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Optimization of Differential System & Drivetrain Assembly for FSAE Car

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ABSTRACT: Traditionally, the differential system was not much preferred for a formula student car. A straight single unit axle (spool) was been used widely. The differential systems used were also taken as a stock unit from the commercial cars and installed directly in the FSAE (Formula Society of Automotive Engineering) car. We aim to optimize the previous FSAE differential system and the subordinate drivetrain assembly according to the forces acting on the system during acceleration, turning, and cornering by considering the racing conditions and not preferring the commercial car drivetrain assembly for the same.

KEYWORDS: Differential, Drag, FEA simulation, ISO chain, Spool, Sprocket, Track-width.

I. INTRODUCTION

Embracing the 21st century in stride, the automobile industry has adopted a step towards optimization of the vehicle to have a worthy result. Optimization is finding an alternative with the most cost-effective or highest achievable performance under the given constraints, by maximizing desired factors and minimizing undesired ones (1). The aspects which are considered to optimize are the study of materials, design overview of standard drivetrain system, drafting, and analysis of drivetrain comparing to the standard drivetrain system with keeping in mind the requirements of the vehicle accordingly.

The thinking behind optimization is to improve the performance of the vehicle with respect to weight to power ratio. The technique of improving the existing model is the target to be achieved successfully. The criteria for selecting the material is durability, economy, and weight. Thus the calculation of necessary wheel forces is calculated, secondary gear ratio improved, switched to standard ISO chain with necessary calculations, we used differential instead of the spool, try to improve efficiency by use of optimum velocity joints, design custom shafts according to the torsional forces and car track-width.

The current drivetrain system includes a differential, drive-shafts, drivetrain mounts, tripod velocity joints, and sprockets with spares of everything. The primary goal is to improve the drivability by optimizing the previous drivetrain system of the FSAE car according to the current need by maintaining a suitable balance of wheel torque and rpm. The chain drive is used and proper measures have been taken for focusing on chain alignment and rigidity. Suitable changes have been made in the secondary engine gear ratio. The differential system is being introduced and the usage of the spool is replaced. Velocity joints are being optimized to match the drivetrain requirements and also aiming for suitable weight reduction for the overall system. Drivetrain Axles are being customized according to the car track-width and the velocity joint bearing connection.

II. DRAWBACKS FROM THE PREVIOUS DESIGN

In order to make sure designs are simply reliable and simulated with simulations and being validated by physical testing. When considering it seems that the best design considerations would be the designs that could be made simpler, reliable, and help increase in the performance of the car.

The following drawbacks had been observed in the previous drivetrain assembly:

1. The secondary gear ratios had a higher ratio which provided us more than the sufficient wheel torque but affected speed (RPM) of the car. Also, the material selected for the sprocket increased the overall weight of the chain drive system.
2. The standard chain drive system used for the engine KTM 390 had a chain of a higher pitch which increased the size of the sprocket and eventually resulted in increased weight of the same.
3. The spool system generated a fishtail turning at the corners and also increased the wear rate of the tires.

4. The turnbuckles were used along with the drivetrain mounts due to inaccurate center-to-center distance between the sprockets to provide proper tension to the chain drive. The material for the drivetrain mounts increased the weight. Also, the FOS was comparatively higher as shown in fig. 01.
5. The drive-shafts used were of commercial cars that had a different track-width. Hence the shafts were adjusted by cutting according to the required track-width by cutting and welding a mild-steel sleeve as shown in fig. 02.
6. The constant velocity joints were used from commercial cars which were over-sized and the type of joint used was heavy.



Fig. 01 Shafts with welded mild steel sleeves and commercial constant velocity joints



Fig. 02 Drivetrain mounts with turnbuckle

The primary objectives of our current prototype are:

1. Calculating the necessary wheel forces.
2. Improving the secondary gear ratio.
3. Switching to standard ISO chain using necessary chain calculations.
4. Using differential instead of spool for improving the turning radius (2)
5. Improve the efficiency by using optimum velocity joints.
6. Designing custom shafts according to the torsional forces and car track-width.

