

Design and Analysis of Portal Gear Lift

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Abstract - Portal gear lift concept is usually used in off-road driving condition. This gearbox takes power from the main engine and transfers it to the wheel. It is usually installed at the wheel end side of the vehicle in between the wheel and the axle shaft so as to increase ground clearance. As it transfers high amount of power so to ensure safety, different analysis and simulation need to be perform to check its behavior. In this study, we analysed input gear tooth to calculate bending stress acting on it. Also calculated the contact stress between the input gear and idle gear through FEM analysis. Theoretically calculated results and FEA results were compared in both the cases and shown in a graph. This concept is applicable for any off-road driving vehicles for example All-terrain Vehicle (ATV). As ATV pass through so many obstacles, portal gear lifts can be an option to avoid those obstacles.

Key Words: Portal Axle, Ground Clearance, Bending Stress, Contact Stress, All-terrain Vehicle (ATV).

1. INTRODUCTION

Portal gear lifts are used in vehicles which are to be operated in off-road conditions. In off-road conditions there are more obstacles on the road compared to normal road conditions, hence it is necessary to protect the underneath components of the vehicles [1]. By using portal lift gearbox, we get higher ground clearance without compromising with the constraints of constant-velocity joint (CV) axle or power transmitting axle. This can be done by connecting axle shaft to the wheel hub with help of set of gears where minimum two number of gears should be used so as to increase the center distance. As the set of gears is placed above the wheel hub it effects in raising the axle and differential housing from the ground and ultimately it results in lifting the body of vehicle.

Portal gear lifts mainly improves the off-road capabilities of the vehicle while keeping the track width as small as possible, which results in compactness of the vehicle without compromising with

other components. Also, we can differ the torque transmitted to the wheels according to our requirements. Portals transfer the force of the engine more closely to the wheels, so there is less stress on the drive shaft and the factory transmission.

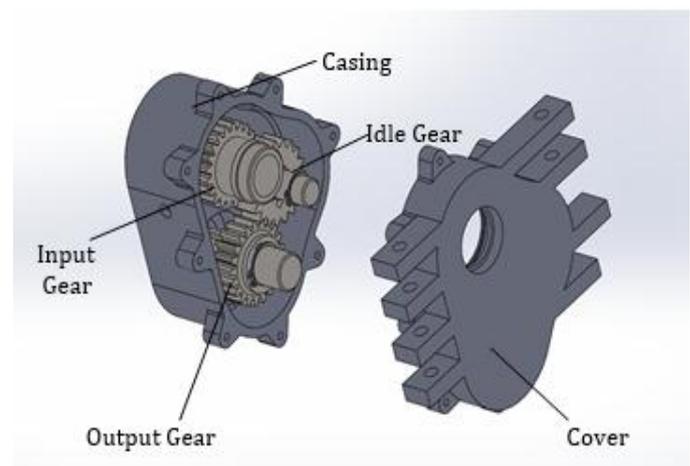


Fig-1: 3D model of gear train with casing and cover

1.1. Literature Review

Previous work done on this topic was finding the contact stress, bending stress for various torque load. Bending stress and contact stress on three different gear trains with various angular positions. Also finding the natural frequencies for different gear trains. All gear trains have similar resonance frequency behaviour when plotted against the first eight mode shapes. The resonance frequency of the gear trains increases significantly when subjected to pre-stressed state [1]. Xiaoyin Zhu provides a brief introduction to the Hertzian contact stress theory, five types of the classical solutions for nonadhesive elastic contact are illustrated, and the applications of the Hertz contact stress theory on optomechanical engineering are also addressed [6]. Contact stress analysis between two spur gear teeth was considered in different contact positions, representing a pair of mating gears during

rotation even torque measurement helped in determining torque provided by portal axle compared with the required tractive effort for various driving conditions [3]. A.R. Hasan stated that pitting is a surface fatigue failure resulting from repetitions of high contact stress. They found contact stress, considering contact ratio, approach angle, recess angle, contact and length of contact [2]. The shaft models were modelled using FEA by Jong Boon Ooi and validated through comparisons with the experimental results. In the analysis of the output shaft of the portal axle, the hollow shaft with a rib is proposed and the final element model is built [9].

