Available online at: http://www.ijmtst.com/vol5issue08.html
International Journal for Modern Trends in Science and Technology
ISSN: 2455-3778 :: Volume: 05, Issue No: 08, August 2019

Material Selection for Optimum Design of Planetary Gear Train used in Automobile Gear Box

G Pardhiv[1] | P Srinivas[2]

[1]Department of Mechanical Engineering, Srinivasa Institute of Engineering and Technology, Cheyyeru, Amalapuram, India.
[2]Associate Professor, Department of Mechanical Engineering, Srinivasa Institute of Engineering and Technology, Cheyyeru, Amalapuram, India.

To Cite this Article
G Pardhiv and P Srinivas, "Material Selection for Optimum Design of Planetary Gear Train used in Automobile Gear Box", *International Journal for Modern Trends in Science and Technology*, Vol. 05, Issue 08, August 2019, pp.-48-59.

Article Info
Received on 13-July-2019, Revised on 02-August-2019, Accepted on 05-August-2019, Published on 06-August-2019.

ABSTRACT

Planetary Gear Trains are used in different applications like power transmitting systems of machine tools and so on. Planetary gear boxes are used frequently to match the inertias, lower the motor speed, boost the Torque, and at the same time provide a study mechanical interface for pulleys, cams, drums and other mechanical components. Gears in planetary gear boxes sometimes get short life due to wear and breakage by repetitive load during operation time. Even if failure of a single tooth of a gear will make the machine to stop. Hence our aim is to strengthen the gear which is a key element of gear box. In this project I needed to suggest appropriate gear material by considering its Young's modulus, Yield Strength, Hardness, coefficient of friction. The materials utilized for gears are nodular graphite cast iron. The material properties and costing of pinion and gear material were studied, and standard gear materials were identified from PSG Design Data Book. The selection of the gear materials is based on some multi criteria decision making methods (MCDM) like TOPSIS, VIKOR and Weighted properties. The material or gear has selected by using above mentioned criteria methods. After the material selection than a planetary gear set has designed by convectional method and MAT Lab coding is generated within allowable bending and compressive strength. Then comparison is done to predict the Weight and Volume of the both gear trains. Then a motion analysis on currently used material and the selected material of gear train. The modelling and motion analysis is done by using the solid works.

KEYWORDS: *Gearbox, Materials, MCDM, Motion analysis, MAT Lab®, Solid works*

I. Introduction

1.1 INTRODUCTION

Planetary gear trains are utilized in different applications. These gear systems are in high demand for efficient manufacturing in advance industrial applications and automations such as power transmission in process equipment, material handling equipment, metal cutting and metal forming machineries etc. Planetary gear systems are capable of transmitting the highest torque with the highest performance and high reduction ratios. In this project we consider automobile gear box in this The planetary gear systems parts are likely to wear and breakage during its normal working cycle by repetitive load in operations. So to oppose this loads the material ought to have some particular properties, for example, Young's modulus, Yield Strength, Hardness, less coefficient of friction and the material expense of the planetary riggings ought to likewise have low qualities so as to give an upper hand among makers. There is no material which satisfies every one of these needs. Every material has distinctive execution for every property. Accordingly, it is important to choose the best elective material that has the most noteworthy level of fulfilment for all the applicable properties.

1.2 PROBLEM STATEMENT

In planetary gear train, the gears should have some specific properties in order to maintain their function during working. In automobile various loads induced on the gears during working, These loads may cause failure to the gear, so to resist failure of gears the gear should meet the following requirements, i.e.; Young's modulus (YM) for high rigidity., Ultimate tensile strength (UT) to prevent failure against static loads., Fatigue strength (FS) to withstand dynamic loads, Low coefficient of friction (CF) and gears should have sufficient hardness (H),there are some other property such as cost(C). The low value of which is desired in order to provide competitive advantage among manufactures. Nine alternative gear materials were taken into consideration: SAE8620, 20MnCr5, 16Mncr5, Nylon, Aluminium, SAE9255, Aluminium Bronze, SCM440, SUS316.The properties required for planetary gears and the materials with their quantitative data are given in Table1.

However, none of the proposed materials met the previously referenced needs. Some material choice techniques have been immersed, so as to choose the best material that has most noteworthy level of fulfilment for all the significant properties. MCDM strategies, i.e; TOPSIS, VIKOR, Weighted properties technique's were utilized in our investigation to assess conceivable material for gears. A traded off gauging strategy made out of AHP and entropy techniques were utilized to decide the criteria loads.

Material	Ultimate Tensile Strength(MPA)	Young's Modulus (GPA)	Hardness	Fatigue Strength (MPA)	Coefficient of Friction	Cost (Rs/kg)
SAE 8620	1157	205	361	590	0.9	68
20MnCr5	692	210	192	292	0.5	75
16MnCr5	1158	190	170	490	0.7	60
Nylon	90	4	88	40	0.43	200
Aluminium	241	69	15	68.9	1.35	131
SAE9255(A)	770	205	241	275	0.65	72
Aluminium Bronze	690	117	327	152	0.4	800
SCM440	980	190	207	490	0.6	70
SUS 316	520	205	152	260	0.25	200

Table 1.1 The required properties for planetary gears and the Materials with their quantitative data

Then a design of planetary spur gear set to transmit 50KW at a pinion speed of 1400rpm. The transmission ratio is 5. For the material which was selected by MCDM technique. And with currently used material nodular graphite cast iron.

After selecting the gear material by MCDM a planetary spur gear set is designed with 50Kw power at a speed of 1400 rpm with a transmission ratio of 5. And at the same time the same designing is made with the existing material i.e Nodular graphite Cast Iron. The main motto is to design the gear set with the both material is to compare weight and volume of the gear set.

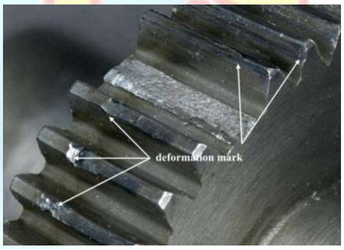


Figure 1.1 Gear teeth deformation

1.3 PLANETARY GEARBOX

Planetary gear box works on planetary motion principle each phase of the planetary gear box comprises of a focal Sun Gear coinciding with precisely situated three Planet Gears around it which thusly work with the inner teeth of the external Ring Gear. Ordinarily, the Ring gear is stationary and forms the part of the housing, input is given to the sun gear and output is gotten from the three planet gears through a planet carrier.

