformation factor = $0.111 / ((1/E_s) + (1/E_p))$
E = young's modulus of sun and planet gear
$E_s = E_p = 2 \text{ x } 10^5$ N/mm^2
Therefore, C = 11100 MPa
e = error occurred during gear manufacturing = 10.205 mm (from PSG data book)
Hence, we can calculate, Fi = 1991.462 N
To find dynamic load $F_d = F_t + F_i = 4734.418 + 1991.462$ N $F_d = 6725.879$ N
STEP 4: Wear Strength (F_w)
It is the maximum value of tangential force that the tooth can transmit without pitting failure. Pitting occurs when the contact stress reaches the magnitude of the surface endurance strength.
$F_W = D \text{ x } b \text{ x } Q \text{ x } k$
Where, D = pitch circle diameter of sun gear = 40mm
b = face width = 20mm
Q = ratio factor = $2i / (i+1)$
Where, i = Speed reduction ratio
Hence, Q = 1.667

$$= \frac{\sigma_s 2}{1.4} \sin\emptyset \; x \; \cos\emptyset \; x \; \left(\frac{1}{E_s} + \frac{1}{E_p}\right)$$

Here, σ_s = surface endurance limit

$= 0.27 \; x \; 9.81 \; x$ (BHN)

$= 0.27 \; x \; 9.81 \; x \; 601$ (60 HRC = 601 BHN)

$\sigma_s = 1592$ MPa

$\emptyset$ = pressure angle = 20°

Substituting the required values, we get k = 5.82

k = Load stress factor =
Therefore, $F_w = 40 \text{ x } 20 \text{ x } 1.667 \text{ x } 5.82$
$F_w = 7756.444$ N
STEP 5: Criteria of safety
After all the necessary loads are found, we have to determine if the gear can sustain these loads without bending and wear failure.
For this purpose, we use the following criteria to ensure a safe design:

- Static load (or beam strength) should be greater than dynamic load- $F_s > 1.2 F_d$
- Wear load should be greater than dynamic load- $F_w > F_d$

For shaft and bearing design standard procedures have been followed along with design data book.

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Exploded View of Planetary Gear box

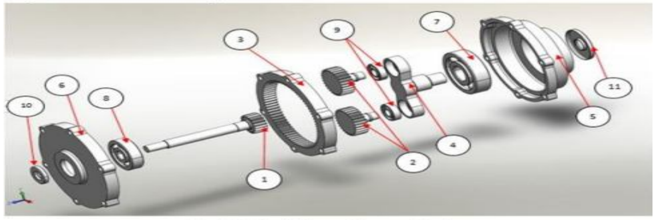

Fig. 27: Exploded View of Planetary Gearbox

Part Number	Part Name
1	Sun Gear
2	Planet Gears
3	Ring Gear
4	Planet Carrier
5	Right Casing
6	Left Casing
7	Carrier Shaft Bearing
8	Sun Gear Shaft Bearing
9	Planet Gear Shaft Bearing
10	Oil seal for Left Casing
11	Oil seal for Right Casing

Figure 1 Planetary Gearbox (without Casing)