The aim of this work was to design a portal axle gearbox and assemble it to the wheel end side of the gearbox so as to increase the ground clearance without doing any reductions in torque transmission. Also, finding the various stresses acting on the gear train.

2. PROBLEM STATEMENT

To enhanced off-roading capabilities of an ATV by increasing ground clearance without compromising with performance. Along with more ground clearance suspension of vehicle become more stable by decreasing sprung mass. They maintain suspension geometry and relieve stress from drive train with a benefit of more lift and long-lasting components. It is Gear Driven Performance.

Design, analysis and manufacturing of Portal gear lift for an All-terrain vehicle (ATV) engine of 305cc.

3. MODEL OF GEAR TRAIN

Fig. (2) shows the assembly of gear train which is used in portal gear lift. It is one stage gearbox which has the input gear, one idle and output gear. Gear ratio is considered as 1:1. FEA analysis carried on this gear train. In this analysis, the model of the input gear, output gear, idler gear is only considered to save computing time. This ignores the interaction of the housing fitting and the bearings fitting in shaft.

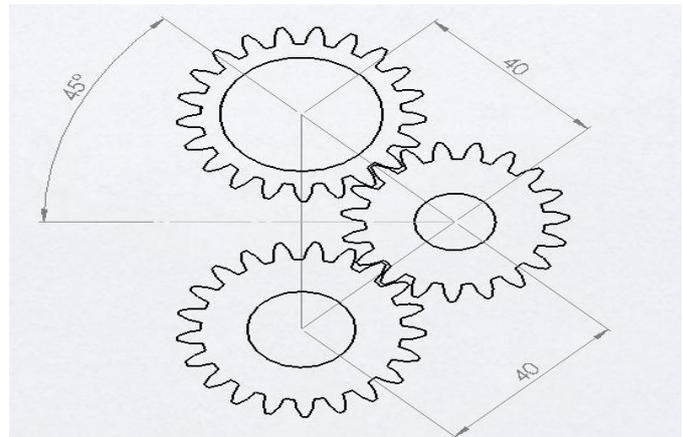


Fig-2: Front view of the gear train with idle gear

The input gear and output gear share the same design parameters which is shown below in the table. The modelled was created in the software name SOLIDWORKS. Design procedure for gear calculation is based on Lewis Form Factor method [10]. The idle gear is positioned between the input gear and output gear to offset the vertical distance and allow only one directional rotation.

Table-1: Gear parameters

Parameters	Input Gear	Output Gear
Module (m)	2mm	2mm
Number of teeth(z)	21	21
Diameter (d)	42 mm	42 mm
Face width (b)	15mm	15mm
Speed (N)	600rpm	600rpm
Addendum (mm)	1.0 *m	1.0 *m
Dedendum (mm)	1.25 *m	1.25 *m
Material	Structural Steel	Structural Steel
Modulus of Elasticity (E)	190 GPa	190 GPa
Poisson's Ratio	0.3	0.3

Design of shaft was carried out using a MATLAB programming. As the diameters of shaft for each gear is different so it becomes very easy on MATLAB to calculate diameters [8]. Material selected for shaft is same as of that gear. Shafts were designed according to principal shear stress theory.

Table-2: Diameter of shaft

Parameter	Input Gear Shaft	Idle Gear Shaft	Output Gear Shaft
Diameter (D)	30 mm	15 mm	20 mm

4. STATIC STRESS ANALYSIS OF GEAR TRAIN

Static stress analysis using FEM was performed. Focusing mainly on the gear tooth bending stress and contact stress caused by two contacting gears. Available constraints were given to the model like material properties and load acting on it. Solution of the problem gave the stresses both contact and bending on gear tooth and suggest if any modification was required or not.

4.1. Bending Strength of Spur Gear Tooth

The analysis of bending stresses in gear tooth was done by Wilfred Lewis in 1982. Today also, the Lewis equation is considered as the basic equation in the design of gears. In the Lewis analysis, the gear tooth is treated as a cantilever beam as shown in fig (3) and fig (4).