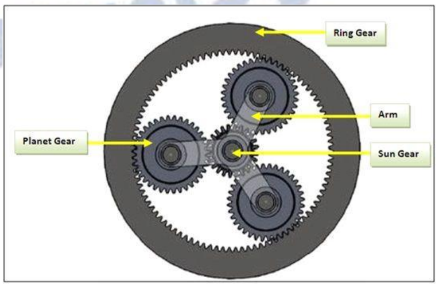


Figure 1.2 Planetary gear box

1.3.1 WORKING OF PLANETARY GEAR SET

Any of the three individuals can be utilized as the driving or input member .In the same time, another part may be kept from rotating and in this manner turns into the response, held, or stationary part. The third part at that point turns into the driven or output part.

Planetary gears are found in numerous varieties and game plans so as to meet an expansive scope of speed proportions in the design requirements. Various designs can be effectively gotten by re-masterminding input part, external part and stationary part, as described in Table I.

S.no	Sun Gear	Carrier	Ring Gear	Speed	Torque	Directions
1	Input	Output	Held	Maximum reduction	Increase	Same as input
2	Held	Output	Input	Minimum reduction	Increase	Same as input
3	Output	Input	Held	Maximum increase	Reduction	Same as input
4	Held	Input	Output	Minimum increase	Reduction	Same as input
5	Input	Held	Output	Reduction	Increase	Reverse of input
6	Output	Held	Input	Increase	Reduction	Reverse of input
7	When any two members are held together, speed and direction are the same as at input. Direct 1:1 drive occurs.					
8	When no member is held or locked together, output cannot occur. The result is neutral condition.					

Table 1.2 Planetary gear train combinations.

1.3.2 ADVANTAGES OF PLANETARY GEARBOX

- Good quality bearings for input and output shafts.
- High efficiency.
- Low noise level.
- No oil leakages.
- Taper roller bearings on output shafts for bigger models.
- Long and trouble free performance.

1.3.3 DISADVANTAGES OF PLANETARY GEARBOX

- More expensive than conventional production of gearboxes.
- More complex than conventional transmissions.

A. 1.3.4 APPLICATIONS

- Planetary gears are used in wheel hub drive in Automated Guided Vehicles (AVG).
- Planetary gears are used in axis drive for laser cutting machines.
- Planetary gears are used in angle gearbox for rotary tables.
- Planetary gears are used in industrial dough mixer.
- Planetary gears are used in height adjustment of operating tables.
- Planetary gears are used in curved stair lifts.
- Planetary gears are used in mobile satellite receivers.

1.4 THESIS OUTLINE

The first chapter deals with the introduction, problem statement, and aims of the project. The second chapter gives the literature review. The third chapter deals with material selection. The fourth chapter deals with theoretical design of planetary gears and also with MAT Lab coding. The fourth chapter deals with modelling and motion analysis of planetary gear train. The sixth chapter details the conclusions and future work.

II. LITERATURE REVIEW

2.1 LITERATURE REVIEW

Halil Calıs_kan. [1] In his work described the material selection problem for the tool holder working under hard milling conditions has been solved utilizing a decision model. The model includes the EXPROM2, TOPSIS and VIKOR methods for the ranking of the alternative materials according to determined criteria.

Hüseyin Filiza, S. Olgunera [2] In his work For a specific PGT configuration, design parameters of the gears involved are determined by going through the optimization study. Objective was to minimize kinetic energy of gear trains which would satisfy geometric and kinematic constraints, together with constraints on the failure of gear teeth by bending and surface contact.

Mohan and Seshaiah [3] carried out a study related with the minimization of center distance,weight and tooth deflection of spur gear set by using genetic algorithm. Contact stress and bending stress equations were used as constraint equations and multi-objective function optimization was done. Module, facewidth and number of teeth of gears are parameters to be obtained after optimization process.

Golabi et al. [4] carried out a study to minimize the volume to weight ratio of the gearbox considering one, two and three-stage gear trains. The general form of objective function and design constraints for the volume to weight ratio of a gearbox were written. Practical graphs of the results of the optimization were presented, by choosing different values for the input power, gear ratio and hardness of gears. From the graphs, all necessary parameters of the gearbox, such as number of stages, modules, face width of gears, and shaft diameter were obtained.

Tamboli et al. [5] carried out a study to minimize the volume, since the most power transmission systems require low-weight energy-efficient and cost-effective system elements. A helical gear pair of a heavy duty gear reducer was considered for the objective of minimum volume. The various factors for sizing and strength of gears were computed for gear geometry parameters using DIN standard. The solution was attempted by using particle swarm optimization. The results are satisfactory and help the designer to employ minimum material and cost,by fulfilling the strength and performance requirements.

J.Stefanović Marjanovic [6] studied practical approach to the optimization of planetary gear trains with ―spur gears‖ In his study he presented the selection of optimal gear trains and also selection of optimal position for shaft axes of gears trains with ―spur gears‖. In this the volume of the planetary gear trains with ―spur gears‖ is drastically reduced by 23%.

Likhitha R Reddy [7] In his work described a generalized C program code was written for computing shaft diameter for simply supported shafts on bearings with any number of gears and pulleys. Thus, use of programming for design of shafts reduces the need of tedious calculations, computation time and eliminates calculation errors.

Syed Ibrahim Dilawer [8] his study was carried out in this research shows the optimization analysis of the epicyclic gear train in INDIA to reduce load failure. The analysis is restricted to the optimization of gear train through load analysis of the gears, pinions and annulus including the sun and plant gears, and finding out the optimal load conditions for the gear train to perform effectively without leading to load failure.