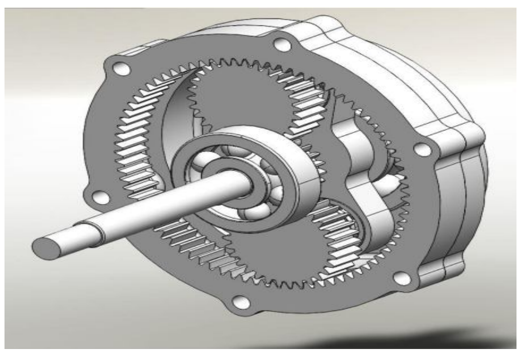

Figure 2 Exploded View of Planetary Gearbox

III. STRESS ANALYSIS WITH ANSYS 2021 R1

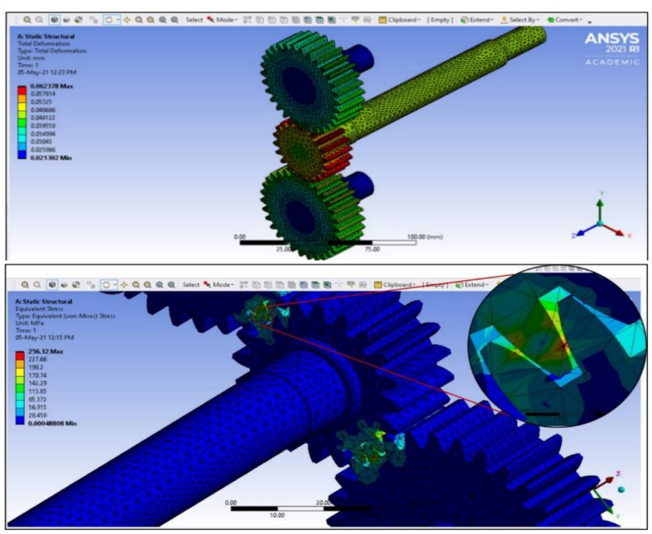

Figure 3 Sun Gear and Planet Gear Analysis

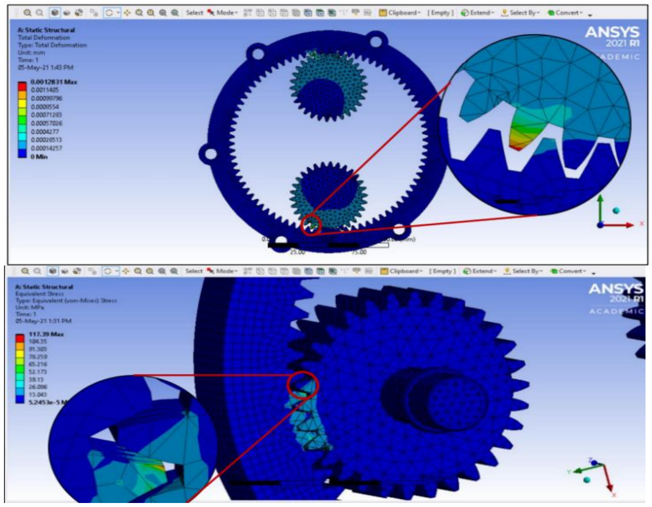

Figure 4 Ring and Planet Gear Analysis

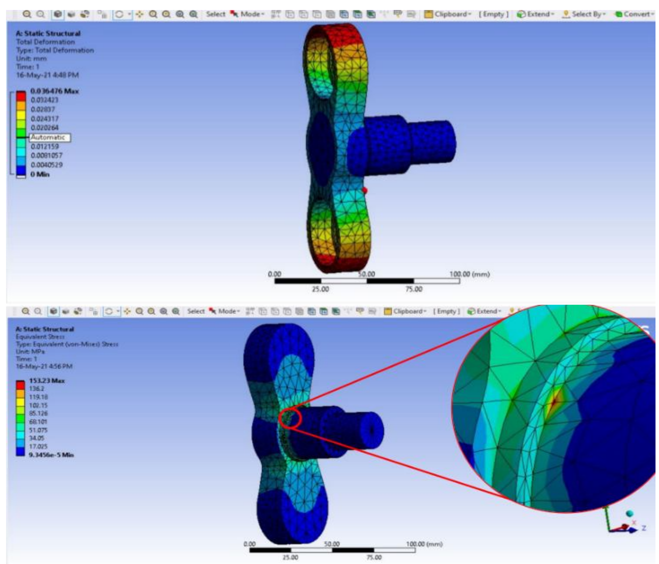

Figure 5 Carrier Analysis

IV. RESULTS

1) The main objective of this project was to design a planetary gearbox instead of going for conventional compound spur gearbox for All-Terrain Vehicle (ATV) as it offers expediency over numerous parameters. These are tabulated below

Checking Safety Of Design

Sr. No	Component	Permissible Stress (MPa)	Maximum Induced Stress (MPa)	Maximum Induced Deformation (mm)	Design Remark
1	Sun Gear and Planet Gear	350	256.12	0.0623	SAFE
2	Ring Gear and Planet Gear	350	117.39	0.00128	SAFE
3	Carrier	350	153.23	0.0365	SAFE

Figure 6 Checking safety of design

Sr. No.	Parameter	Compound Gearbox	Planetary Gearbox
1	Maximum Height	280.5 mm	200 mm
2	Maximum Width	92.5 mm	106.5 mm
3	Maximum Breadth	150 mm	200 mm
4	Total Volume	0.0011 m^3	0.000995 m^3
5	Total weight	9.6 Kg	6.829 Kg
6	Number of bearings used	6	4

2) Above table explicitly shows that our designed planetary gearbox is much better than the existing compound gearbox.
3) The designed planetary gear train is 10% smaller in size and 2.8 kg lighter than the compound spur gear train. At the same time number of bearings used is reduced significantly.