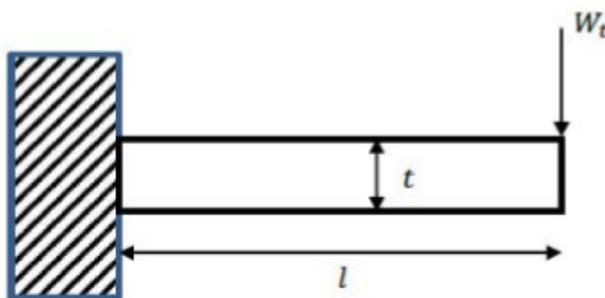


Fig-3: Tooth acts as a cantilever beam [1]

Radial force component induces a direct compressive stress of relatively small amount and hence its effect is neglected.

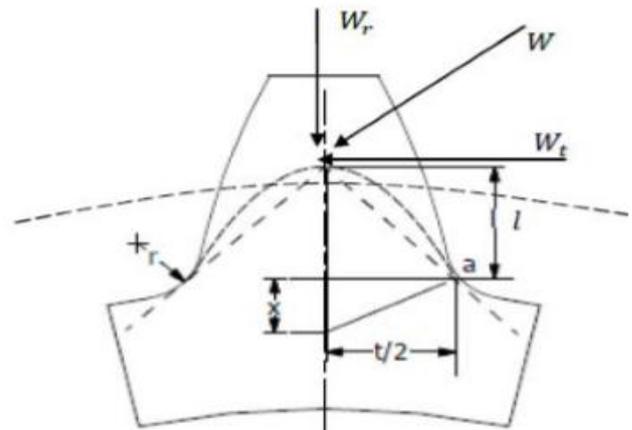


Fig-4: tooth section considered for calculation of bending stress [1].

Bending stress can be calculated by using the following equation

$$\sigma_b = \frac{W_t}{b * m * Y}$$

Where, W_t = Tangential Force (N), b = Face Width of gear, m = Module, Y = Lewis Form factor

To analyze the worst load condition, it is calculated by both theoretically and using FEM analysis. It is done for both input gear and idle gear as input and output have same design parameters whereas idle gear has change in the number of teeth. Lewis form factor can be calculated as.

$$Y = \frac{t^2}{6 * l * m}$$

The Lewis equation is based on the following assumptions:

- The effect of radial component (W_r) of force which produces compressive stress is neglected.
- The tangential component (W_t) of force is uniformly distributed over the face width of gear.
- The effect of stress concentration is neglected.

4.2. FEM Analysis of Gear Tooth

A three-dimensional (3D) model of gear tooth was created using a SOLIDWORKS as a software. Only the model of input gear tooth was created. Mesh refining

given is fine with a size of 0.60mm to get accurate results. Fig (5) shows the mesh model for gear tooth.

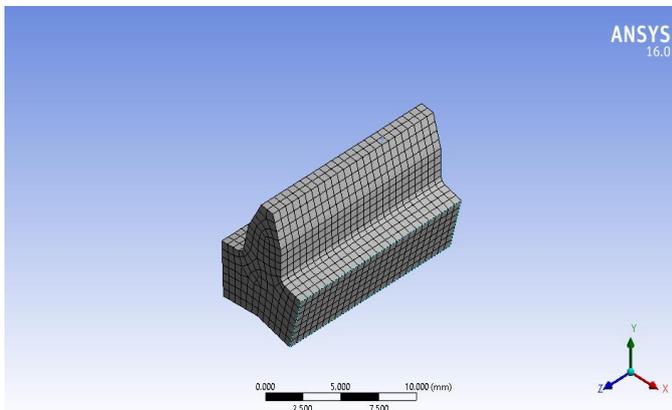


Fig-5: Mesh Model of Gear Tooth

For input gear tooth, tangential force acts on one side of the gear tooth to get critical bending stress. It is assumed that it acts as uniformly over the surface. Boundary condition given as Fixed support at the bottom of the tooth so that it can bend as shown in fig (6). The fig (7) shows that 645.64 MPa is maximum bending stress occurring at one side of the gear tooth. FE simulations predict that the root fillet is more prone to compressive stress rather than tensional stress.