III. METHODOLOGY

Studying basic wheel calculations of the vehicle and the various forces acting on it in static and dynamic conditions would help us in getting proper parameters to design and analyze the drivetrain components. These calculations helped us to know the magnitude of the forces which resist the vehicle from moving. Numerous forces act on the vehicle at a time, but considering the major forces and neglecting the other forces with comparatively very low magnitude for basic wheel calculation.

- (i) Drag force due to air (3)
- (ii) Resistive force due to rolling (4)
- (iii) Resistive force due to gravity.
- (iv) Resistive force due to inertia (4).

Regarding all the above forces, the secondary gear calculation is being done and the number of teeth on the driven and driver sprocket is decided. The number of teeth on the driving sprocket was taken 15 and 43 teeth on the driven sprocket was considered after the iterations. Standard ISO chain was selected and the use of the commercial bike KTM 390 chain was eliminated. The center-to-center distance between the driving and driven sprocket and the number of chain links is also calculated (5). An open differential system is used instead of a spool (2). The commercial differential systems are purely designed for particular purposes, using it for a different condition would not be preferred. The stock differentials are heavier due to their material selection. It also has different bevel gear teeth ratios which may not be suitable for the currently designed prototype. This will, in turn, lead to unstable cars around corners, and a miscalculated turning radius will be traced in practical. The high unbalancing force acting on the system would in turn affect the speed and torque transmission. Non-optimized drivetrain assembly will provide a variable performance result of the overall prototype. Taking suitable gear ratios according to the cornering forces improved the turning radius and hence the overall performance during turns. Also, having a mono-casing concept helps in reducing the overall weight of the drivetrain system and eliminates the space complexities by making the overall differential system compact. The differential mounts have a direct mounting to the chassis hence eliminating the use of tensioner with the help of the calculations for the center-to-center distance between the second gear ratios. Necessary reaction forces were calculated on both the mounts for analysis (5). The bearing selection was done too according to the obtained reaction forces. Commercial CV (constant velocity) joints were replaced with the Tripod velocity joints. Optimization was done in the material selection to reduce the overall weight of the drivetrain system. The shaft diameter is calculated and analyzed according to the torsional equation (5). Various iterations for obtaining a suitable diameter of the shaft, as well as a material selection under the torsional loads is been done.

IV. DESIGN OVERVIEW (DRAFTING AND ANALYSIS)

Secondary gear ratio (Sprocket):

Structural Analysis for the selected secondary gear ratio according to the calculated wheel torque and RPM was necessary in order to look upon the load carrying capacity and behaviour of sprocket under bending stresses as shown in fig. 03 (6). A factor of safety of 1.7 with necessary weight reductions was obtained. The final design had a sprocket teeth number of 43 instead of the previous design having 48 teeth thereby optimizing the speed and required torque of the car.

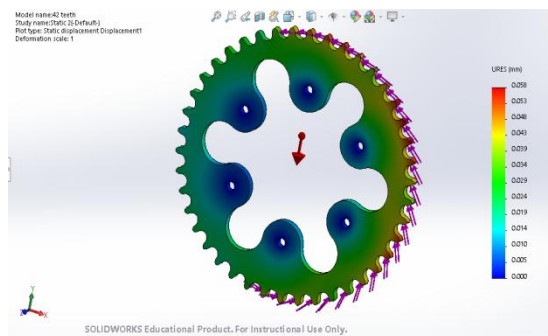


Fig. 03 Sprocket analysis

Secondary driving chain:

The previous design used a 15.875 mm pitch chain. This was optimized by using a 12.7mm which lead to decrease in the overall sprocket pitch circle diameter from 242.725 mm to 173.98 mm.

Open Differential:

Various design iterations were done in order to eliminate the outer casing from the commercial differential design and introduce a mono-casing design. The mono-casing differential concept was drafted and necessary static analysis was done. The casing supported the mild steel differential gears as well as provided a covering for preventing the leakage of lubricant for the gears at once as shown in fig 04.

