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Modelling and Analysis of Electric Powertrain using Open Modelica and ANSYS Workbench

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ABSTRACT: The demand for vehicles with Alternative Propulsion technologies has increased in the recent past due to the rapid decline of fossil fuels caches, and an increased awareness of the impact of emissions on our environment. Electric vehicles (EV's) however, have lower acceptance than conventional gasoline engine vehicles due to factors like initial cost, range, and life. The implementation of a gear box in low cost, BLDC motor driven EV's will result in a better performance by balancing the torque and speed requirements during different situations. The gearing was chosen based on the Traction diagram of the given Motor-Vehicle configuration generated in MATLAB. Further, a Mathematical model of the complete electric Powertrain was simulated in the OpenModelica software to generate the required input torque characteristics of the motor for the chosen gearing, so as to compare it with actual motor characteristics. The gears were analyzed in the Static structural condition of the ANSYS workbench to check for the equivalent stresses, deformation and contact frictional stresses. The simulations and analysis mentioned were carried out for an Open-wheel EV, and the design proven safe for the chosen operating conditions, before moving on to the detailed designs.

KEYWORDS: Electric vehicles (EV's), Traction diagram, BLDC, OpenModelica, ANSYS Workbench, Open-wheel.

I. INTRODUCTION

Powertrain in any vehicle is needed to enable the vehicle to move from rest. It consists of power plant, clutch, gearbox, final drive, axle and wheels. It plays an important role in terms of acceleration and speed of the vehicle. The design of Powertrain has to be made with due consideration of the operational constraints of the complete system. The Commuter and Commercial Vehicle segments of the automotive industry are the major constituents of the 'mass-market'. The success of a product in these segments (in developing markets) depends on the perceived value of the product, the perception being based majorly on its visual and tangible features. This makes it imperative for a manufacturer to allocate lesser (and cheaper) resources towards the development of the Powertrain of the vehicle, when compared to its other subsystems. This kind of allocation impacts the Hybrids and pure Electrics more so, where the cost of the Battery and the Motor are a significant part of its overall price. Thus their sale is impeded when compared to the more established ICE vehicles of a similar class. Considering the prevailing scenario, it becomes all the more necessary to develop mechanical systems that can extract the most out of lower powered Electric motors at all their operational frequencies, thus providing a suitable compromise between the acceleration and top speed characteristics of the vehicle. Open-wheeled cars are lending themselves to be the ideal test rigs for any desired experimental layouts of the Powertrain due to their 'bare-bones' nature and the absence of design constraints normally applicable to Passenger vehicles. This also makes it easier to identify and locate the anomalies, if any, in the overall assembly of the Powertrain without the risk of admittance of noise from nearby subsystems. Hence a similar class of vehicle is considered in the development of our analysis.

Vehicle modelling and simulation in general, and electric and hybrid vehicle modelling in particular, are a field of engineering in which Modelica excels, because of its inherent capability to simulate multi-domain systems, from the ground up. ANSYS checks the structural integrity of the critical components of the system to ensure safe operability.

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II. RELATED WORK

There has been immense research carried out in the field of power transmission, and hence a large amount of literature on gear modelling and analysis has been published. The report by Staffan Enocksson [16] introduced techniques of modelling a gearbox in Simscape, which were used to implement the same on an open-source platform (OpenModelica). The gear stress and deformation was found by meshing and analysis by Ghosh .et al. [13] for a single tooth of the gear in Static Structural condition, for different vehicles at specified torques. The maximum contact stress was determined using Hertz equation and Finite Element Method by Mushiri .et al. [14].