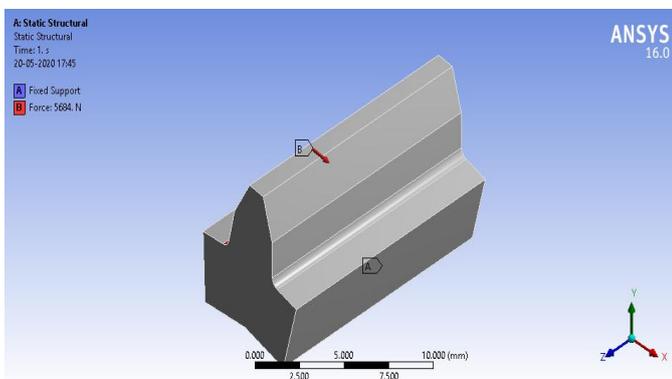


Fig-6: Boundary condition and force acting on gear tooth

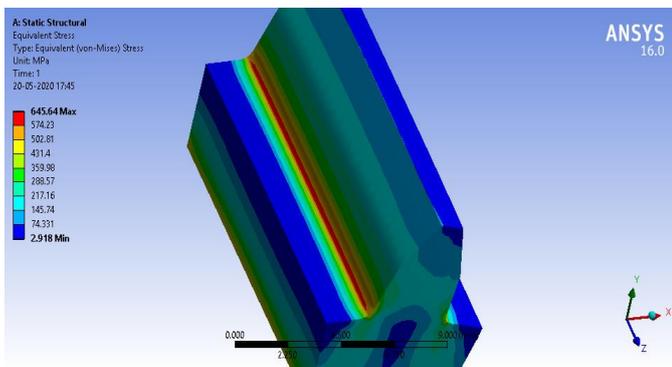


Fig-7: Simulation result with maximum bending stress

3D gear model was analysed using FEA and found the bending stress. The gear tooth bending stress was calculated using both methods with respect to the increased tangential force. The FEM stress results are slightly higher than the one calculated from the results calculated using the Lewis formula. This is because FEM takes into account the radial load component of the resultant force exerted from the tangential force, which causes higher stress results [1].

4.3. Results between theoretical values and calculated FEA values of bending stress

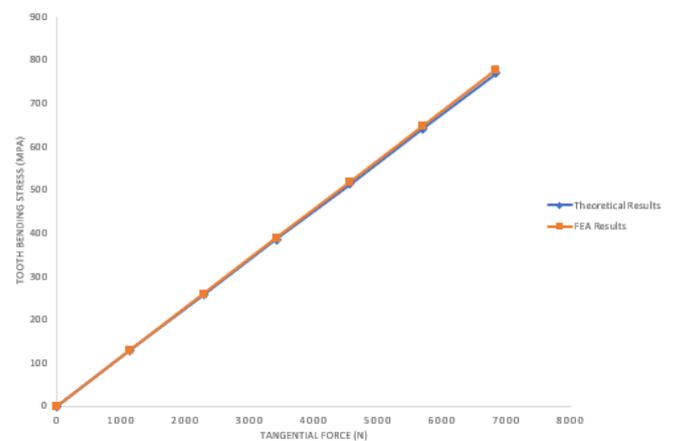


Fig-8: Graph against calculated values and FEA results

Then both results were compared and plotted in graph as shown in fig (8). It shows the linear increase for the both results and the difference between theoretical results and FEA results is very small. The average percentage difference between two values is 1.5% which is totally acceptable. So, it validates that calculated values and FEA results were almost same.