III. MATERIAL SELECTION

3.1 Introduction

Planetary gear trains are utilized in different applications. These gear systems are in high demand for efficient manufacturing in advance industrial applications and automations such as power transmission in process equipment, material handling equipment, metal cutting and metal forming machineries etc. Planetary gear train are fit for transmitting the most elevated torque with the most noteworthy execution and high decrease proportions. In this paper we consider car gear enclose this The planetary gear train parts are probably going to wear and breakage amid its typical working cycle by redundant burden in activities. So to oppose this heaps the material ought to have some particular properties, for example, Young's modulus ,Yield quality, Hardness, less coefficient of contact and the material expense of the planetary riggings ought to likewise have low qualities so as to give an upper hand among makers. There is no material which satisfies every one of these needs. Every material has distinctive execution for every property. Accordingly, it is important to choose the best elective material that has the most noteworthy level of fulfilment for all the applicable properties.

Material choice has extraordinary significance in plan and improvement of the items. The achievement and aggressiveness of the makers additionally relies upon the chose material. The destinations of execution, cost and ecological affectability drive building structure, and are commonly restricted by materials. Choice of the materials that best meet the prerequisites of the structure and give most extreme execution and least expense is the objective of ideal item plan . Nonetheless, some clashing circumstances are commonly seen between these destinations and criteria (for example youthful modulus/cost, or durability/hardness) and there is a need to choose which property could easily compare to other people. Utilizing basic and intelligent techniques, the criteria that impact material determination for a given designing application must be recognized to kill inadmissible options and to choose the most suitable one.

3.2 Multi-criteria decision making methods

3.2.1 AHP strategy

The AHP strategy comprises of following advances

Developing a various levelled structure with an objective at the top dimension, the qualities/criteria at the second dimension and the choices at the third dimension

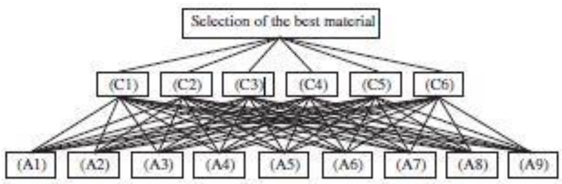
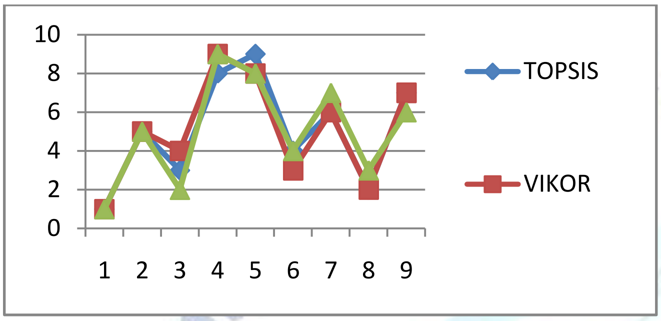


Fig 3.1 The decision hierarchy of material selection for the gears.

Decide the overall significance of various characteristics or criteria as for the objective, the pair insightful correlation framework is utilized and it tends to be made with the assistance of size of relative significance. 1 for "equivalent significance", 3 for "moderate significance", 5 for "solid significance", 7 for "solid significance", 9 for "outrageous significance", and 2,4,6,8 for "middle of the road values",1/3,1/5,1/7,1/9 for opposite examination. The pair savvy correlation network for given criteria is appeared Table 3.1.



Graph 3.1 Comparison of rankings obtained with the MCDM methods.

IV. DESIGN OF PLANITARY GEAR TRAIN

In this chapter we are going to design planetary gears to transmit 50kW power at pinion speed of 4000rpm.The transmission ratio is 5 for the materials SAE8620 and Ductile (or) nodular Iron

4.1 a THEORITICAL DESIGN FOR DUCTILE IRON MATERIAL GEARS

Power to be transmitted, P = 50kW = 50000W

Pinion speed n1 =4000rpm

Gear ratio I =5

4.1.1 a Material selection

The material used here is Ductile (or) nodular Iron Which was currently used gear box material

Material	Compressive strength(σ_c) in N/mm^2	Bending strength(σ_b) in N/mm^2
Ductile (or) nodular Iron	560	170

Table 4.1 Ductile (or) modular Iron Material Properties

4.1.2 a Minimum centre distance

Minimum centre distance based on surface compressive strength is given by

$$a \geq (i+1)\left[\left\{\frac{0.74}{\sigma c}\right\}^2 \frac{E[Mt]}{i\Psi}\right]^{1/3}$$ for the pressure angle α=20⁰

where E = Equivalent Youngs modulus

= 1.46 x 10^5 N/mm^2

σ_c = 560 N/mm^2

I = 5

$\Psi = \frac{b}{a}$ = 0.3 (Assumed initially)

$[M_t] = M_t * k*k_d$

Let $k*k_d$ = 1.3 initially

Now $M_t = \frac{60p}{2\pi n1}$

$M_t = \frac{60*50000}{2*\pi*4000}$

= 119.37 N-m

=119.37*10^3N-mm

$[M_t]$ =119.37*10^3*1.3

=155.181*10^3N-mm

Substituting all the values in the equation for "a" we get

$$a \geq (5+1)\left[\left\{\frac{0.74}{560}\right\}^2 \frac{1.46*10^5*155.181*10^3}{5*0.3}\right]^{1/3}$$

a≥178.59 mm

4.1.3 a Minimum module

Minimum module required based on beam strength is given by

$$m \geq 1.26\left[\frac{[Mt]}{[\sigma b]\Psi m Z1 y}\right]^{1/3}$$

where [Mt] = 155.181*$10^3$$N/mm^2$

σ_b= 170 N/mm^2

$\Psi_m = \frac{b}{m}$ =10 assumed

Z_1 = 20 assumed

Y =0.389 coresponding to Z_1 = 20

Then $m \geq 1.26\left[\frac{155.181*10^3}{170*10*20*0.389}\right]^{1/3}$

m≥2.27mm

Take m = 3

Now number of teeth of planet gear is corrected as

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

$Z_1 = \frac{2*178.59}{3*(5+1)}$

Z_1 = 19.83 Take Z_1=20

Hence Number of Teeth on sun gear is $Z_2 = i*Z_1$

=5*20 =100

And Number of Teeth on Ring gear is $Z_3 = 2*Z_1+Z_2$

=2*20+100 =140

4.1.4 a Pitch circle diameters

Pitch circle diameter of planet gear is $d_1 = m*Z_1$ = 60mm

Pitch circle diameter of sun gear is $d_2 = m*Z_2$ = 300mm

Pitch circle diameter of planet gear is $d_3 = m*Z_3$ = 420mm

4.1.5 a Corresponding centre distance

The corrected centre distance is

$a = \frac{d1+d2}{2} = \frac{60+300}{2}$ = 180mm

or we may use

a=$\frac{d3-d1}{2}$ = $\frac{420-60}{2}$ =180mm

Since this value is more than the required minimum centre distance hence the design is safe.

