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Design and Analysis of Drive Train of FSAE

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Abstract.

An essential part of chain drive for transferring power from the engine to the tyres through shafts is the chain sprocket. Chain sprockets should be correctly designed and constructed to provide effective power transmission. In the chain drive sprocket, weight reduction is a possibility. In this work, a chain sprocket's design and dependability are examined using finite element analysis. For static and fatigue analysis of sprocket design, ANSYS software is employed. These findings have been used to optimise the sprocket for weight reduction. During sprocket vibration, modal analysis is carried out.

Keywords:FSAE, Drive Train, Ansys

1. Introduction

The power that is sent to the drive shaft in a car is produced by the engine. Chain drive is one of the mostly used drivetrains for this power transfer. The chain assembly is made up of the driven, chain, and chain sprockets. The driven sprocket receives power through a chain once the driving sprocket is fastened to the engine's output shaft. The driven sprocket also sends

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energy to the drive shaft. As a result, there is a chance to lighten the chain assembly by designing and optimising the driving sprocket. High power transfer and rotational speed result in significant stress and vibrations on the sprocket teeth. Therefore, proper sprocket design, construction, and attachment are essential [1]. Chain places a heavy stress on the teeth of the driven sprocket as it transfers power from the driving sprocket [2]. Therefore, the maximal stresses placed on teeth are computed. The yield stress of the material should not exceed the stress caused by the load. Failure is a possibility if the stress exceeds the material's yield stress. In order to ensure that the suggested design has a safety factor larger than one, static analysis was carried out. It's vital to test the sprocket for fatigue loading due to the cyclic load the chain places on it as well. In a fatigue analysis, the sprocket's fatigue life is estimated, and it is made sure that the minimum fatigue life is higher for the sprocket to be used safely for an adequate amount of time. Component failure occurs when a crack begins to form after the minimum fatigue life and progresses over time. Because of this, it's critical for every component to have a long enough fatigue life. The static analysis and fatigue analysis of a component are performed using FEA. This guarantees the component's dependability and safety. The component is further optimised for weight reduction using the FEA data. Prior to implementation, the updated design underwent a new analysis. For FEA analysis of the sprocket, ANSYS software is employed. After being actually installed on a vehicle and undergoing extensive testing, this sprocket design has been experimentally proven.

Further vibration testing was done on the sprocket design since vibrational forces are also very important in sprocket design. Modal analysis makes sure that the sprocket resonance frequencies are outside of their operating range.

2. Preliminary Design

In Catia V5, a CAD model with the necessary measurements and characteristics was produced. ANSYS Workbench's meshing tool was used to produce a very thin mesh with elements that were 3mm in size which is shown in Fig.1.

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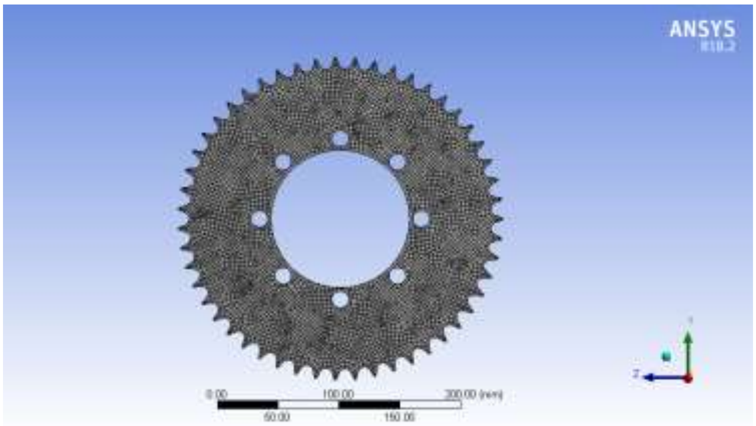

Figure 1 CAD model of sprocket and meshing

2.1 Specifications

Due to the strength and weight of the sprocket, material choice is a significant factor. To guarantee that the component had adequate strength, mild steel was employed. Mild steel can also handle high temperatures, variable loads, and simultaneous compression, tensile, and shear loads without giving way.

Properties of mild steel:

- Density = 7850 kg/m3
- Young's Modulus = 250 MPa
- Poisson's Ratio = 0.3

Weight distribution:

a) To the front axle: 144Kgs

b) To the rear axle: 176Kgs

Wheelbase: 1560mm

a) L1 (Dist. From front axle to C.G) = 858mm

b) L2 (Dist. From rear axle to C.G) = 702mm

Bottom height (h):300mm (Dist. From ground to C.G)

Track width:

a) Front axle: 47in (1193.8mm)

b) Rear axle: 45in (1143mm)

Tire Specifications:

Hoosier tire code (Dry): 175/60 R13

Section width: 175mm

Aspect ratio: 60

Rim Diameter: 13in (330.2mm)

Section height: (60*175/100) =105mm

Outer Diameter: 540.2mm

- JK Ultima sport tire code (Wet): 185/60 R13

  Section width: 185mm

  Aspect ratio: 60

  Rim Diameter: 13in (330.2mm)