V. CONCLUSION

1) The purpose of this project was to design a planetary gear train suitable for All-Terrain Vehicle (ATV) where space and weight constraints were the prime objectives.
2) In the designed planetary gearbox reduction ratio of 5:1 is achieved in a single stage rather than two stage in the compound spur gearbox for the same reduction ratio.
3) The weight of the planetary gear train is 2.8 kg less than the compound spur gearbox while the size reduction attained is 10%.
4) The proposed planetary gear train successfully meets the intended objectives.

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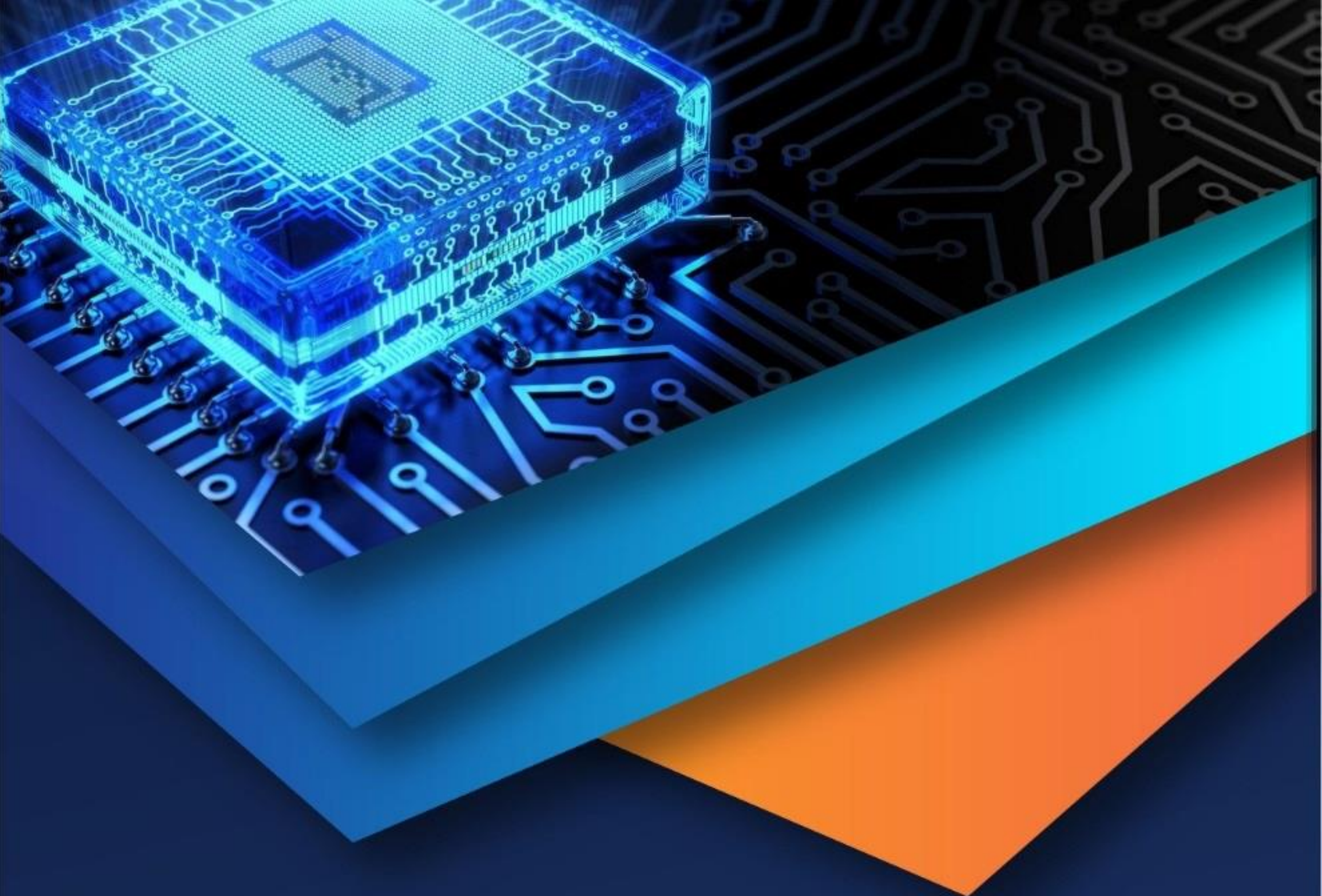

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