4.1.6 a Face width

Now, face width b = Ψ*a =0.3*180 =54mm

(or)b = Ψ_m*m = 10*2 =30mm

Take larger of these 2 values hence face width is b=54mm

4.1.7 a Checking of induced compressive stress and bending stress

The induced surface compressive stress

$$\sigma_c = 0.74\left\{\frac{i+1}{a}\right\}\left[\frac{(i+1)*E*[Mt]}{i*b}\right]^{1/2}$$

$$\sigma_c = 0.74\left\{\frac{5+1}{180}\right\}\left[\frac{(5+1)*1.46*10^5*155.181*10^3}{5*54}\right]^{1/2}$$

σ_c= 553.47 N/mm² Which is less than the allowable compressive stress for the material

The induced bending stress,

$$\sigma_b = \frac{(i+1)*[Mt]}{a*m*b*y}$$

$$\sigma_b = \frac{(5+1)*155.181*10^3}{180*3*54*0.402}$$

σ_b=79.43 N/mm²

Which is less than the allowable bending stress for the material Since both σ_c and σ_b are less than their design values, our design is correct

4.2 a DESIGN USING MATLAB COADING

The design process is very time consuming and made some calculation errors also in this a generalized MAT Lab code is written to design planetary gears for any material just by changing material properties at any speed and power.

```
%% Design of Planetary gears for SAE 8620 material
P=50000;        %% P= Power to be transmitted in Watts.
n1=4000;        %%n1=Pinion speed
i=5;            %%i=Gear ratio
Sc=560;         %%Sc=Allowable compressive stress
Sb=170;         %%Sb=Allowable bending stress
E=1.46*10^5;    %%E=Young's modulus
%% Minimum Centre distance
Si=0.3;         %%Assumed initially
mt=(60000*P)/(2*pi*n1);
k=1.3;          %%assumed
Mt=mt*k;
A=(i+1)*(((0.74/Sc)^2)*((E*Mt)/(i*Si)))^(1/3);
%% Minimum module
Sm=10;          %% Assumed
Z1=20;          %%Z1= number of teeth on pinion assumed initially
Y=0.389;        %%constant value
M=1.26*((Mt)/(Sb*Sm*Z1*Y))^(1/3);
m=round(M)      %%m=module
%% Now number of teeth on pinion is corrected as
Z=(2*A)/(m*(i+1));
Z1=round((Z-2)/2)*2+4
Z2=i*Z1         %%Z2=Sun gear teeth
Z3=2*Z1+Z2      %%Z3=Ring gear teeth
%% Pitch circle diameter
d1=m*Z1         %% PCD of Pinion
d2=m*Z2         %% PCD of Sun Gear
d3=m*Z3         %% PCD of Ring Gear
%% Corresponding centre distance
a=(d1+d2)/2
%% Face width
b1=Si*a;
b2=Sm*m;
b=max(b1,b2)
%% Checking of induced compressive stress and bending stress
y=0.402;
Sc=0.74*((i+1)/(a))*(((i+1)/(i*b))*E*Mt)^(1/2)
Sb=(((i+1)*(Mt))/(a*m*b*y))
```

After putting this code into the editor window in MAT Lab then click on run then save it as design.m file then in command window resuts are obtained i.e;

```
>> Design
m = 3
Z1 =20
Z2 =100
Z3 =140
d1 = 60
d2 =300
d3 =420
a =180
b =54
Sc = 553.4686
Sb =79.4260
```

Now we design the gears by using SAE8620 material which was obtained from MCDM techniques

4.1 b THEORITICAL DESIGN FOR SAE8620 MATERIAL GEARS

Power to be transmitted, P = 50kW = 50000W

Pinion speed n1 =4000rpm

Gear ratio I =5

4.1.1 b Material selection

The material selection process was explained in chapter3.The best material is SAE8620 and their compressive and bending strengths are tabulated below

Material	**Compressive strength(σ_c) in**	**Bending strength(σ_b) in**

	N/mm^2	N/mm^2
SAE8620	870	245

Table 4.2 SAE 8620 Material Properties

4.1.2 b Minimum centre distance

Minimum centre distance based on surface compressive strength is given by

$$a \geq (i+1)\left[\left\{\frac{0.74}{\sigma c}\right\}^{2} \frac{E[Mt]}{i\Psi}\right]^{1/3}$$ for the pressure angle $\alpha=20^0$

where E = Equivalent Youngs modulus

= $2.15 \times 10^5\ N/mm^2$

$\sigma_c = 870\ N/mm^2$

I = 5

$\Psi = \frac{b}{a} = 0.3$ (Assumed initially)

$[M_t] = M_t * k*k_d$

Let $k*k_d$ = 1.3 initially

Now $M_t = \frac{60p}{2\pi n1}$

$M_t = \frac{60*50000}{2*\pi*4000}$

= 119.37 N-m

$=119.37*10^3$N-mm

$[M_t] = 119.37*10^3*1.3$

$=155.181*10^3$N-mm

Substituting all the values in the equation for “a” we get

$$a \geq (i+1)\left[\left\{\frac{0.74}{870}\right\}^{2} \frac{2.15*10^5*155.181*10^{\wedge}3}{5*0.3}\right]^{1/3}$$

a≥151.47 mm

4.1.3 b Minimum module

Minimum module required based on beam strength is given by

$$m \geq 1.26\left[\frac{[Mt]}{[\sigma b]\Psi mZ1y}\right]^{1/3}$$

where [Mt] = $155.181*10^3 N/mm^2$

$\sigma_b = 245\ N/mm^2$

$\Psi_m = \frac{b}{m} = 10$ assumed

Z_1 = 20 assumed

Y =0.389 coresponding to $Z_1 = 20$

$$\text{Then} \quad m \geq 1.26\left[\frac{155.181*10^{\wedge}3}{245*10*20*0.389}\right]^{1/3}$$

