Design and topology optimization of a rack and pinion steering system using structural and vibrational analysis

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Abstract. The rack and pinion steering system, in automobiles, is the most commonly used steering system. Being directly connected to the suspension system in most vehicles, it is a critical component. The main intent of this research is to design a rack and pinion steering system for an All-Terrain Vehicle (ATV) and to reduce its weight by topology optimization. The design calculations are performed using AGMA bending, and contact stress equation and modelling is done using SolidWorks. ANSYS is used for structural analysis and the stress and deformation distribution is visualized. Vibrational analysis from ANSYS is utilized to obtain the natural frequencies, which are compared with the operating frequencies of the system. Topology optimization is conducted from this analysis to obtain an optimal and lightweight design.

Keywords: Rack and Pinion, Steering, FEA, Topology Optimization

1. INTRODUCTION

Due to its simplicity in construction and compactness the rack-and-pinion steering is one of the most recurrent type of steering systems. Its implementation is mostly in, but not limited to cars, small trucks and SUVs [1]. It is a simple mechanism that directly transforms the rotational motion of the steering wheel into linear movement at the wheels [2]. It constitutes a rack-and-pinion gear set enclosed inside a metal tube, with each extremity of the rack protruding from the tube. A rod, known as tie rod, attaches to each end of the rack. The pinion gear, linked with the steering shaft, turns in tandem with the steering and consequently moves the rack. The tie rod at the extremities of the rack attaches the spindle with the steering arm.

The objective of this study is to design the Rack and pinion gearbox for an All-Terrain Vehicle while reducing its weight by topology optimization. Weight is considered as one of the most important parameters in an All Terrain Vehicle [3]. Less weight leads to better acceleration, more fuel efficiency, less material costs and better handling characteristics [4].

Topology optimization is a mathematical method to optimize material arrangement contained by a given design space, directed at a particular set of boundary conditions, loads and constraints [5]. It implements Finite Element Analysis (FEM) to evaluate design performance and optimizes the component geometry [6].

The original component is designed using AGMA stress equations after assuming parameters for a typical All Terrain Vehicle [7]. Two topology optimization iterations are performed on this model to give an optimal design. All the iterations and the original model are compared on the basis of mass, maximum deformation, maximum Von Mises stress, and the range of natural frequencies.
The Vibrational Analysis performed helps in determining the range of natural frequencies [8, 9]. As the steering system is directly attached to the wheels through the suspension system, it faces large vibrational loads [10]. According to previous works, the values of vibration for operating conditions are 20 Hz to 200 Hz [11-13]. If these vibrations are close to the natural frequencies, resonance can occur leading to large deformations [14]. Thus, this paper aims to iterate a model having a minimum range of natural frequencies, far from the operating frequencies in addition to the structural analysis for deformation and stress.

2. DESIGN
The rack and pinion steering system has been designed using the AGMA stress equation and modeled in SolidWorks CAD software [15, 16].

Calculation – AGMA Stress Equation
Material taken is alloy steel AISI 4340 with tensile yield strength = 710 MPa
Consider 1 rotation of the pinion = 4 inch displacement of the rack
Torque applied = 15Nm
N Pinion = 60 rpm
Length of teeth on rack = 6 in, Total length of rack = 14 in
\[ L = \pi d_1 = 3.14 d_1 \]
\[ \therefore 4 = 3.14 d_1 \]
\[ \therefore d_1 = 1.28 \text{ in} = 32.6 \text{ mm} \approx 33 \text{ mm} \]
Assuming: module = 3
\[ \Rightarrow N_1 = 33/3 = 11 \text{ teeth} \]

Considering lock to lock steering in 1.5 turns
Rack length = 1.5 \times 4 = 6 in, Circumference of gear equivalent = 6 in = 152.4 mm
\[ 152.4 = \pi d_2 \]
\[ \therefore d_2 = 48.51 \text{ mm} \approx 48 \text{ mm} \]
\[ \therefore N_2 = 48/3 = 16 \text{ teeth} \]
Steering gear ratio = 1.45
From AGMA stress equation
Pitch line velocity \( V = \frac{\pi d_1 N_1}{60000} = \frac{\pi \times 33 \times 60}{60000} \)
\[ V = 0.1036; \quad KV = 1.02 \]
Power = 94 W
Ft = 94/0.1036 = 907.16 N; J = 0.34;
Endurance limit \( \sigma_e = \sigma_{yt}/2 \times K_L \times K_r \)
Take \( \sigma_{yt} = 710 \text{ MPa} \);
\[ \sigma_e = 351.12; \]
Taking FOS = 2;
\[ \sigma = 175.56 \text{ MPa} \]
\[ 155.81 \times 103 = (907.16 \times 1.02 \times 2.25 \times 1.6)/(3 \times 10^4 \times 0.34) \]
\[ \therefore b = 20.96 \approx 21 \text{ mm} \text{ (face width)} \]

3. MODELLING AND ANALYSIS
Three models were created and analyzed in this study; one original design and two iterations. Boundary conditions used while performing FEA are
1) Fixed support for the central region (supported by the casing)
2) Vertical loads of 500N simulating the tie rod forces are applied to the eyes of the rack
3) Mesh parameters used are – Adaptive, Relevance center 75 and size 5mm