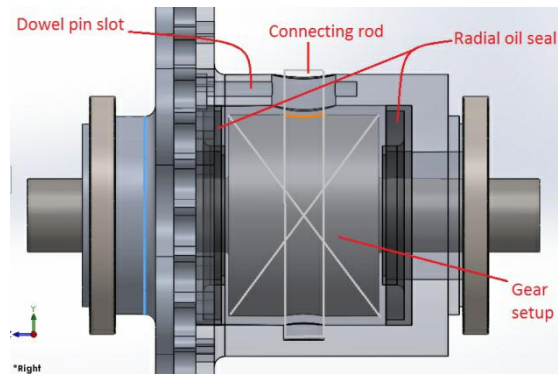


Fig. 04 Mono-casing differential setup

Tripod velocity joints:

Tripod velocity joints were drafted and analysed with the wheel torque obtained under various graded materials other than the standard mild-steel. Material selected for the same was graded aluminium. Steel hexagonal inserts were introduced at the tail of the joints in order to prevent the wearing of aluminium joints while in contact with the bevel gear shafts as shown in fig 05. Necessary structural analysis was done at the contact region of steel inserts and aluminium joints and final results shown in fig 06. By suitable material analysis and tripod bearing selection, the weight of a single velocity joint was reduced from 1300 grams to 600 grams.

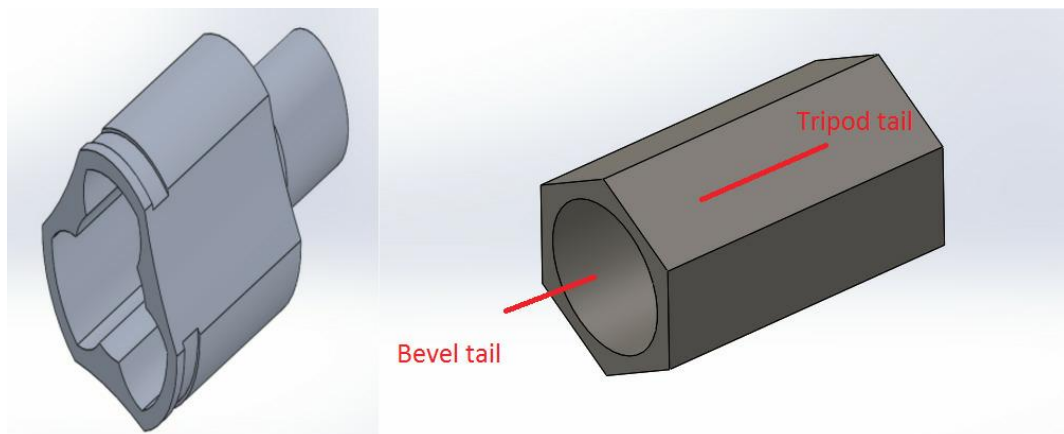


Fig. 05 Tripod velocity joints with hexagonal inserts

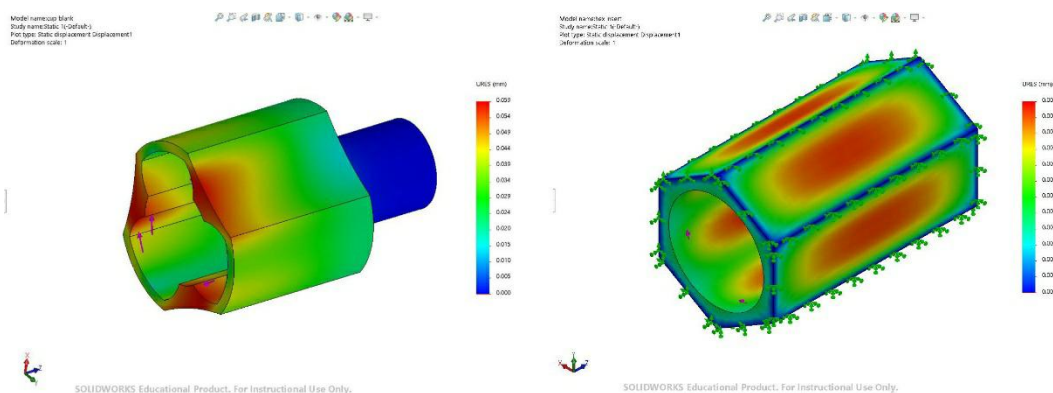


Fig. 06 Tripod cup & Hexagonal insert material displacement due to applied load

Drivetrain Mounts:

The reaction forces obtained by considering the overall sprocket force and the drivetrain assembly weight was analysed by creating basic mounts design and then optimizing it according to the stress concentrations (7). Necessary weight

reduction was also done as shown in fig. 07. Taking graded aluminium as the material rather than previous mild steel mounts, the overall weight per mount reduced from 920 grams to 300 grams.