m≥2.53mm

Take m = 3

Now number of teeth of planet gear is corrected as

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

$Z_1 = \frac{2*151.47}{3*(5+1)}$

$Z_1 = 16.83$ Take $Z_1=18$

Hence Number of Teeth on sun gear is $Z_2 = i*Z_1$

$=5*18=90$

And Number of Teeth on Ring gear is $Z_3 = 2*Z_1+Z_2$

$=2*18+90=126$

4.1.4 b Pitch circle diameters

Pitch circle diameter of planet gear is $d_1 = m*Z_1 =$ 54mm

Pitch circle diameter of sun gear is $d_2 = m*Z_2 =$ 270mm

Pitch circle diameter of planet gear is $d_3 = m*Z_3 =$ 378mm

4.1.5 b Corresponding centre distance

The corrected centre distance is

$a=\frac{d1+d2}{2} = \frac{54+270}{2} = 162$mm

or we may use

$a=\frac{d3-d1}{2} = \frac{378-54}{2} = 162$mm

Since this value is more than the required minimum centre distance hence the design is safe.

4.1.6 b Face width

Now, face width b = Ψ*a =0.3*162 =48.6mm

(or) b = Ψ_m*m = 10*2 =20mm

Take larger of these 2 values hence face width is b=48.6mm

4.1.7 b Checking of induced compressive stress and bending stress

The induced surface compressive stress

$$\sigma_c = 0.74\left\{\frac{i+1}{a}\right\}\left[\frac{(i+1)*E*[Mt]}{i*b}\right]^{1/2}$$

$$\sigma_c = 0.74\left\{\frac{5+1}{162}\right\}\left[\frac{(5+1)*2.15*10^5*155.181*10^{\wedge}3}{5*48.6}\right]^{1/2}$$

$\sigma_c = 786.64\ N/mm^2$ Which is less than the allowable compressive stress for the material

The induced bending stress,

$\sigma_b = \frac{(i+1)*[Mt]}{a*m*b*y}$

$\sigma_b = \frac{(5+1)*155.181*10^{\wedge}3}{162*3*46.8*0.402}$

$\sigma_b = 101.83\ N/mm^2$ Which is less than the allowable bending stress for the material

Since both σ_c and σ_b are less than their design values, our design is correct

4.2 b DESIGN USING MATLAB COADING

The design process is very time consuming and made some calculation errors also in this a generalized MAT Lab code is written to design planetary gears for any material just by changing material properties at any speed and power.

```
%% Design of Planetary gears for SAE 8620 material
P=50000;      %% P= Power to be transmitted in Watts.
n1=4000;      %%n1=Pinion speed
i=5;          %%i=Gear ratio
Sc=870;       %%Sc=Allowable compressive stress
```

```
Sb=245;        %%Sb=Allowable bending stress
E=2.15*10^5;   %%E=Young's modulus
%% Minimum Centre distance
Si=0.3;        %%Assumed initially
mt=(60000*P)/(2*pi*n1);
k=1.3;         %%assumed
Mt=mt*k;
A=(i+1)*(((0.74/Sc)^2)*((E*Mt)/(i*Si)))^(1/3);
%% Minimum module
Sm=10;         %% Assumed
Z1=20;         %%Z1= number of teeth on pinion assumed initially
Y=0.389;       %%constant value
M=1.26*((Mt)/(Sb*Sm*Z1*Y))^(1/3);
m=round(M)     %%m=module
%% Now number of teeth on pinion is corrected as
Z=(2*A)/(m*(i+1));
Z1=round((Z-2)/2)*2+4
Z2=i*Z1        %%Z2=Sun gear teeth
Z3=2*Z1+Z2     %%Z3=Ring gear teeth
%% Pitch circle diameter
d1=m*Z1        %% PCD of Pinion
d2=m*Z2        %% PCD of Sun Gear
d3=m*Z3        %% PCD of Ring Gear
%% Corresponding centre distance
a=(d1+d2)/2
%% Face width
b1=Si*a;
b2=Sm*m;
b=max(b1,b2)
%% Checking of induced compressive stress and bending stress
y=0.402;
Sc=0.74*((i+1)/(a))*(((i+1)/(i*b))*E*Mt)^(1/2)
Sb=(((i+1)*(Mt))/(a*m*b*y))
```

After putting this code into the editor window in MAT Lab then click on run then save it as design.m file then in command window resuts are obtained i.e;

```
>> Design
m = 3
Z1 =18
Z2 = 90
Z3 =126
d1 = 54
d2 = 270
d3 =378
a =162
b = 48.6000
Sc =786.6328
Sb =98.0567
```

Now we design planetary gears for nodular graphite cast Iron the allowable compressive and bending stress for this material is tabulated below

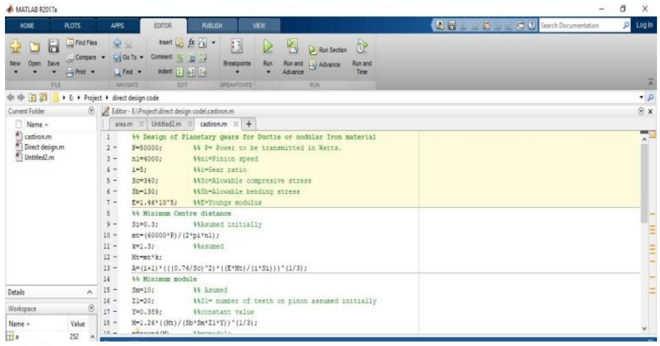


Fig.4.1 Editor window in MATLAB

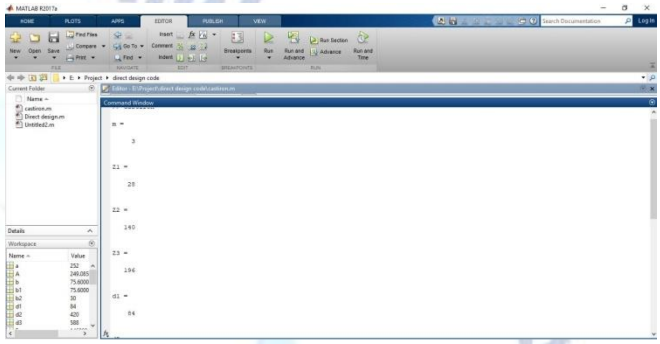


Fig.4.2 Command window in MATLAB

4.3 CALCULATION OF MASS AND VOLUME

In this work we consider three planet gears and one sun gear and one ring gear so the volume of the gear train can be find out by using following formula