  Section height: (60*185/100) =111mm

Outer Diameter: 552.2mm

- Maximum static friction between tyre and asphalt is μ=0.7
- Rolling friction coefficients (fr): 0.01

  a) Car tyres on concrete or asphalt: 0.013

  b) Car tyres on Rolled gravel: 0.02

  c) Tar Macadam: 0.025

  d) Unpaved road: 0.05

  e) Field: 0.1-0.35

Tractive Force calculations:

Ra= Aerodynamic resistance force

W= Weight of vehicle

Rrf= Rolling resistance front wheel

Rrr= Rolling resistance rear wheel

Wf= Weight distribution in front wheel

Wr= Weight distribution in rear wheel

Ɵs= Angle of inclination

L1= length from front wheel to c.g

L2= length from rear wheel to c.g

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L= Total length b/w wheels

h= Height of c.g from ground

- (Fmax)r= Maximum tractive force in rear-wheel-drive vehicle
- (Fmax)r= Maximum tractive force in front wheel drive vehicle

By keeping the vehicle at an inclined position from horizontal at an angle of ϴs.

And by considering the forces acting on the vehicle and applying them on the vehicle we drew the free body diagram of it, after equating and solving the forces we got the traction force required for the vehicle to move from its rest condition when the friction, rolling resistance, center of gravity location in all three directions, track width and wheelbase, these values are acquired from vehicle dynamics team and design team [3]. We assume ϴs is very small i.e., ideal condition.

We also assume that the traction force in the front wheel is zero because the vehicle is rear wheel driven.

2.2 Assumptions

cosϴs=1

fr=0.013

μ=0.75

W=320*9.81 N

h=270mm

L=1560mm

L1=858mm

(Fmax)r=(μW(L1-fr*h)/l)/((1-μh/l))

(Fmax)r=1391.64 N

The maximum traction force is 1391.64N i.e. the force of 1391.64N is required to move the the vehicle from rest position to moving condition initially. As we assumed some parameters and excluded some other minor parameters which might affect the maximum traction force of the vehicle we are also considering the Factor of Safety of value 1.25.

The Required force is = 1391.34 * 1.25

= 1739.5 N

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So, the traction force required to move the vehicle is 1739.5 N.

2.3 Final Drive Ratio Calculation

All the calculations were carried out as per the guidelines given by standard text books [5], [6].

ix = Gear ratio

Tg = Torque after Gear box

io = Gear ratio of Final Drive (Sprockets gear ratio)

Td = Torque at differential

Twl = Torque at rear left wheel

Twr = Torque at rear right wheel

Fwr = Force(Tractive) at rear right wheel

Fwl = Force(Tractive) at rear left wheel

rw = Radius of the wheel

- From the Tire data,

  rw = 0.2761 m,

From the above tractive force calculation,

  Fw =1750 N, (875N for one wheel)

From engine specifications,

  Te =23.5 Nm (@4500rpm from the graph) [Assume the driver achieves 4000rpm at first gear],

  ix = primary gear reduction ratio * gear ratio of the 1st gear

  = 2.667 * 2.5

  ix =6.6675

Tw / Te = Total gear ratio (ix * io)

Tw = Fw* rw * 2 (Either we can take force(and torque) of both wheels and excluding the multiplication of 2 in the formula or we can directly take force of one wheel and multiply by 2)

(Must include the torque splitting at the differential, so the torque which is available at differential is split into two and the tire acquires half the torque)

Fw = (ix*io*Te)/(2*rw)

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By solving the equation, we get, io = 3.06

So we can take value of 3.69 for the gear ratio of sprockets assembly before entering the differential to achieve more torque.

2.4 Sprockets design, Force On Tooth of Sprocket Calculation

Our main aim is to reduce the loses, i.e. by decreasing the velocity difference we can achieve efficient velocity transmission with less losses, so in the above mentioned formula it is clear that the velocity difference is directly proportional to [1- cos(180 / z)], so by doing iterations and considering the practical feasibility of the fabrication of sprocket, 15 tooth sprocket is selected over other by satisfying the polygon effect.

As we require sprocket's gear ratio of 3.2 , at the engine the size of the sprocket has a dimensional constraint and including all the constraints It is assumed, that the tooth on the engine sprocket as 13 to be practically possible and feasible to be fit in that place.

Gear ratio = (no.of tooth on driven sprocket)/(no.of tooth on driving sprocket)

Therefore,

no. of tooth on driven sprocket = Gear ratio * no. of tooth on driving sprocket

= 3.69* 13

no. of tooth on driven sprocket = 48

Hence, the driving sprocket (or) sprocket placed at the differential will have 48T to be coupled with a driven sprocket (or) sprocket placed at the engine having 13T to give a sprocket gear ratio of 3.69.

