Design and Analysis of Powertrain

Yogesh, Shreyansh Gaddamwar, Aniket Singh, Karthikeyan K, Saurabh Rai, Jaspreet Singh

School of Mechanical Engineering, Lovely Professional University, Phagwara 144411, India

Abstract

A powertrain is a machine-driven part in an automobile that transfers the power from engine to the wheels as a torque. Through the driveshaft, the engine's torque is conveyed to the wheels of the automobile. Basically, powertrain is made up of a transmission system and a driveshaft. It provides a diverse leverage between the engine and the drive wheels. The core objective of this report is to design a powertrain for an automobile engine by doing on paper calculation and made a model through computer aided designing.

Keywords: Power train; Computer aided designing; Leverage; driveshaft; transmission system

1. Introduction

The powertrain is at the core of automobile design; the engine provides the power, which is then accomplished and meticulous through the transmission and final drive mechanisms.
The design of the powertrain has predictably been undertaking by analysing each of the sub-systems independently and the distinct components. Automobile Powertrain Systems runs thorough report and analysis of all the powertrain components and then treats them collected so that the complete performance of the automobile can be understood and calculated. The text is well maintained by applied complications and operated illustrations. Model is made of the Solidworks software.

a) Torque:

Torque is a rotating force about its axis that might or might not result in motion. An automobile travels because of the torque the drive axle exerts on the wheels and tires to make them rotate. Being a form of power-driven energy, torque cannot be created or destroyed—it is transformed from one form of energy to alternative form of energy. Engine torque is established when ignition pressure forces a piston downward to rotate the crankshaft. The quantity of torque formed will differ depending on the size and design of the engine and the throttle opening.

b) Horsepower:

The word horsepower means the rate of doing work. Work is done when a defined amount of mass is moved a defined distance by a force. Whether the object is moved in 5 minutes or 5 hours does not make a difference in the amount of work accomplished, but it does disturb the quantity of power required.

c) Power train and its parts:

1. Engine
2. Clutch
3. 8 speed gear-box
4. Universal joint
5. Differential
6. Axle

2. Engine specification

Engine used in this project was 2020 dodge challenger 8 speed manual transmission:

1. Cylinder - v6 in 60 degree
2. Bore and stock – 96 mm * 83 mm
3. Power - 305 BHP; 227 KW; 6300 rpm
4. Torque - 363 NM; 4800 rpm
5. Wheel dimensions - 240/45, R18 inches
6. Gear box - 8 Speed transmission
7. Frontal area – 2.38 m^2
8. Wheel base – 2948 mm
9. Track width – 1610 mm front; 1621 mm Rear
10. Weight - 1750 kg
11. Distribution of weight – 52% front; 48% rear

3. Engine calculations

1. Rolling resistance – $F_r = C_r*m*g$

- Cr = Coefficient of friction = 0.02 tires on tarmac
- M = Mass of the vehicle = 1750 kg
- g = Acc. due to gravity = 9.81 m/s

F_r = 0.02*1750*9.81

F_r = 343.45 N

2. Aerodynamic drag – $F_d = (ʄ*a*(v)^2*0.5)/2$

- ʄ = 1.225 kg/m^3 density of the air
- a = frontal area = 2.30 m^2
- V = Velocity of the car = 210 km/h = 58 m/s

$F_d = (1.225*2.30*(58)^2*0.5)/2$

Fd = 2370 N

3. Grade resistance – F_g = (m*g*sin(x))

- x = 30 degree of elevation

F_g = 1750*9.81*sin (30)

F_g = 8584 N

4. Acceleration force – F_a = (m*v)/t

- t = time taken to reach the top speed in seconds

F_a = (1750*58)/35 = 2900 N

5. Total resistance – 2900 N + 8584 N + 2370 N + 343 N = 14198 N
6. Initial torque on wheel - T_w = Total resistance*r*rolling resistance

