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Rigid Body Dynamic Simulation Of Steering Mechanism

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Abstract

Project work present rigid multi body dynamic analysis approach in design. The application of this methodology simplify design process and give correct result. For the case study here work of design done on Ackerman steering mechanism for tipper. In this first according to Ackerman conditions basic geometry is design and then optimize it for static loading, modal analysis and then for dynamic forces generated on steering linkages while turning using Rigid Dynamics tool in Ansys. The results obtained from rigid body dynamic analysis are used for testing individual components and if necessory, corrective actions are taken in design in order to sustain that load. Results shows rigid dynamics approach for design reduces time for optimization, simulation and provide the chance to take most corrective action. Rigid dynamics approach is used in modern design techniques for various domains.
Keywords - Ackerman Steering Condition, Ball Joint Forces, Rigid Dynamics

1 Introduction

Rigid bodies have wide ranging applications across fields including molecular dynamics, robotics, machine assembly, human motion, and computer graphics for video games and feature films. In many applications, the focus is on kinematics or inverse kinematics, and dynamic behavior is of secondary or limited interest with only a few instances of predictable contact and collision, e.g. consider manipulating a robotic arm in a controlled environment. Multibody simulation consists of analyzing the dynamic behavior of a system of interconnected bodies comprised of flexible and/or rigid components.The bodies may be constrained with respect to each other via a kinematically admissible set of constraints modeled as joints.These systems can represent an automobile , an aircraft as an assemblage of rigid and flexible parts, a robot with manipulator arms, and so on. In all such cases, the components may undergo large rotation,large displacements.

2 Methodology

To understand the rigid body dynamic simulation, Ackerman steering mechanism is taken as a case study. For this TATA Tipper (Model-SA1212) is considered. Firstly, steering mechanism is designed considering only wheel track and wheel base by fulfilling Ackerman steering condition. It is then tested for static loading condition. Modal analysis is also done to ensure it does not create resonance while moving on roads. The force acting on the ball joint of tie rod and tie rod arm is calculated from analytical formulae with the help of the specifications of the vehicle. Rigid body dynamic simulation of steering mechanism is done in Rigid Dynamics tool in Ansys. The results obtained through FEA are validated using above analytical equations.

3 Design of Steering Mechanism

To design Steering mechanism, it is first necessary to know the wheel base & wheel track terminology. Wheel track is the distance between the centerline of two roadwheels on the same axle, each on the other side of the vehicle. Wheel base is the distance between the centers of the front and rear wheels. In case of tipper, wheel base is 3700mm and wheel track is 2050mm.

3.1 Satisfying Ackerman Steering Condition

The intention of Ackermann geometry is to avoid the need for tyres to slip sideways when following the path around a curve. The geometrical solution to this is for all wheels to have their axles arranged as radii of a circle with a common center point. As the rear wheels are fixed, this center point must be on a line extended from the rear axle. Intersecting the axes of the front wheels on this line as well requires that the inside front wheel is turned, when steering, through a greater angle than the outside wheel.

Consider a front-wheel-steering vehicle that is turning to the left. When the vehicle is moving very slowly, there is a kinematic condition between the inner and outer wheels that allows them to turn slip-free. The condition is called the Ackerman condition and is expressed by

$$\cot \partial_o - \cot \partial_i = \frac{w}{l}$$

Where, ∂_i is the steer angle of the inner wheel, and ∂_o is the steer angle of the outer wheel. The inner and outer wheels are defined based on the turning center O.

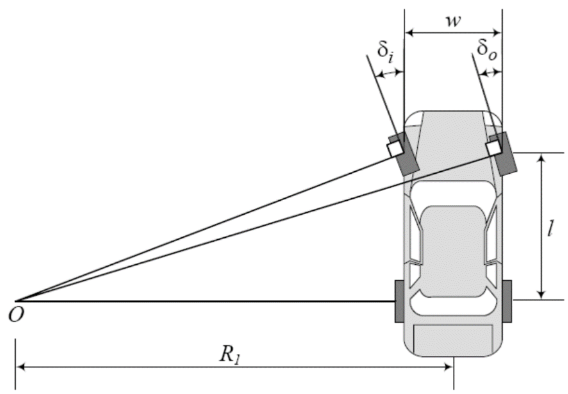

Fig 1 : A front-wheel-steering vehicle and the Ackerman condition.