4.4. Contact stress using Hertz equation

In addition to considering the critical bending stress in gears, analysis of gear tooth contact stress is equally important because excessive contact stress may cause failure such as pitting, scoring, and scuffing of surfaces [2]. The transfer of power takes place at the contact of the gear. So, it is necessary to find the stress acting on that contact point. Theoretically, the contact stress can calculate using Hertz equation. The theory provides mathematical expressions of stresses and deformations of curved bodies in contact

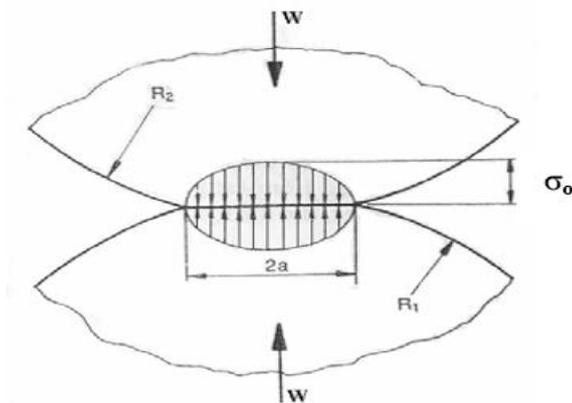


Fig-9: Hertzian model of Cylinders which are in contact [2]

The band of that contact phase can be calculated using following equation.

$$a = 2 \sqrt{\frac{W(1 - \nu_1^2) + \frac{1 - \nu_2^2}{E_2}}{b\pi \left(\frac{1}{R_1} + \frac{1}{R_2}\right) E_1}}$$

The Hertz theory assumes an elliptic stress distribution, the maximum stress is in the middle and equals.

$$\sigma_0 = \sqrt{\frac{W \left(\frac{1}{R_1} + \frac{1}{R_2}\right)}{b\pi \left[\frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2}\right]}}$$

Where, W is the normal load, E_1 and E_2 are the modulus of elasticity of the pinion and gear respectively. ν_1 and ν_2 are Poisson's ratio of pinion and gear respectively. b is the face width of pinion. R_1 and R_2 are the respective radii of the involute curve at the contact point as shown in fig (10).

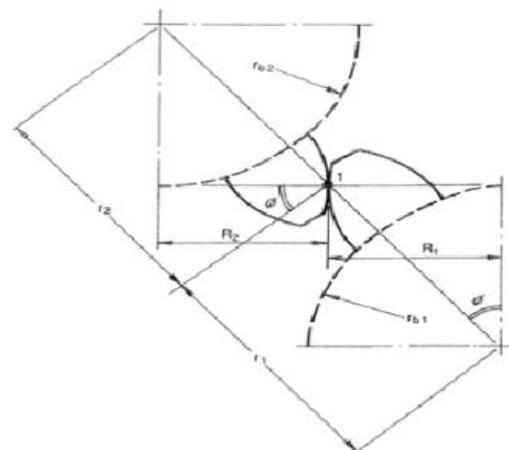


Fig-10: Two involute teeth in contact [2]

However, the pitch radius of the involute of pinion and gear can be denoted as r_{b1} and r_{b2} respectively. It can be related to gear involute radii as $R_1 = r_{b1} \sin \phi$ and $R_2 = r_{b2} \sin \phi$. Hence the contact stress is given by.

$$\sigma_0 = \sqrt{\frac{W \left(1 + \frac{r_{b1}}{r_{b2}}\right)}{r_{b1} b \pi \left[\frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2}\right] \sin \phi}}$$

In the Hertz contact stress equation, a few assumptions are made, such as pure bending of short beam, elliptic distribution of stresses at tooth contact, and friction between the gear contacting surfaces is not accounted in the stress equation. A question therefore arises concerning their accuracy [1]. Some account must be taken for the shape and size of the bodies themselves and the way in which they are supported. In most practical circumstances such calculations are difficult to perform, which have resulted in a variety of approximate formulae for calculating the elastic compression of bodies in line contact such as gear teeth and roller bearings in line contact [2].

4.5. Gear tooth contact analysis using FEA

In the analysis section design was checked which confirmed the design is safe. The 3D gear train with input gear and idle gear is imported into the ANSYS Workbench. As the idle gear is mid part of the gear train as shown in fig (2) so the contact between the input and idle gear is angular. In contact stress analysis, a hexahedron element was selected for the construction of the FE mesh. For higher degree of accuracy, mesh refinement selected as fine refinement

with size of 0.6mm to obtain more accurate contact stress.