- r = radius of the wheel = 0.22 m

T_w = 14198*0.22*0.02 = 62 Nm

7. Final reduction - Initial torque/engine torque *primary reduction*5th gear ratio

- Primary reduction = 1
- 5th gear ratio = 1.28

I_f = 62/ (363*1*1.28) = 0.13

8. Over-all gear ratio

An = Gear ratio*primary reduction* final reduction

Table 1 – Overall gear ratio

Gear	Gear ratio	Overall gear ratio
1	4.71*0.13	0.61
2	3.14*0.13	0.40
3	2.11*0.13	0.27
4	1.67*0.13	0.21
5	1.28*0.13	0.16
6	1.0*0.13	0.13
7	0.84*0.13	0.10
8	0.67*0.13	0.08

9. Velocity on each gear – V_n = (4*∏*power*r)/1000*An

V_n = (4*3.14*6300*0.22)/1000*An

V_n = 17.40/An

Table 2 – Velocity on each gear

Gear	Velocity/overall gear ratio	Velocity on each gear
1	17.40/0.61	29
2	17.40/0.40	44
3	17.40/0.27	64
4	17.40/0.21	83
5	17.40/0.16	108
6	17.40/0.13	134
7	17.40/0.10	174
8	17.40/0.08	217

4. Gearbox design calculations

The dynamic diameter of a gear is the pitch diameter. The pitch diameter is the diameter of the gear at the point where the teeth of the two gears meet and transmission power. The gear teeth are designed to be able to slide in and out of network with a minimum quantity of friction and wear.

Helical gear—Helical gears are the furthermost used of all gears used in transmissions. These gears have teeth cut in a helix or spiral shape. Helical gears are noiseless, but produce end or axial thrust beneath a weight. A helical gear is stronger than a spur gear and has an practically non-stop power stream because of the angled teeth.

No. of transmission gear (n) = 8

N_{max} = 6300 rpm, N_{min} = 1000 rpm

$\varphi^{n-1} = N_{max}/N_{min}$

$\varphi^7 = 6.3 = 1.289$;

which is not a standard as per PSG data book

So, we took 1.12 which is a R20 speed series

Table 3 - Calculating the rpm (skipping 2 rpm speed)

S.no	Rpm calculation	Final rpm
1	1.12*1000	1120
2	1.12*1120	1500
3	1.12*1500	1900
4	1.12*1900	2300
5	1.12*2300	3000
6	1.12*3000	3700
7	1.12*3700	4800
8	1.12*4800	6300

1. Transmission parameters

Maximum climbing ability - $Ig >= (m*g*\psi_{max}*r)/(T_{max}*I*\eta t)$

- M*g = 1750*9.81 = 1716.75 N
- ψ_{max} = F*cos(x)+sin(x)
- F = 0.0076 rolling resistance; x = 30 maximum gradeability of vehicle
- ψ_{max} = 0.50
- I = final drive ratio = 3.07
- ɳt = efficiency = 0.85

Ig> = (1716.75*.50*.45)/363*3.07*.85 = 4.0

2. Centre distance - $A=Ka(\sqrt[3]{T_{max}})$

- Ka = centre distance factor of sedan, multi shaft transmission 9.5-11
- T_{max}= torque* first speed gear ratio*efficiency
- R= diameter of the wheel

A = 113 mm

3. Helical gear module – $(\sqrt{2*T_1*Kg*Cos\beta})/\prod_z*K_c*k_e*y*\sigma_w$

- As per PSG design data book
- T_1 = load; Stress concentration factor Kg = stress in C45 material = 1.65
- β = helix angle = 30; Tooth width coefficient K_c = 7 - 8.6
- K_e = overlapping = 2
- Tooth coefficient y = 0.38; Bending stress σ_b = 1350

On the basis of the national standard **MODULE** came out to be = 3

4. Calculation of number of teeth - Z1= (2*a)/m(I+1)

I=6300/1000

Z1= (2*113)/3(6.3+1) = 10

Table 4 - Calculating teeth from gear ratio

Gear ratio*Z1	Total no of teeth	Diameter of the gear
4.71*10	47	47*3=141mm
3.14*10	32	32*3=93 mm
2.11*10	21	21*3=63 mm
1.67*10	17	17*3=41 mm
1.28*10	13	13*3=39 mm
1*10	10	10*3=30 mm
0.84*10	8	8*3=24 mm
0.67*10	7	7*3=21 mm