III. PROPOSED METHODOLOGY

For the vehicle Powertrain to be designed, we first come up with its realistic target performance characteristics.

i. Targets:-

- +0.6g longitudinal acceleration
- 70 kmph+ top speed
- 10⁰ Gradability
- Minimal-to-zero driver input in Powertrain functions

ii. Assumptions:-

- Motor → Brushless DC, LBC 125-8D series, 72V, 6kW
- Dimensions (l x b x h) = 342.8x140x140 mm
- Rated torque = 18 Nm
- Peak torque = 54 Nm
- Rated speed = 3000 rpm
- Shaft diameter = 24 mm

iii. Vehicle specifications:-

Mass of car, M	300kg(with driver, worst case)
Coefficient of friction (sliding) between road and tire, μ_s	0.7
Tire rolling diameter, d_r	530mm
Centre of gravity height, h	360mm
Wheelbase, w_{bs}	1570mm

iv. Preliminary design:-

1. Considering the above goals, we first find the maximum torque that the wheels can sustain (without sliding) during acceleration.

$$X = h/w_b = 0.2293$$

Dynamic axle weights:

$$M_{dyn_f} = ([\{1-0.6\}] - (X*a))*M = 78.73 \text{ kg} \quad (a=0.6 \text{ g units})$$

$$M_{dyn_r} = M - M_{dyn_f} = 222.27 \text{ kg}$$

$$\text{Traction limit of tyres} = (\mu_s * (M_{dyn_r}) * g) * (d_r/2) = 404.48 \text{ Nm}$$

The upper limit for the Torque at wheels (both combined) is now set to 450Nm to account for losses in the Powertrain. Each wheel therefore receives 225Nm (50%) at any given time for an open-differential.

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2. Gear reduction for achieving the torque is determined. The maximum speed achievable for this reduction is found.

$$\text{Gear Ratio} = 450/54 = 8.33$$

$$\text{Wheel rpm} = 3000/8.33 = 360.14\text{rpm}$$

$$\text{Road speed} = (\pi \cdot d_r \cdot N/60) \cdot 3.66 = \mathbf{36.6\text{kmph}}$$

3. The top speed is insufficient with regards to the goals set above and hence, we need another gear which pushes the speed up to the desired value.

On back calculating,

$$\text{Wheel rpm at } 70 \text{ kmph}$$

$$= 19.44 \cdot 60 / (\pi \cdot d_r) = 700\text{rpm}$$

$$\text{Gear ratio} = 3000/700 = \mathbf{4.28}$$

4. The net resistance (Torque-Speed) plot is drawn in MATLAB for speeds up to 80kmph in terms of wheel rotations per minute. On the same graph, a general BLDC motor characteristic curve is plotted. This curve incorporates the two gear steps as determined above. The point where the resistance and motor curves intersect gives the top speed, while the area between the two is used to determine the Drawbar pull, and subsequently, the acceleration achievable at every speed.