$V = \frac{\pi}{4} * b * (d_2^2 + 3d_1^2)$

Where b = face width in mm

d_1 = Pitch circle diameter of planet gear in mm

d_2 = Pitch circle diameter of sun gear in mm And the mass can be find out by using

$m = \frac{\pi}{4} * b * \rho * (d_2^2 + 3d_1^2)$

Where ρ = Density of the material in kg/mm^3

4.3.1 Mass and volume calculation for SAE 8620 material Gear train

The mass and volume for the planetary gear train is calculated by using MAT Lab code given below

```
d1=54;     %%Pitch circle diameter of planet gear
d2=270;    %%Pitch circle diameter of sun gear
b=48.6;    %%face width
p=7.85*10^-6;%%Density of the material
V=(pi/4)*b*(d2^2+3*d1^2)
m=V*p
```

Then we run the program to get the result

>> Result

V = 3.1165e+06 in mm^3

m =24.4648 in Kg

4.3.2 Mass and volume calculation for Ductile (or) modular Iron material Gear train

The mass and volume for the planetary gear train is calculated by using MAT Lab code given below

```
d1=60;      %%Pitch circle diameter of planet gear
d2=300;     %%Pitch circle diameter of sun gear
b=54;     %%face width
p=7.3*10^-6;%%Density of the material
V=(pi/4)*b*(d2^2+3*d1^2)
m=V*p
>> area
```

V = 4.2751e+06 in mm^3

m = 31.2081 in Kg

4.3 COMPARISON OF THE RESULTS

In this section the number of teeth of all three gears, face width, centre distance, module, mass and volume for the both materials is tabulated below

S.no	Variables	SAE8620	Nodular Graphite cast Iron
1	No.of teeth on Planet gear (Z_1)	18	20
2	No.of teeth on Sun gear (Z_2)	90	100
3	No.of teeth on Ring gear (Z_3)	126	140
4	Module(mm)	3	3
5	Face width(mm)	48.6	54
6	Centre distance(mm)	162	180
7	Mass(kg)	24.46	31.2
8	Volume(mm^3)	$3.1165*10^6$	$4.2751*10^6$

Table 4.3 Results obtained in MAT Lab

V. GEOMETRY MODELLING

5.1 INTRODUCTION

Geometric modelling is the starting phase of collaborative product development. Geometric modelling is the computer compatible mathematical description of an object, which allows displaying model on the screen and modifying the object. Commercial CAD packages such as CATIA, IDEAS, CREO, SOLIDWORKS, and UNIGRAPHICS are available in the market for geometric modelling. In the present work SOLIDWORKS is used to create planetary gear train.

The SOLIDWORKS 3D CAD software delivers powerful design functionality with the intuitive SOLIDWORKS user interface to speed your design process and make you instantly productive.

5.1.1 Modules used in Solid Works

Solid Works extends design applications through full integration with best in class solutions. The following modules will be used in this project

- Part modelling
- Assembly Modelling
- Drawing
- Motion analysis

Motion analysis is a powerful tool that's available in SOLIDWORKS Premium. Motion can simulate moving or dynamic systems and will give outputs to size your design. Some of the outputs are

- Displacements
- Reaction Forces
- Accelerations
- Motor Power
- Friction Power
- Kinetic energy

5.2 PROCEDURE

5.2.1 Modelling of SAE8620 Material gear

- First we model Sun gear
- Open Solid Works and chose the command File > New > Part.
- Select the system of units as MMGS and choose the sketch plane
- Open design library open transmission systems then drag on Spur gear to the window

Fig 5.9 Nodular Graphite Cast Iron Material Planetary gear train with dimensions

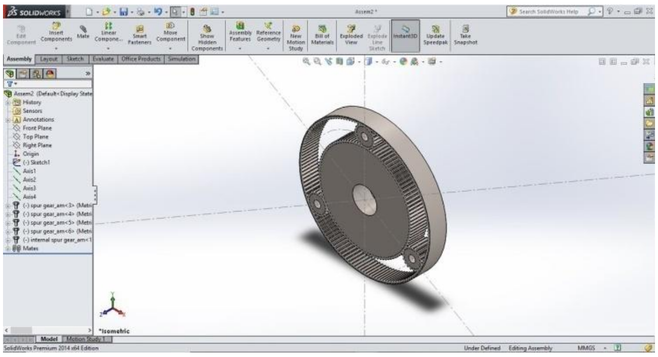


Fig 5.10 Nodular Graphite Cast Iron Material Planetary gear train

5.3 CALCULATION OF MASS AND VOLUME

- Open SAE8620 material gears assembly drawing then select one gear go through part modelling then click on material here add material properties to the part

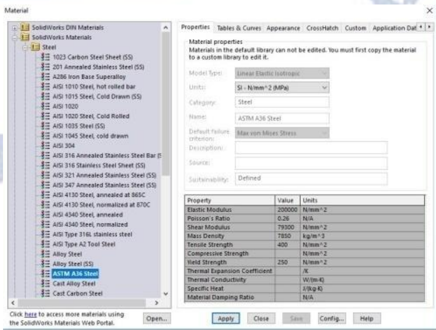


Fig 5.11 Material properties window in Solid Works

- Similarly add material properties to all gears in the gear train

- Now open Evaluate window click on mass properties the results will be displayed as below

```
Mass properties of SA new
   Configuration: Default
   Coordinate system: -- default --

Mass = 24.21 kilograms

Volume = 3083860.44 cubic millimeters

Surface area = 454682.23 square millimeters

Center of mass: ( millimeters )
    X = 0.00
    Y = 0.00
    Z = -24.30

Principal axes of inertia and principal moments of inertia: ( kilograms * square
Taken at the center of mass.
    Ix = (1.00, 0.00, 0.00)        Px = 168510.53
    Iy = (0.00, 1.00, 0.00)        Py = 168510.53
    Iz = (0.00, 0.00, 1.00)        Pz = 327491.21

Moments of inertia: ( kilograms * square millimeters )
Taken at the center of mass and aligned with the output coordinate system.
    Lxx = 168510.53        Lxy = 0.00             Lxz = 0.00
    Lyx = 0.00             Lyy = 168510.53        Lyz = 0.00
    Lzx = 0.00             Lzy = 0.00             Lzz = 327491.21

Moments of inertia: ( kilograms * square millimeters )
Taken at the output coordinate system.
    Ixx = 182805.29        Ixy = 0.00             Ixz = 0.00
    Iyx = 0.00             Iyy = 182805.29        Iyz = 0.00
    Izx = 0.00             Izy = 0.00             Izz = 327491.21
```