5. Distance between the shaft

A1 = (d2+d1)/2 = 117 mm

A2 = (d5+d6)/2 = 35 mm

6. Face width - ψ*m =10*3 = 30 mm

5. Design calculations of clutch

Engine power necessity be stationary when making a shift in a manual transmission. The clutch is used to break the power stream to permit the transmission to be raised. It is also used to extravagance the engagement of the power stream when the automobile starts from a slowdown. The negligeable slippage as the clutch participates permits the engine speed to stay up where it produces working torque as the automobile begins moving. Most automobiles use a single-plate clutch assembly that is mounted on the engine flywheel.

According to design parameters of clutch

- Outer diameter = 230 mm; Inner diameter = 200 mm
- Intensity = 0.1 N/mm^2
- Coefficient of friction = 0.65

1. Axial force – $F_a = 2*\prod*p*(r_0-r_i) = 2*3.14*0.1*10^6*0.1(.115-.1) = 942.47$ N

2. Uniform pressure theory - T=µ*W*r

- µ = coefficient of friction =0.65
- W = axial force = 942.47 N

$R = (2/3)\ (r_1^3-r_2^3)/(r_1^2-r_2^2)$

T = 63.81 Nm

3. Uniform wear theory: $T=0.5*u*W*(r_1-r_2) = 65.85$ Nm
4. Spring - 1.25*Fa = 1.25*942.47 = 1178.08 N

- 6 springs = 1178.08/6 = 196 N

5. Wahls stress factor = C = 6

K = ((4C-1/4C-4) + 0.615)/C = 1.25

- Maximum shear stress induces in wire is 420 Mpa
- Considering F.O.S = 1*420 Mpa = 420 Mpa
- $420 = (1.25*8*196*6)/3.14*d^2$

D = 3 mm

6. Diameter of spring: D = C*d; 6*3 = 18 mm
7. Let assume working turn = 5

- $G = (8*W_s*C^3*n)/g*d$
- $G = 85*10^3$
- $G = (8*196*6^3*5)/255*10^3$
- G = 6.64 mm

8. Square and ground head

- N = n+2 = 7
- $L_f = n*d+G+0.15G = 28.63$

9. Pitch = L_f/N-1 = 4.77 mm

6. Design calculations of propeller shaft

Driveshafts, similarly termed a propeller shaft, transfer power from one part to another. Rear-wheel-drive vehicle driveshafts are mostly made from steel tubes, and generally have a universal joint (U-joint) joint at each end. A U-joint permits the shaft to modification angle as the drive axle moves down and up when the wheels travel over bumps. Speed instabilities occur in the driveshaft as the U-joints transfer power at an angle, but these instabilities are cancelled out or rejected by the position of the U-joint at the other end of the driveshaft.

Assuming diameter = 40 mm

L = 1000 mm; Torque = 363Nm; Power = 305 BHP; Incline = 3 degree

Material = C45; Density = 7600kg/$m^{3;}$ Y = 370 Mpa; E=$38*10^4$ Mpa; N = 6300 rpm

1. I = Moment of inertia – $(\prod*d^4)/32 = 251200$
2. Torsional stress = $(T*d/2*I = 363*10^3*40)/2*251200 = 28.90$ Mpa
3. Deflection = G = $(5*m*g*\cos(\phi)*L^3)/384*E*I$

M=mass of the rod

G = (5*9.86*cos (3) *1000^3)/384*(38*10^4) *251200 = 0.0013 mm

4. Critical speed – N_c = 30/(∏*√g/G) = 8329.5

7. Design of universal joint

A U-joint permits the shaft to alteration angle as the drive axle moves down and up when the wheels travel over bumps. Speed instabilities occur in the driveshaft as the U-joints transfer power at an angle.