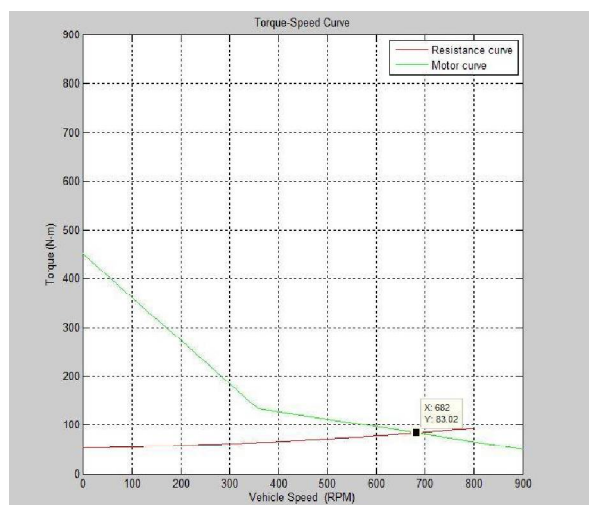


Fig (1.1)
Traction diagram

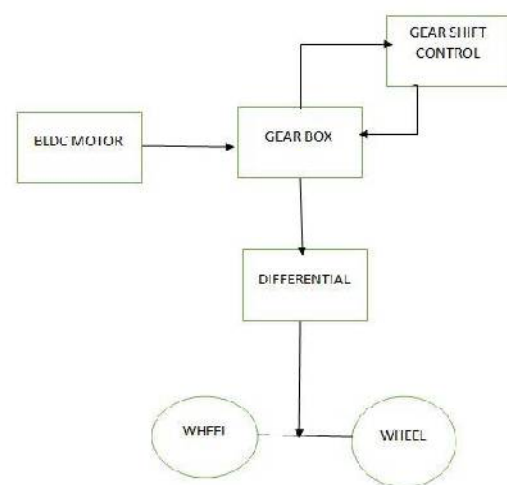


Fig (1.2)
Powertrain layout

5. The overall gear ratios are finetuned (trial & error) using the graph in Fig (1.1), to maximize the net area under the curve as well as the top speed achievable. A shift speed at which the gear change takes place is also automatically fixed at the intersection of the two motor characteristic curve's step.

6. Since the 1st Gear ratio chosen is high, it needs to be stepped so as to avoid excessive stresses on the Powertrain. The steps are chosen such that they don't exceed more than 3:1. Here we also introduce a sprocket-driven open differential which will serve as a secondary reduction/Final drive. The basic outline of the Powertrain is now described in the form of a schematic as seen in Fig (1.2). For the applied limit loads on Powertrain, a gear module of 2mm is chosen. The two gear ratios for the first reduction are chosen from the preferred values tables in the design data handbook, and adjusted for matching the distance between the two gearbox shafts.

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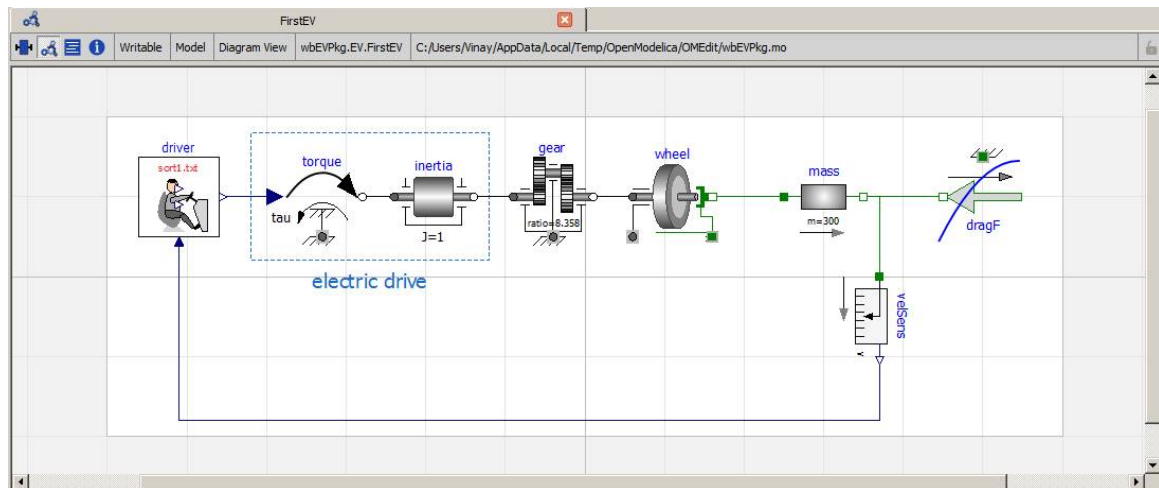