Fig 5.12 Mass properties for SAE8620 Material Gear train

- Now open Nodular Graphite Cast Iron material gears assembly drawing
- Similarly add material properties as described above for all gears then evaluate mass properties. The results shown below

```
Mass properties of ci
   Configuration: Default
   Coordinate system: -- default --

Mass = 30.83 kilograms

Volume = 4223054.70 cubic millimeters

Surface area = 560383.68 square millimeters

Center of mass: ( millimeters )
    X = 0.00
    Y = 0.00
    Z = -27.00

Principal axes of inertia and principal moments of inertia: ( kilograms * square
Taken at the center of mass.
    Ix = (1.00, 0.00, 0.00)        Px = 259989.15
    Iy = (0.00, 1.00, 0.00)        Py = 259989.15
    Iz = (0.00, 0.00, 1.00)        Pz = 504995.76

Moments of inertia: ( kilograms * square millimeters )
Taken at the center of mass and aligned with the output coordinate system.
    Lxx = 259989.15        Lxy = 0.00             Lxz = 0.00
    Lyx = 0.00             Lyy = 259989.15        Lyz = 0.00
    Lzx = 0.00             Lzy = 0.00             Lzz = 504995.76

Moments of inertia: ( kilograms * square millimeters )
Taken at the output coordinate system.
    Ixx = 282462.98        Ixy = 0.00             Ixz = 0.00
    Iyx = 0.00             Iyy = 282462.98        Iyz = 0.00
    Izx = 0.00             Izy = 0.00             Izz = 504995.76
```

Fig 5.13Mass properties for Nodular Graphite Cast Iron Material Gear train

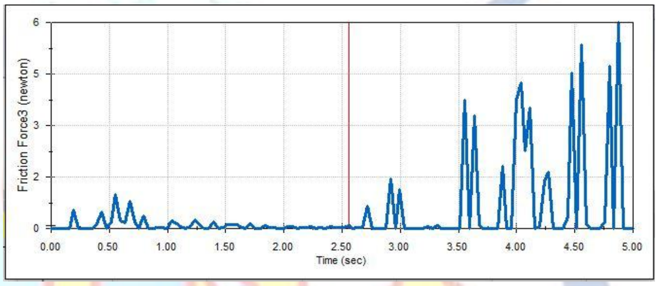
Material	Mass(Kg)	Volume(mm^3)
SAE8620	24.21	3083860.44
Nodular Graphite Cast Iron	30.83	4223054.70

Table 5.1 Mass and Volume of Gear train set in Solid works

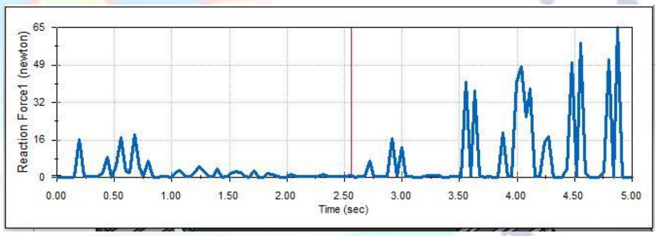
The results obtained in this calculation almost equal to the results that was obtained in MAT Lab coding i.e; in Table16 so our model is almost like realistic model now we conduct motion analysis for each model

5.4 MOTION ANALYSIS

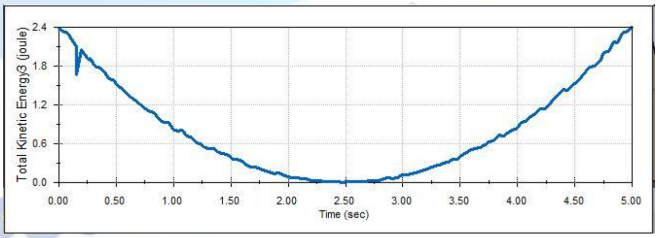
- First we conduct motion analysis on SAE8620 material gear train for that open assemble model of the gear train in Solid Works
- Now move on to motion analysis tab in Solid works. In that add motor to the sun gear with 800rpm with constant speed
- Then add contact between parts by selecting all the parts
- Now run the motion study for 5sec time
- Then generate Friction force, Contact force, Total kinetic energy, Power consumption plots along with time
- The plots were shown below



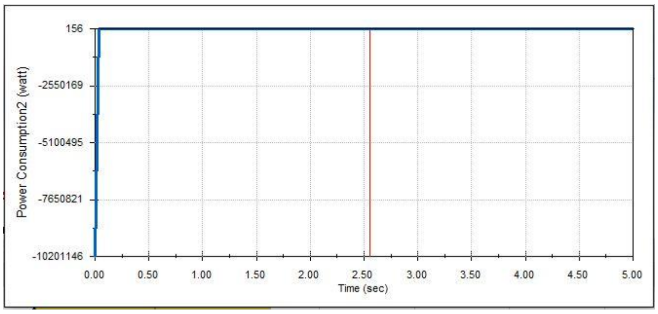
Graph 5.1 Friction force VS Time plot for SAE8620 Material Gear train