![Figure 2. Mesh Parameters used](image)

### 3.1 Original Design

The dimensions obtained from the calculations are used to create a CAD model. This is referred to as the Original design in this research. From the Evaluate tab in SolidWorks, the mass of this model is obtained. Mass = 995.34 grams

![Figure 3. SolidWorks model of the original design](image)

![Figure 4. Total Deformation plot for Original design](image)
The load conditions are simulated and the original model is analyzed using ANSYS software [15]. The total deformation is visualized using this contour plot. The highest value of deformation obtained is at the ends of the rack. Maximum total deformation = 0.1657mm

Figure 5. Equivalent stress plot for Original design

Using the same boundary conditions and loads, contour plot for equivalent Von Mises stress is obtained. The largest values obtained are close to the fixed support while the lowest values obtained are at the rack ends. Maximum stress = 91.578 Mpa

Figure 6. Vibrational analysis for Original design

Vibrational Analysis was performed using the Modal analysis system in ANSYS [16]. The range of natural frequencies and the deformation at each frequency is visualized using the contour plot. Natural frequency range = 764.36 to 4550.1 Hz

3.2 Iteration 1

Figure 7. SolidWorks model of Iteration 1
The original model has less stress in the regions towards the ends of the rack and close to the axis. Thus, slots were added at those locations. For this iteration, all the slots are of the same thickness. Mass=836.49gram

Similar to the earlier plot, a total deformation plot of the first iteration is obtained using ANSYS. The deformation is maximum at the ends and minimum close to the center. Maximum Deformation= 0.20993mm

The contour plot for Equivalent Von Mises stress shows better optimization than the Original design. However, there are still some regions with less stress present such as the end regions. Maximum stress is still at the teeth near the central support. Maximum Stress= 102.97 Mpa.
The characteristic deformation and the shape for vibration analysis of Iteration 1 are similar to those of the Original design. However, they have different range of frequencies. The shape of the 1st mode of vibration is visualized in this plot. Natural frequency range = 687.04 to 3023.4 Hz.

3.3 Iteration 2

![Figure 11. SolidWorks model for Iteration 2](image1)

The two slots at the ends still had regions with less stress around them and had potential for further weight reduction. Thus, the two end slots were made bigger for this iteration. Mass = 780.66 grams.

![Figure 12. Total Deformation Plot for Iteration 2](image2)

The Iteration 2 is analyzed using ANSYS software to give the total deformation plot. The highest value of deformation obtained is at the ends of the rack and the lowest is near the central support. Maximum deformation = 0.2448 mm.

![Figure 13. Equivalent Stress plot for Iteration 2](image3)

The stress plot for Iteration 2 shows a more efficient stress distribution than the earlier models. The weight reduction is optimized and there are lesser regions with low stress. Max stress = 102.45 MPa
4. RESULTS AND DISCUSSION

A rack and pinion steering system was designed using AGMA stress equations and modelled using SolidWorks. Structural and Modal analysis was conducted using the FEA software ANSYS. Iteration based Topology Optimization was performed with an aim to reduce weight of the component while allowing a marginal increase in maximum stress. The mass, maximum deformation, maximum stress and the range of natural frequency were analyzed and compared for the original design and the two iterations [17, 18]. The results for all three models are tabulated for comparison and further analysis.

| Analysis                      | Original Design | Iteration1 | Iteration2 |
|-------------------------------|-----------------|------------|------------|
| Mass (grams)                  | 995.34          | 836.49     | 780.66     |
| Maximum deformation (um)      | 165.7           | 209.93     | 244.8      |
| Maximum stress (Mpa)          | 91.57           | 102.97     | 102.45     |
| Natural frequency range (Hz)  | 764.36 to 4550.1| 687.04 to 3023.4| 648.16 to 2836.1 |

Iteration 1 gives a 15.96% reduction in mass from the original model. The deformation and the stress are increased by 26.69% and 12.44% respectively. The range of the natural frequency is 2336.36 Hz. Iteration 2 has 23.58% less mass than the original model. Deformation and stress increase by 47.74% and 11.87%. The natural frequency range is 2187.94 Hz.

The natural frequencies of all three models do not match with the operating frequencies of 20 Hz to 200 Hz so there is no possibility of resonance. However, there could be sudden changes in the operating frequencies in real world conditions such as due to irregular road surfaces and unbalanced or worn out tires [19]. Therefore, a model with a smaller range of natural frequencies is preferred.
Iteration 2 is an optimal design because it gives the most reduction in weight. It also has the smallest range of operating frequencies so it is the preferred model from Vibrational Analysis. However, it has a small increase in maximum stress over the original design and the largest deformation. If deformation is not a critical parameter, then Iteration 2 is the best model.

The proposed method used for the modeling carried out in this project can be used in investigations to solve the system faster than by using experimental methods [20]. The performance of different types of suspension and steering systems can be analyzed and compared using this procedure. Additionally, the methodology can be applied to the entire front suspension system of a car for better optimization of all components.

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