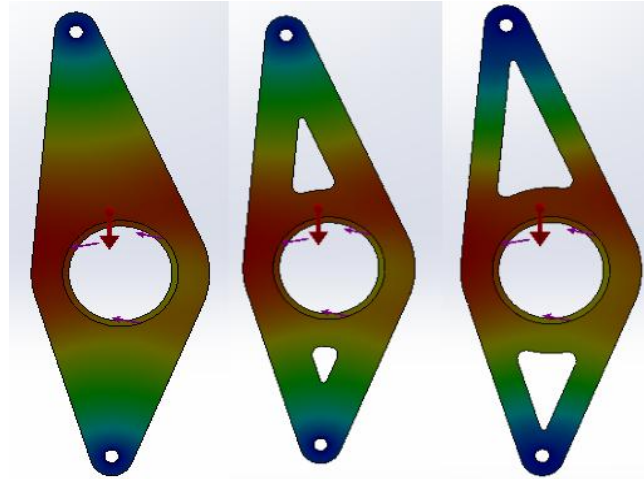


Fig. 07 Necessary weight reduction

Drive shafts:

The torsional load obtained according to the diameter and material chosen was being analysed according to the track width of the prototype (8). Various iterations were done with material as well as the dimensions in order to get optimum diameter for the required power transmission as shown in fig. 08. The final diameter fixed was 21mm rather than the previous 24mm diameter drive shafts which helped in a smooth transmission as well as reducing the overall weight of the single shaft from 6270 grams to 4800 grams. Choosing tripod velocity joints helped in easier assembly as well as improved the serviceability of the same. The final assembly CAD model shown in fig. 09.