1. Diameter of shaft - T = ∏/(16*t*d^3)

- Shear stress allowable = 60 Mpa

363*10^3=∏/(16*60*d^3) = 313 mm

2. Diameter of pin - 363*10^3=∏/(2(Dp)^2*60*313) = 35 mm

8. Design of axle (semi-floating)

An axle is a compacted combination of a transmission, the final drive gear reduction, and the differential. A transmission usually has single output channel that pairs to the rear axle through the driveshaft. An axle is a fundamental shaft for a revolving wheel or gear. On automobiles having wheels, the axle might be secure to the wheels, rotating with them, or fixed to the automobile, with the wheels rotating about the axle.

Frictional force = μ*W

- Coefficient of friction =0.6; W = fully loaded = 1750 kg
- 48% rear = 840 kg; Taking 500 kg on one wheel

1. Torque=μ*W*r = 0.6*500*0.22 = 66 Nm
2. Diameter of shaft – T = ∏/(16*325*d^3)

325 = shear stress average; D = 1m = 1000mm

3. Bending – F_d = (32*500√1+(0.6)^2)/3.14 = 6000 N
4. Shear stress = Fs = T*Y/Ip

Ip =(∏*d^4)/32 = 66 Nm

Fs = (16*0.6*500*0.22)/3.14*1 = 336 N/m^2

5. Weight on spring - 840*3g = 2520 kg; 2520*9.81 = 24.72 kN
6. Axle hosing – Torque = √1+μ^2 * Ws*L

L = 0.117; Ws = 420 Mpa

Bending stress - Fb=32*420*0.11*1.16/3.14 = 546.16N

Shear stress - Fs=16μ*W*r/∏*d^3 = 336 N/m^2

9. Differential gear box

The differential is a torque-splitting mechanism that permits the dual axle shafts to function at dissimilar speeds so that an automobile can turn angles. When an automobile turns a corner, the wheel on the external of the rotating radius necessity travel farther than the internal wheel, but its necessity does this in the identical period of time. So, its necessity rotates quicker while turning. Furthermost differentials are self-possessed a group of tetrad or further gears. Single gear is joined to respective axle and two are riding on the differential pinion shaft.

The equations are from the PSG data book;

- Input shaft power = 305 bhp; Speed = 6350 rpm; Torque=4800 rpm
- Material C45; σ_p= 620; Hardness=175; $\sigma_p/3$=206.67
- Gears = cast iron grade 25
- GR = 6350/4800 = 1.25

Height of addendum in bevel gear = 1*m = 5

Assuming module = 5

1. Minimum no. of teeth = (2*ha*cos (45))/(m*sin (20))
   = 13.6 = 14 (approx.)

Pitch cone angle = 45; Back cone angle = 45; Pressure angle = 20

2. Gear ratio = Zg/Zp; Zg = 18
3. Pitch diameter = m*Zg = 90 mm
- Pinion = 5*14 = 70 mm
4. Distance - AO = $\sqrt{(dg/2)^2+(dp/2)^2}$ = 57 mm
5. Face width = AO/3 = 19 mm

tanYp = Zp/Zg

Yp=37.87; Yg = 90-38.87 = 52.13

6. Beam strength - Sut/3=1000/3 = 333.33
7. Virtual teeth = Z'p*cos (37.87)/14 = 17.73; Z'g=29.32
8. Lewis form factor = Y'p = 0.55- (2.64)/Z'p = 0.41
9. Beam strength=Fb = σb*b*m*Y'p[1-(b/AO)]
   = 333.33*19*5*0.41[1-(19/57)]
   =856.91 N
10. Ratio factor = (2*Z'g)/(Z'g)+(Z'p) = 1.24
11. Load stress factor = (0.16*t*BHN)/ $(100)^2$ = 0.16*16 = 2.56
12. Wear strength = (0.75*dp*b*q*k)/cos (Y'p) = 4013.26
13. Pitch line velocity – (∏*dp*Np)/60*10^3
    = (3.14*70*6000)/60*10^3 = 2.19
14. Tangent force=P/V

Power to watt = 305 Bhp - 2274.80 watt

2274.80/2.19=103850

15. Velocity factor – Kv = 6/(6+v) = 0.73
16. Eff load = (Ka*Km*Ft)/Kv
    = 1.25*1*103850/.73 = 177825
17. Fos=1*177825=177825
18. Mean radius – Pinion = dp/(2-[bsin(Yp)/2])
    = 35-5.83 = 29.16