Fig (1.3)
Powertrain Model – Open Modelica

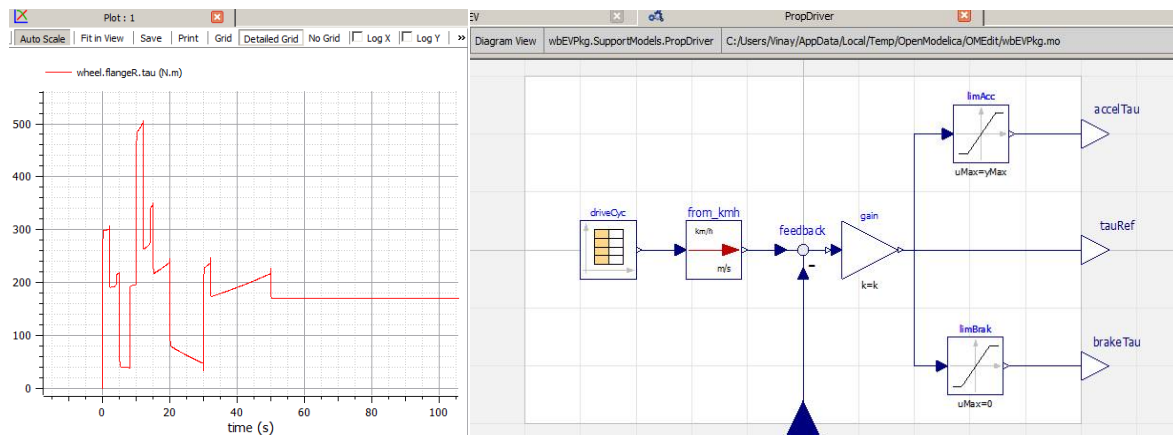


Fig (1.4)
Wheel Torque Characteristics

Fig (1.5)
Driver Sub-class Model

7. At this point, we develop the Modelica model of the Powertrain using the derived overall Gear ratios to check for correspondence with the available motor characteristics, as seen in Fig (iii). If there is a mismatch, the gearing (or motor output if possible) is modified iteratively to bridge the gap. The driving cycle selection is an important step in the design of this model since it aids in the evaluation of fuel consumption, emissions, and performance characteristics of the vehicle. A driving cycle commonly represents a set of vehicle speed points versus time. For our application, we develop a cycle where the average acceleration values are close to the chosen target values. The plot of the wheel torque in Fig (iv) shows that the peak torque values are within the limits of the MATLAB derived torque values, hence validating the thus far developed design.

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v. Final values:-

Overall Gear ratios		Top speed	Average acceleration	Gradability	Shift speed
1 st	8.358	68kmph	0.5g	34.17 ⁰	36kmph
2 nd	2.470				

GEARBOX:

Gear #	Gear Ratio	Number of teeth(Driven/Driver)	Diameter in mm (Driven/Driver)
1 st Gear	2.900:1	87/30	174/60
2 nd Gear	0.857:1	54/63	108/126

Shaft centre distance - 117mm

FINAL DRIVE/SPROCKET:

Sprocket speed reduction - 2.882:1

Sprocket	Number of teeth	Diameter (mm)
Driving	15	76.2
Driven	43	217.5

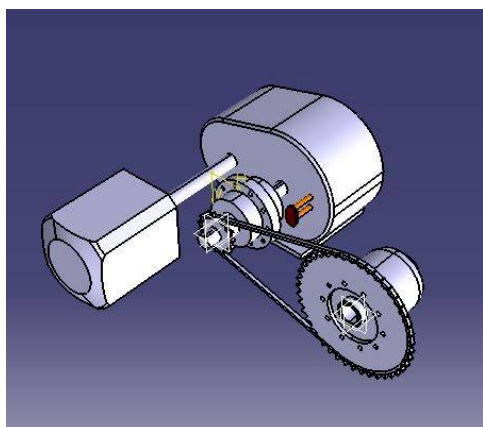


Fig (2.1)
CATIA model of Powertrain

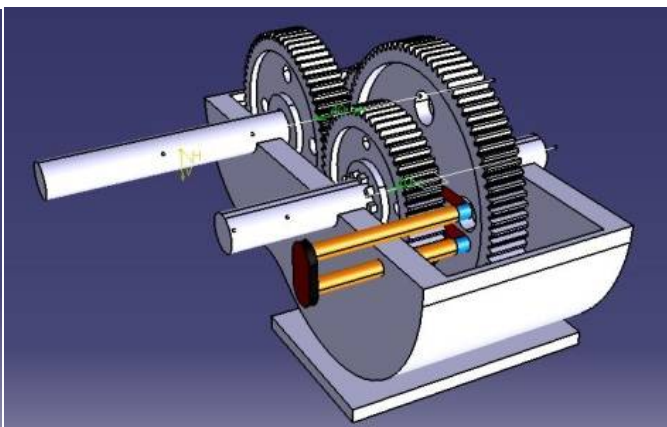


Fig (2.2)
CATIA model of detailed Gearbox

A crude rendition of the Powertrain is pictured in Fig (2.1) with the detailed view of the 2-Speed sliding mesh Gearbox in Fig (2.2). The chain pitch and bushing were chosen in accordance with the standard load/rating tables from different manufacturers. Sprocket for the driven side is fabricated to meet the bolting dimensions of the open differential cage, while the differential is chosen from a front wheel drive TATA IRIS since its power and torque ratings were closest to our application.