Ackerman Steering Mechanism is a four bar mechanism consisting of two symmetric inclined links and two parallel links. Out of these four bars, one link is fixed with dimension 'w' i.e. the distance between the steer axes of the steerable wheels. Best practile way to design two incined symmetric links is to draw them from the extrem ends of fixed link till the center of rear wheel track, as shown in figure.

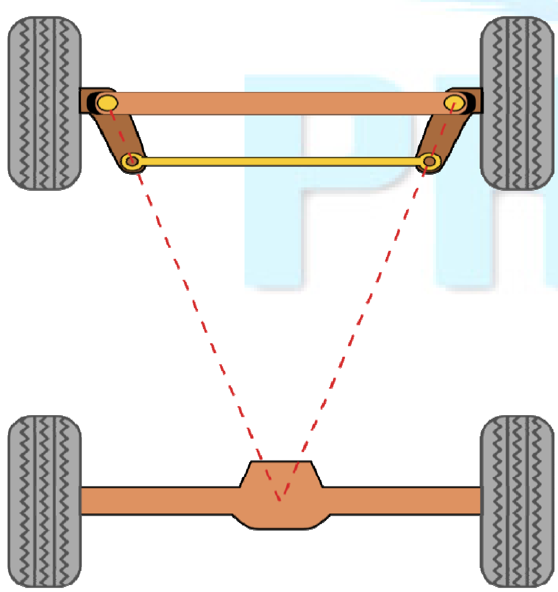

Fig 2 : Approach for Ackerman Steering Mechanism

Now the forth link is to be constracted accros these two links such that it satisfies the basic equation of Ackerman Steering condition. This condition can be achieved by making three lines (two lines perpendicular to two front tyres in inclined position and a line joining two rear wheels) to intercept at a single point. These four links of Ackerman steering mechanism are front axle, two tie rods connected to front axle at extream ends seperately and a tie rod connecting these two tie rod arms.By satisfying Ackerman Steering condition, we get the dimentions of other links (i.e. a tie rod & two tie rod arms) Length of tie rod = 2050 Lenght of tie rod arm = 1516.

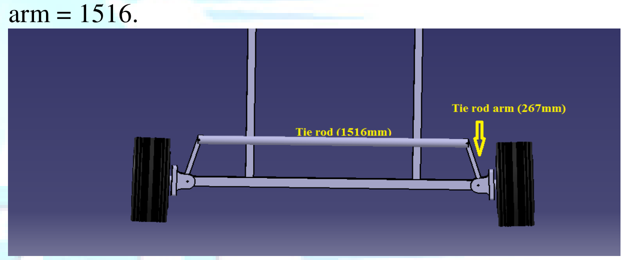

Fig 3 : Specification of steering mechanism

3.2 Modelling of Tipper

A tipper can be modelled by taking referance to above designed Ackerman Steering Mechanism.

Specifications of Tipper (Model-SA1212)
Gross Vehicle Weight (GVW) = 12500kg
Max Front axle weight (FAW) = 5890kg
Wheel base = 3700mm
Wheel track = 2050mm

1) Tie rod
External Diameter of Tie rod = 60mm
Internal Diameter of Tie rod = 48mm
Tie rod length = 1516mm
Yield strength of material (AISI-1019M) = 80 $\frac{kg}{mm^2}$ = 780 $\frac{N}{mm^2}$

2) Tie rod arm
Length of tie rod arm = 267mm
Cross section = 105mm × 30mm
Yield strength of material (40Cr4(TYPE-B)) = 290 MPa

3) Chassis
Type of cross sction = C channel
Cross section = 150mm × 70mm (6mm thick)

Yield strength of material (FE410WASS4017) = 400MPa

3.3 Static Analysis

Out of the total weight of vehicle, fixed load of vehicle (mainly engine) is acted on front axle while the pay load (variable load like material, passengers) is acted on rear axle. Static analysis is perform on basic steering frame. Average load that is acted on front axle at static condition is 5890kg (57.78kN). Exactly this load applied on frame.