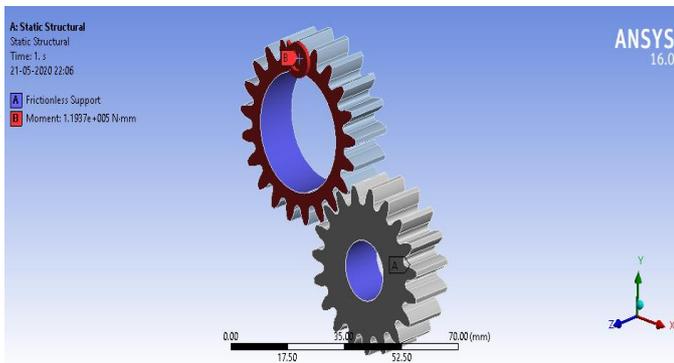


Fig-11: Boundary condition for evaluating contact stress

Boundary condition given as no separation contact between gears. The input gear surface defined as a 'Contact' and the contact surface of the idle gear defined as a 'Target'. Only rotational degree of freedom was allowed. Frictionless support is given for the smooth rotation of gears. Then moment of 119.62 Nm applied to one gear so as to obtain the results as shown in fig (11).

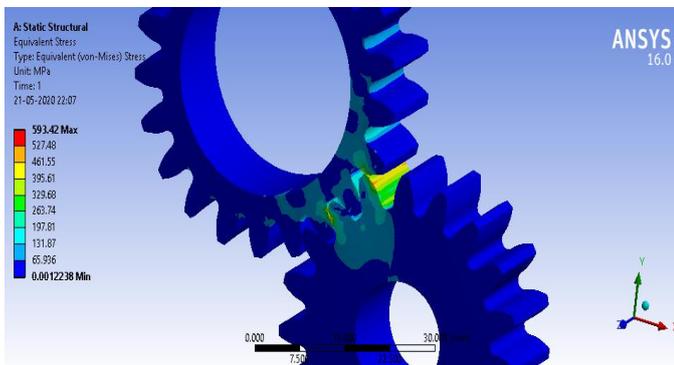


Fig-12: Stress distribution between contact surface of the gears.

Fig (12) shows the stress distribution of the two-contacting surface of the gear when subjected to moment of 119.366 (Nm) torque. The maximum stress of 593.42 MPa occurs. As the contact stress between two gears are not precise because the contacting nodes are not arranged in a similar pattern. The nodes built from the element of both gear surface must coincide to form precise contact stress.

4.6. Comparison between theoretically calculated contact stress and FEM results

The 3D gear train model was analysed using ANSYS and found out the contact stress. Theoretically calculated contact stress was found out using Hertzian contact stress equation. Then both the results were compared with the help of increase in torque load as shown in fig (13).

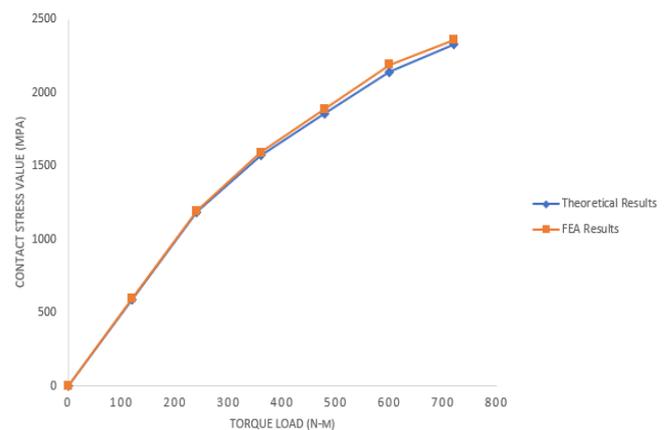


Fig-13: Comparison of Hertzian Contact stress and FEA results.