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IV. NON-LINEAR ANALYSIS IN ANSYS WORKBENCH

i. Methodology

The Non-linear analysis was carried out on two sets of gear pairs i.e. 87/30 and 54/63 in ANSYS R18.2. The gears were in mesh with their respective pairs, with frictional co-efficient of 0.2 between the contacting faces. The element mesh was fine with Proximity and Curvature conditions chosen. The driver gears of 30 teeth and 63 teeth were subjected to fixed support, while the other gears were on frictionless support. The remote points were created and a moment of 54Nm was applied at the respective driver gears for both the ratios. The non-linear solution was obtained to check for Equivalent stresses, Total Deformation and Contact Frictional stresses.

ii. Discussion and results

Alloy Steel (EN-19) is used in the manufacturing of the Gears blanks and Shafts. The Gears also undergo Case Hardening and the subsequent Heat treatment for stress release, after they've been hobbled for generating the Involute-profiled teeth. Wire cutting is employed in machining the splines on the shaft required for the Sliding-mesh mechanism.

Mechanical Properties (subject to ruling section)					
Condition	Tensile N/mm ²	Yield N/mm ²	Elongation %	Izod KCV J	Hardness Brinell
S	775-925	555	13	22	223-277
T	850-1000	680	13	50	248-302
U	925-1000	755	12	42	269-331

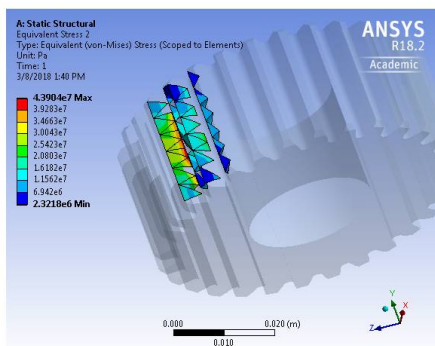


Fig (3.1)

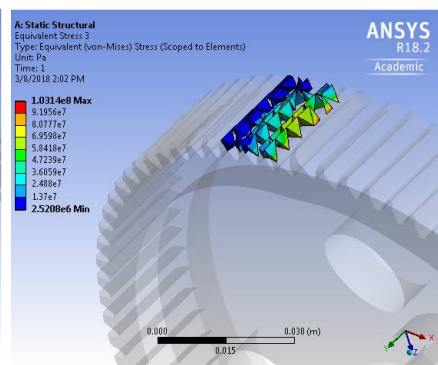


Fig (3.2)

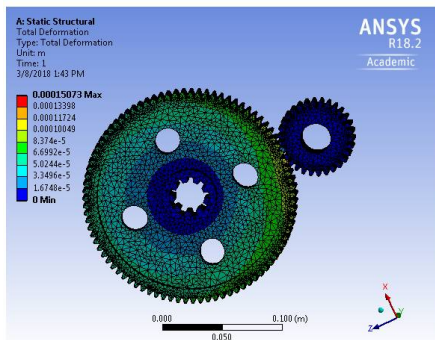


Fig (3.3)

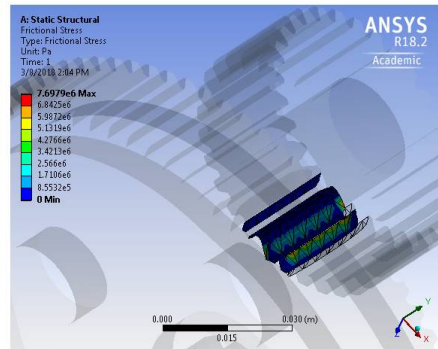


Fig (3.4)

In the Fig (3.1) and (3.2) the Elemental equivalent stresses were obtained by hiding one of the gears from the pair. The stresses obtained were 43.9MPa and 103.13MPa which are much less than the material's yield strength of 555MPa. In Fig (3.3) the Total deformation was obtained as 0.15mm and in Fig (3.4), the frictional stress at the contact region is 7.69MPa.