Graph 5.2 Contact Force VS Time plot for SAE8620 Material Gear train

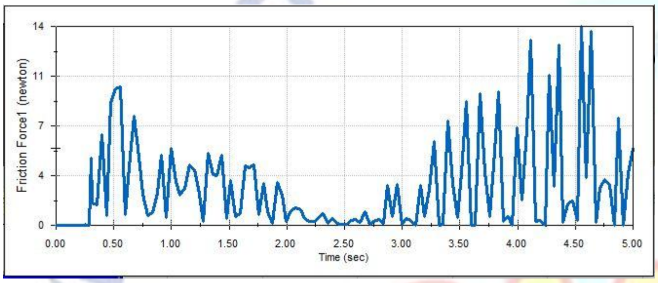


Graph 5.3 Total K.E VS Time plot for SAE8620 Material Gear train

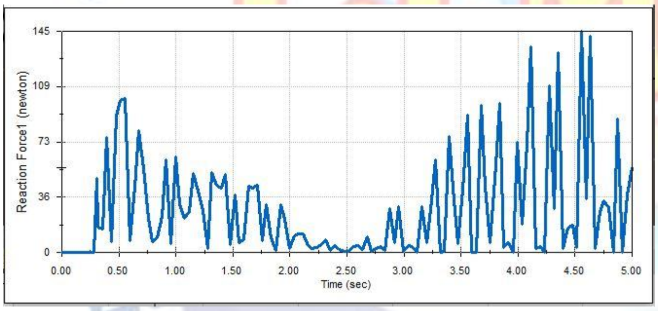


Graph 5.4 Power Consumption VS Time plot for SAE8620 Material Gear train

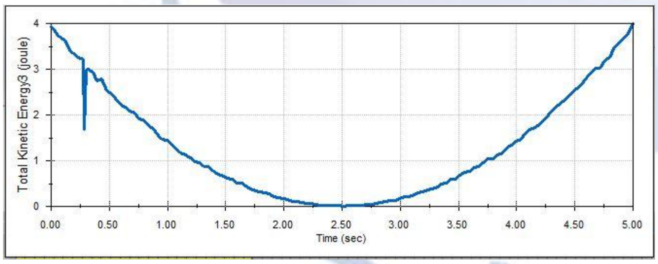
- Similarly perform motion analysis for Nodular Graphite Cast Iron Material Gear train the results were shown below



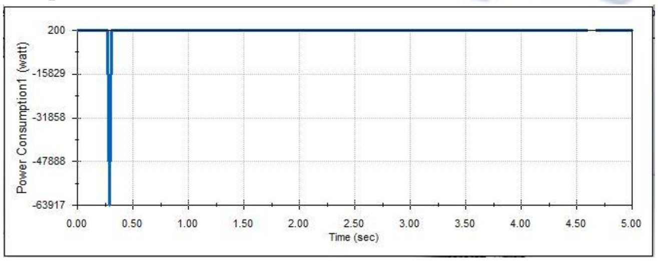
Graph 5.5 Friction force VS Time plot for Nodular Graphite Cast Iron Material Gear train



Graph 5.6 Contact Force VS Time plot for Nodular Graphite Cast Iron Material Gear train



Graph 5.7 Total K.E VS Time plot for Nodular Graphite Cast Iron Material Gear train



Graph 5.8 Power Consumption VS Time plot for Nodular Graphite Cast Iron Material Gear train

VI. CONCLUSIONS

In this Project the detail explanation is given, how to resolve material choice issue for the planetary gear utilized in car gear box through MCDM. The MCDM incorporates the weighted Sum, TOPSIS and VIKOR techniques for the positioning of the elective materials as per decided criteria. The material weighting of the material properties was performed by utilizing the traded off weighting strategy that sorted out of AHP and Entropy strategies.

The selection of material is considered through the ranking which has taken form comparison Graph(3.1). The table gives the Ranking of the materials. According to ranking the SAE 8620 has selected for planetary gear material.

Then, a planetary gear has designed with the materials of SAE 8620 and also with Nodular Graphite Cast Iron. And hence a mass and volume calculations are made with the MAT lab coding and the same have been compared with Solid works software.

The maximum Frictional force obtained for SAE8620 material Gear train is **6 N** and for Nodular Graphite Cast Iron material Gear train is **14 N**

The maximum Contact force obtained for SAE8620 material Gear train is **65 N** and for Nodular Graphite Cast Iron material Gear train is **145 N**

The maximum Kinetic energy obtained for SAE8620 material Gear train is **2.4 Joules** and for Nodular Graphite Cast Iron material Gear train is **4 Joules**

The maximum Power consumption obtained for SAE8620 material Gear train is **156 Watts** and for Nodular Graphite Cast Iron material Gear train is **200Watts**

Therefore it is was conclude that the gear train manufactured with SAE8620 material gives gear to more strength when compare to other material for the automobile gear box application.

A generalized MAT Lab code was written for Gear designing. The results obtained by MAT Lab code was verified and validated with Theoretical solution. Thus, use of programming for design of Gears reduces the need of tedious calculations, computation time and eliminates calculation errors.

FUTURE WORK

- The material selection methods can be applied to other mechanical components for material selection problems.

- The Research could be further extended through further optimization by assigning different materials to all the three gears. And for different gears (helical, worm etc.).

REFERENCES

[1] Halil Calıs_kan a,⇑, Bilal Kurs_uncu a, Material selection for the tool holder working under hard milling conditions using different multi criteria decision making methods

[2] Hüseyin Filiza, S. Olgunera;* A Study on Optimization of Planetary Gear Trains

[3] Y. Mohan, T. Seshaiah, Int. J. Engin. Res. Appl.(IJERA) 2, 311 (2012).

[4] S. Golabi, J.J. Fesharaki, M. Yazdipoor, Mech.Mach. Th. 73, 197 (2014).

[5] K. Tamboli, P.M. George, R. Sanghvi, ProcediaTechnol. 14, 513 (2014).

[6] J.Stefanović-Marinović , Load Allocation and Equilibrium for Planetary Gear Reducers of Earth-Pressure-Balance Shield Machine, Journal of Mechanics Engineering and Automation, 2,2012, 365-373

[7] Likhitha R Reddy Design of Transmission Shafts Using C Programming Language

[8] Syed Ibrahim Dilawer Design, Load Analysis and Optimization of Compound Epicyclic Gear Trains

[9] A Text book of Machine Design by S.Md.Jalaludeen.

[10] Design data hand book by S.Md.Jalaludeen.

[11] Design data hand book by PSG College of engineering and technology

[12] Text book of Machine Design by R.S.Khurmi & J.K.Gupta