As seen from fig (13), both results vary exponentially when plotted against the increasing torque load. At the of the graph, there is a slightly more difference between both the results. This because the Hertz equation does not consider the tangential force, which contributes to frictional force on the gear tooth surface [1]. The average percentage difference between both results is 2.3% which is acceptable. It validates the calculated contact stress and FEA results.

5. CONCLUSION

Static structural analysis was done on the gear train using FEA analysis on ANSYS. The bending stress was calculated on the input gear tooth while theoretically calculated using Lewis equation. Contact stress was analysed between the input and idle gear through FEA and analytically using Hertz contact stress equation. Both the results were validated by comparing FEA stress results and stress results calculated from Lewis equation and Hertz equation. Both results show average percentage difference as 1.5% and 2.3% respectively. In the bending stress analysis, bending stress acting on the input gear tooth is more as compared to other two gears.

Main aim of this work is to increase the ground clearance of an All-terrain vehicle. (ATV) without compromising with constant-velocity joint (CV) angle, track width, torque transmission. Fig (14) and Fig (15) shows the line diagram of an ATV before assembly of portal gear lift and after assembly of the portal gear lift respectively. Before assembly of this portal gear lift, the ground clearance was 13.05 inch as shown in Fig (14).

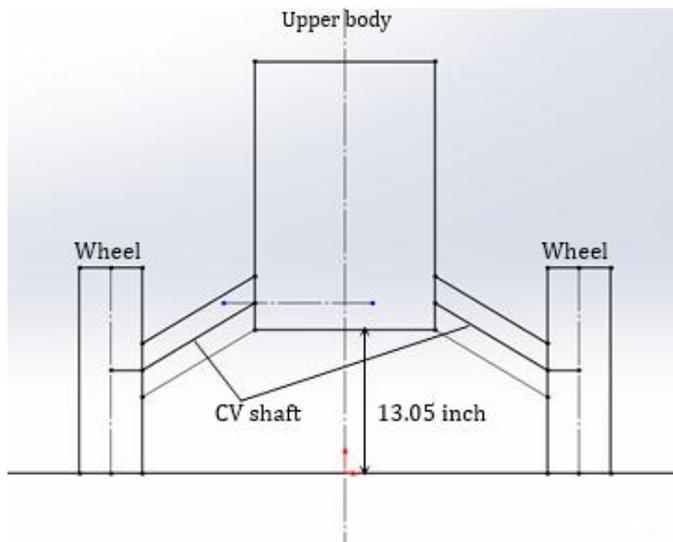


Fig-14: Line diagram of an ATV before portal gear lift

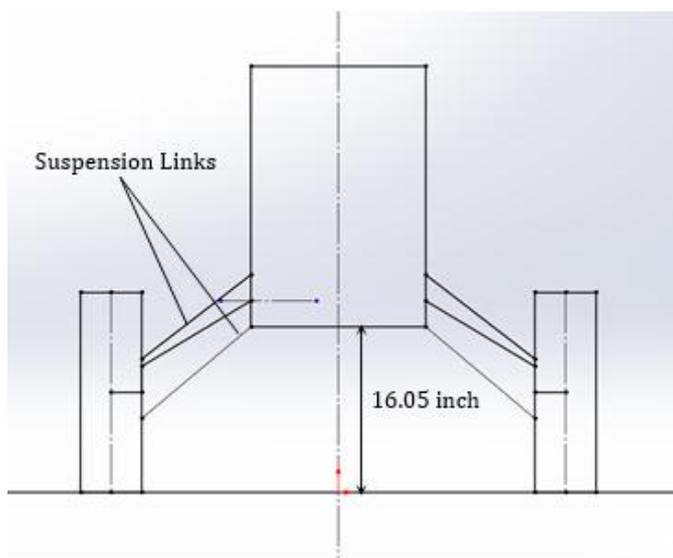


Fig-15: Line diagram of an ATV after portal gear lift

After assembly of this portal gear lift, ground clearance increases from 13.05 inch to 16.05 inch. Increase of 3 inch in an ATV was really a great achievement. Without compromising with the compactness of vehicle, ground clearance was increased and this is why, use of portal

gear lifts are mainly considered to increase the ground clearance of an ATV.

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