Velocity ratios of chain drive is given by-

V.R = N1/N2

= T2/T1

N1 = Driving sprocket RPM

N2 = Driven sprocket RPM

T1 = Driving sprocket teeth

T2 = Driven sprocket teeth

The sprocket outside diameter,

(D0) = D+(0.8*d1) (For satisfactory operation)

d1 – diameter of chain roller

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Average velocity of the chain,

V = (π*D*N)/60 = (T*P*N)/60

The geometric details of sprocket are given in Table 1

3. Results and Discussion

After creating the geometry with the above calculations, analysis is carried out in Ansys.

3.1 Static Analysis

Since each tooth is equally spaced apart, it is first necessary to compute the force acting on only one tooth. Once the amount of the force has been determined, it is applied to the surfaces of the teeth in a direction tangential to the pitch circle of the sprocket teeth. Bolt holes also have fixed limitations attached to them. Ansys Workbench 18.2 was used for the analysis, and frictionless support is used in places where the differential is attached to a chain sprocket. The von-mises stress and deformation are shown in Fig. 2

Table 1: Sprocket geometry

P = Chain Pitch **N** = Number of Teeth **Dr** = Roller Diameter (See Table)	**yz** $= Dr\left[1.4\sin(17° - \frac{64}{N}) - 0.8\sin(18° - \frac{56°}{N})\right]$
Ds = (Seating curve diameter) = 1.0005 Dr + 0.003	**ab** = 1.4 Dr
R = Ds/2 = 0.5025 Dr + 0.0015	**W** $= 1.4\ Dr\ \cos\frac{180°}{N}$
A $= 35° + \frac{60°}{N}$ **B** $= 18° - \frac{56°}{N}$	**V** $= 1.4\ Dr\ \sin\frac{180°}{N}$
ac = 0.8 x Dr	**F** $= Dr\left[0.8\cos(18° - \frac{56°}{N}) + 1.4\cos(17° - \frac{64°}{N}) - 1.3025\right] - .0015$
M $= 0.8 \times Dr\cos(35° + \frac{60°}{N})$	**H** $= \sqrt{F^2 - (1.4Dr - \frac{P}{2})^2}$
T $= 0.8 \times Dr\sin(35° + \frac{60°}{N})$ **E** = 1.3025 Dr + 0.0015	**S** $= \frac{P}{2}\cos\frac{180°}{N} + H\sin\frac{180°}{N}$
Chordal Length of Arc xy $= (2.605\ Dr + 0.003)\sin(9° - \frac{28°}{N})$	**PD** $= \frac{P}{\sin\left[\frac{180°}{N}\right]}$

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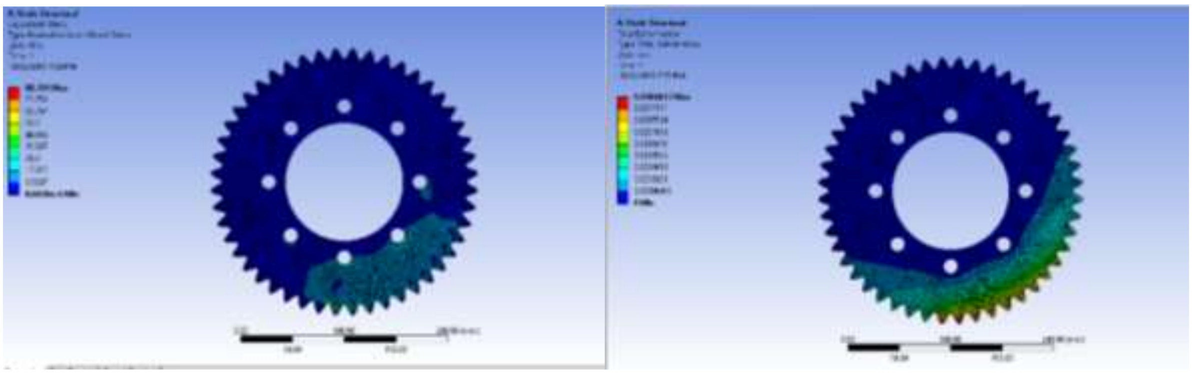

Figure 2: Plot showing Von-mises stress and deformation

3.2 Fatigue Analysis

To determine the sprocket's fatigue life, a fatigue analysis was conducted. After the shortest amount of fatigue life, the first crack evidence appears. For component design, it is crucial to take fatigue life into account. The fatigue tool from ANSYS Workbench was utilised to calculate fatigue life. Analysis of the stress-life (S-N curve) was done. For extremely accurate results, zero-based force and the Goodman mean stress theory were applied.

4. Conclusions

Using the results that are illustrated in the paper, the overall strategy is secure, convincing, portable, and suitable for the needs.Analysis results also become much more reliable, but other tests, like the exhaustion test and clasping, can still be carried out to find out whether our results are risky or reliable.As an alternative to AISI 4130 Different materials, including titanium and carbon fibre, can be used.The plan is limited by the carbon fibre and titanium compound because they are both pricey. It appears to be much more sturdy than the stuff we used. As a result, the qualified candidate may be selected depending on the requirement.Because AISI 4130 was chosen, the stacking circumstances were constrained; however, if a more grounded material had been chosen, the stacking requirements may have been expanded..

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