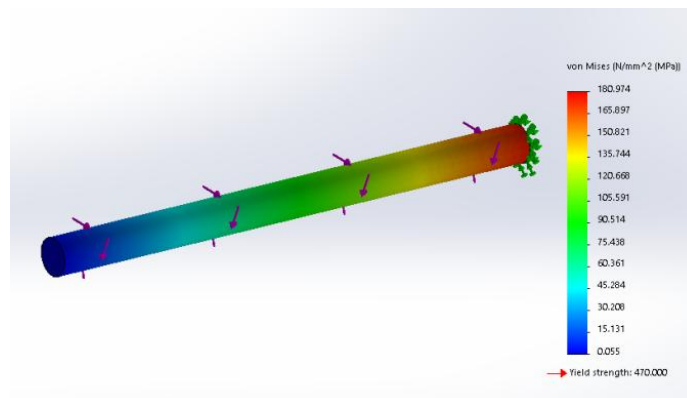


Fig. 08 Torsional analysis of shafts

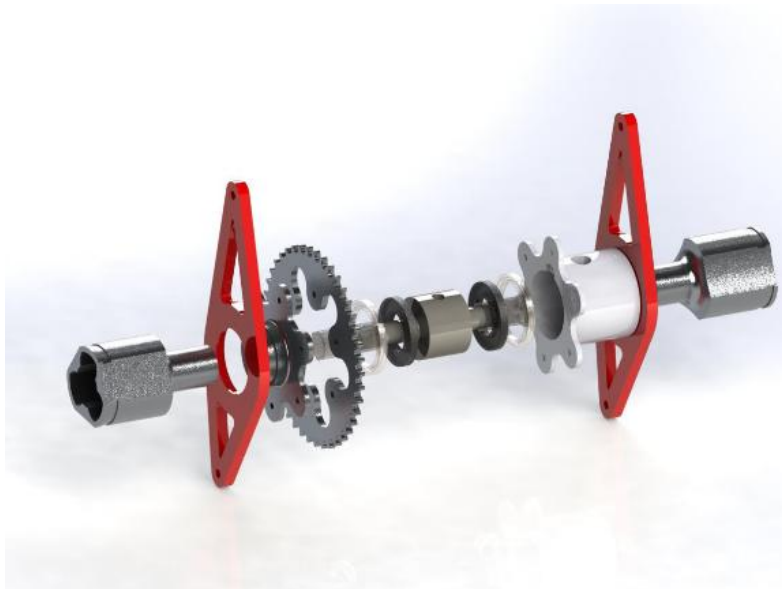


Fig: 09 final assembly CAD model

IV. RESULTS

The current gear ratio improved the wheel RPM from 318 RPM to 364 RPM along with the required torque thereby improving the overall performance. The engine rpm and gearbox shifting rpm graph show in fig 10. Using standard ISO chain helped in making the design compact by reducing the overall size of the chain drive. Eliminating the commercial differential would help in providing a calculative turning radius from the rear end of the prototype. Aluminium tripod velocity joints had 60% lesser weight than the commercial mild steel joints. Custom shafts eliminated the steel sleeve which was used in the previous design hence keeping the machining simple. The overall optimization of all transmission part shown in table 1.

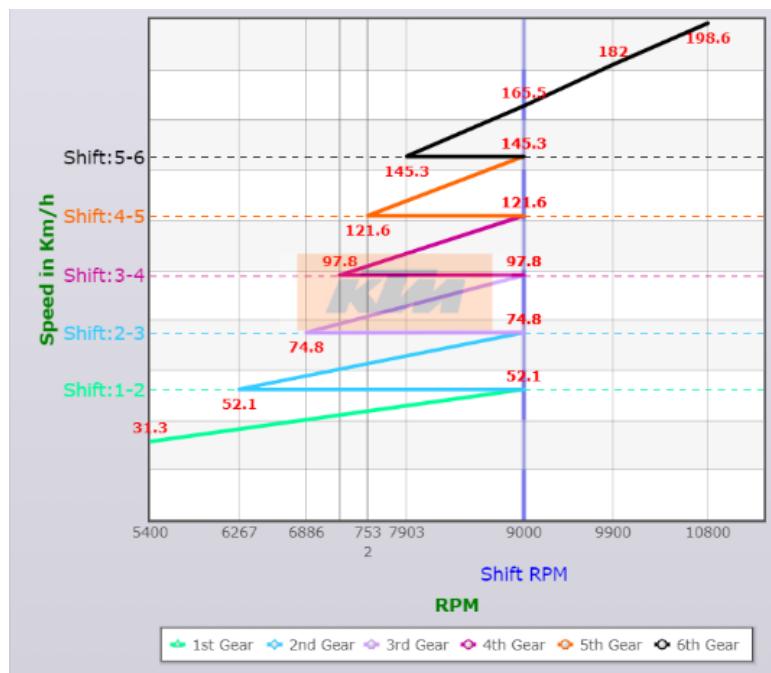


Fig: 10 Graph of Engine RPM v/s Gearbox shifting RPM (15:42)

Sr. No	Parameter	Previous	Current
01	Wheel RPM	318.603	355.65
02	Wheel torque (N-m)	796.44	713.48
03	Sprocket centre-to-centre distance (mm)	280	218.61
04	Chain tensioner Turnbuckles	Yes	No
05	Velocity Joint	Rzeppa	Tripod
06	Differential	Spool	Open
07	Overall assembly weight (Kg)	14.4	6.9

Table 1: Overall optimization table:

V. CONCLUSION

Increased weight, reduced efficiency, and unwanted forces acting on the system only lead to a decreased output of the overall system. Comprehensive design of the drivetrain system by eliminating the gear transmission and replacing it with the chain drive would in turn lead to a proper and optimized system for the desired prototype. A commercial drivetrain system is designed for the long run, and in turn has a higher factor of safety, which is installed in the FSAE car would lead to an increase in weight factor of the complete prototype. A designed drivetrain would in turn help in eliminating the issue of space complexity. The serviceability of the system would improve due to proper analysis and calculation of the forces acting during the running conditions of the prototype.

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