Similarly, Gear = 37.50

19. Pitch error = 8+.63[m+.25*√Yp]; Pinion=12.35; Gear=12.51
20. Measuring teeth = 12.35 + 12.51 = 24.86*10^3
21. Ft_{max} = 1.25*1*103850 = 129812.5
22. Deformation factor = 11000*C;
    C=30*10^{-3} = 330 Nm
23. Buckingham's equation = Fb=(21*v*[bc+Tmax])/21v√bc+tmax

= 15084.92

24. Feff = 144897.42

FOS = 177825/144897 = 1.22

10. 3D Modeling of parts:

The following are parts of Powertrain done in Solidworks 2020: -

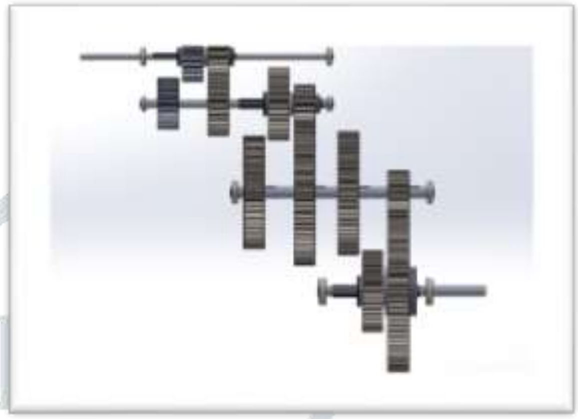


Figure 1 – (A) Orthogonal view of Clutch Assembly (B) Front view of Gear box

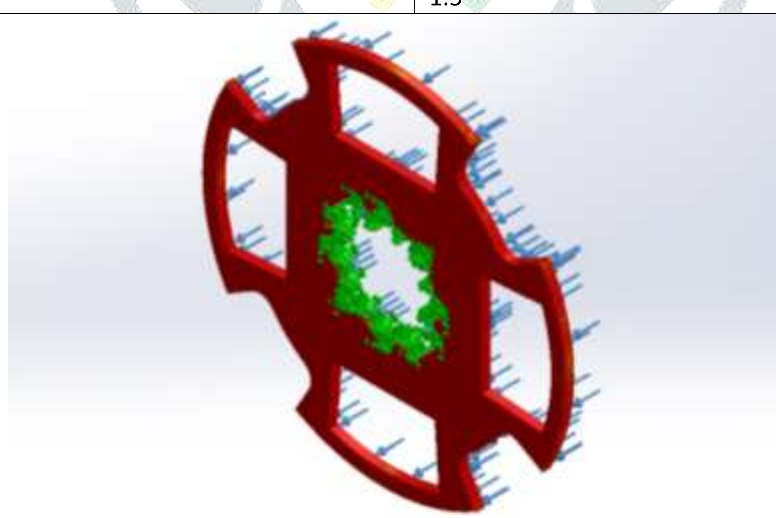
Figure 2 – (A) Orthogonal view of Universal joint (B) Orthogonal view of Propeller shaft


Figure 2 – (A) Orthogonal view of Differential (B) Orthogonal view of Rear axle

11. Analysis of parts of powertrain:

A. Analysis of Pressure plate:

Load name	Load Image	Load Details
Force		**Entities:** 1 face(s) **Type:** Apply normal force **Value:** 64 N

<table>
<tr><td>Name</td><td>Factor of Safety</td></tr>
<tr><td>Factor of Safety</td><td>1.5</td></tr>
<tr><td colspan="2">
clutch 06-UNIFROM PRESSURE ANALYSIS-Factor of Safety-Factor of Safety1</td></tr>
</table>

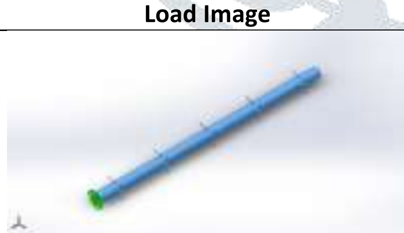
B. Analysis of Friction plate:

Load name	Load Image	Load Details
Torque		**Entities:** 48 face(s) **Type:** Apply torque **Value:** 66 N.m

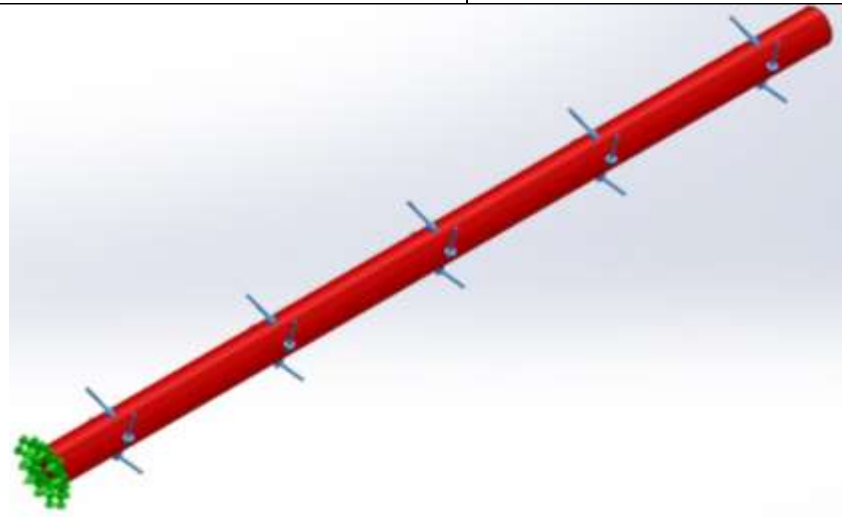
Name	Factor of safety
Factor of Safety	1.8

clutch 11-UNIFORM WEAR OF FRICTION PLATE-Factor of Safety-Factor of Safety1

C. Analysis of Propeller shaft:

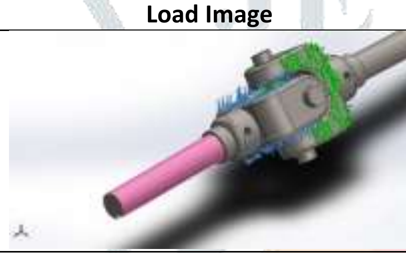
Load name	Load Image	Load Details
Torque		**Entities:** 1 face(s) **Type:** Apply torque **Value:** 2,890 N.m

Name	Factor of Safety
Factor of Safety	1.3

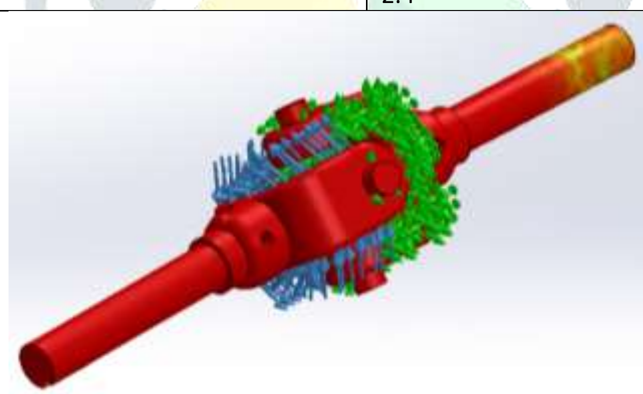


Part1-torsial stress-Factor of Safety-Factor of Safety1

D. Analysis of Universal Joint:

Load name	Load Image	Load Details
Torque		**Entities:** 2 face(s) **Reference:** Face< 1 > **Type:** Apply torque **Value:** 400 N.m

Name	Factor of Safety
Factor of Safety	2.4



universal joint-Torque on universal joint-Factor of Safety-Factor of Safety1

Conclusions:

The design development established the entire design technique of a powertrain assembly. All the mechanisms of the powertrain were designed systematically. The stress produced in the defined mechanisms of the power train were analysed through software. The hard work taken to explain all the compulsory design and analysis discussions have provided a solid preliminary point into the fundamentals of powertrain system design.

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