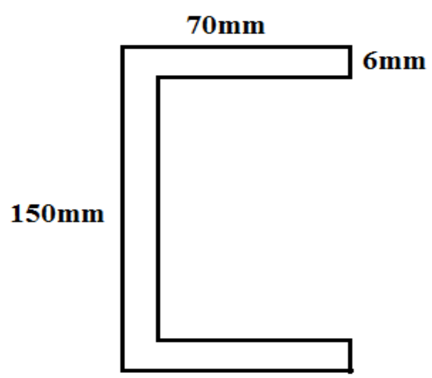

Fig 4 : Cross sectional area of chassis

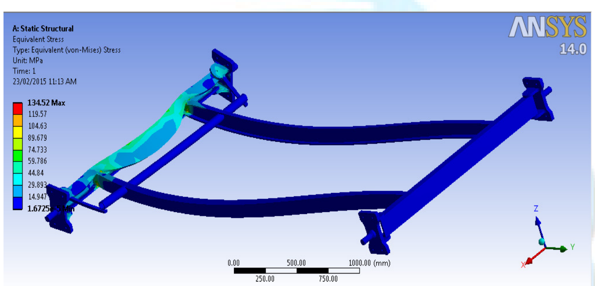

Fig 5 : Static analysis of chassis

3.4 Modal analysis

Modal analysis is the analysis of structure for natural frequencies. This frequencies decided the vibration amplitude of structure. If this frequencies match with current situation frequencies then structure body vibrates with very large amplitude which damage body. According to Indian authority for road data vibration generated frequency generated due to road shocks is between 0.3Hz to 20Hz. In India due to such vibrations automobile chassis static deflection take place over a time.

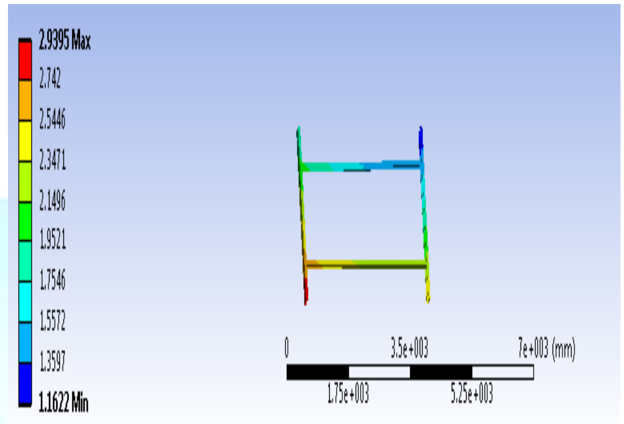

Fig 5 : Deflection after Modal analysis

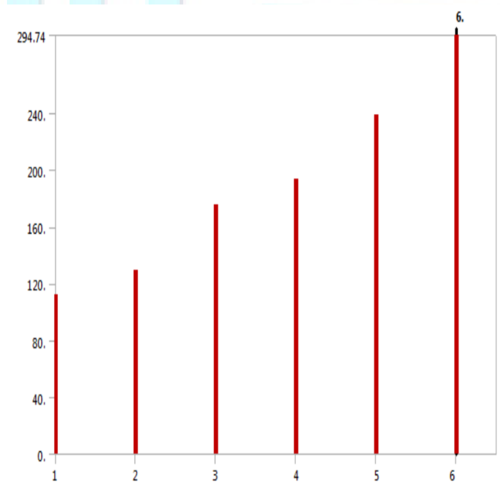

Fig 5 :