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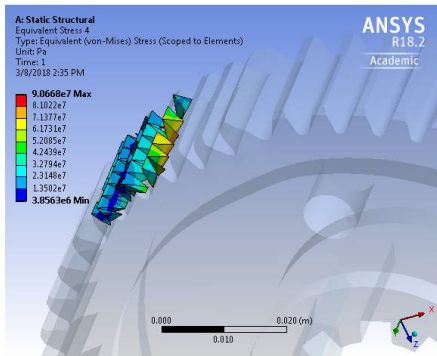


Fig (3.5)

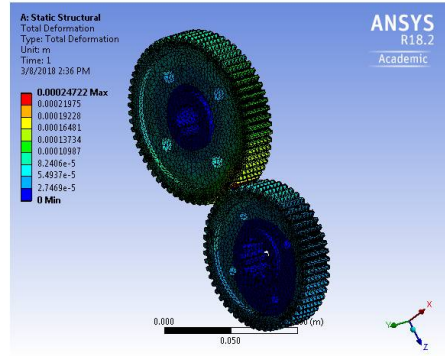


Fig (3.6)

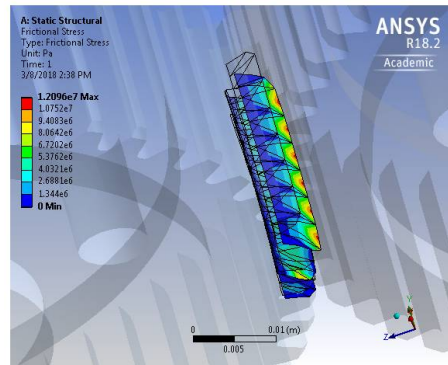


Fig (3.7)

In the Fig (3.5), the Elemental equivalent stresses were obtained by hiding one of the gears from the pair. The stress obtained was 90.68MPa which is much less than the material's Yield strength of 555MPa. In Fig (3.6) the total deformation was obtained as 0.247 mm and in Fig (3.7), the Frictional stress at the contact is 12.096MPa.

A few engineering drawings and the final prototype are shown below:

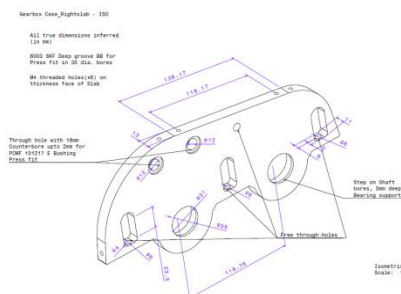


Fig (4.1)
Gearbox Shaft support Slab

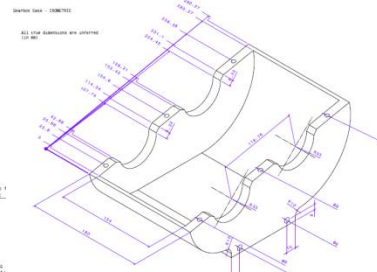


Fig (4.2)
Gearbox Case



Fig (4.3)
Physical assembly prototype

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V. CONCLUSION

A simplified approach has been devised to design and validate a Powertrain, and its subsystems using the given operational constraints. The mathematical model developed in OpenModelica serves as a base for future analysis involving a higher resolution of the details that go into modelling a real physical system. The gears were analysed in the Static Structural condition using FE software and were found to be safe for the given operating conditions. The deformation was minimal and the maximum stress was below the Yield stress of the material. The contact stresses due to friction were also considered and they were found to be below the Yielding limit due to Shear. Both sets of gears were designed with a Safety Factor of more than 2.5.

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