Natural frequencies of steering mechanism

4 Steering System

A steering system begins with the steering wheel or steering handle. The driver's steering input is transmitted by a shaft through a gear reduction system, usually rack-and-pinion or recirculating ball bearings. The steering gear output goes to steerable wheels to generate motion through a steering mechanism. The lever, which transmits the steering force from the steering gear to the steering linkage, is called Pitman arm. The direction of each wheel is controlled by one steering arm. The steering arm is attached to the steerable wheel hub by a keyway, locking taper, and a hub. In some vehicles, it is an integral part of a one-piece hub and steering knuckle.

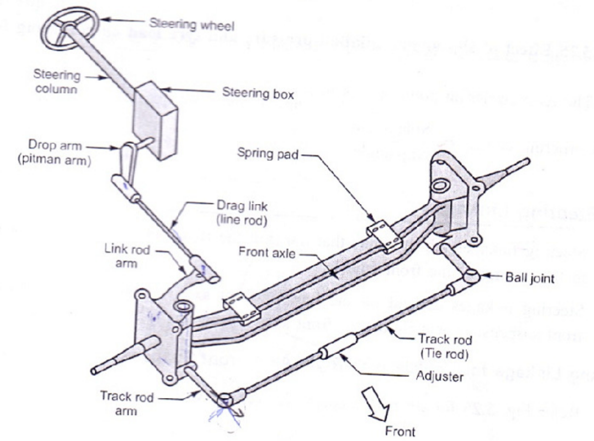

Fig 6: Dynamic forces acting on Steering linkages

For turning a vehicle, we apply steering effort. The maximum torque that can be transferred further by steering is totally dependent on gear box (also known as steering box) which is connected to steering column. For a tipper under consideration, the torque on output shaft of gear box is 453372.69 Kgmm i.e. 4447.5 kNmm.

This output shaft is connected to the drop arm (pitman arm) of length 268mm whose other end is attached to the drag link. Maximum force that can be transitted by the drop arm is found out as,

Force = moment/length = $\frac{4447.5}{268} = 16.59\ kN$. This drag link force is nothing but the steering arm force.

Drag link is connected to the connected to the stub axle. So, the same maximum force is transmitted by the drag link on the stub axle. Stub axle swives around king pin which is 250mm away from it. Generated torque about king pin is

$$16.59 \times 274.2 = 4550.47 kN.mm$$

Tie rod arm is connected to the king pin on one side and tie rod on other side through ball joints. Maximum force on tie rod which is at ball joint can be found out by dividing torque by the length of tie rod arm.

Maximum force on ball joint is $\frac{4550.47}{267} = 17.04kN$

5 Rigid Body Dynamics

Rigid body dynamic simulation is done using the Rigid Dynamics tool in Ansys. The input to the four bar mechanism of steering linkages is the moment of 4.55 kN.m acting on king pin and tie rod arm through drag link. As the tipper under consideration is right hand driven vehicle, this input moment is given to the right side. The rigid body dynamic simulation is performed on these linkages and dynamic forces acting on ball joint is found out through analysis.

5.1 Steps for Rigid Body Dynamic Analysis

As we open Ansys Workbench, at the left side we can see number of tools like “Explicite Dynamics”, “Harmonic Response”, “Linear Buckling” etc each having its own specific function. From this list we are choosing “Rigid Dynamics” option.

Selecting Geometry -: Input to the Rigid Dynamic Analysis is the geometry of the model. There are two ways to provide geometry for the analysis. Models which are in the “CAD”, “igs” etc format drawn seperately in any CAD software like AUTOCAD, CATIA, PRO-E etc can be directly imported. While importing model, care should be taken that model is made by assembling different parts and not just a single rigid body that we use for static anlysis. Other option to provide model for analysis is creating it in ‘Geometry’ tool of Rigid Dynamics. Geometry should be created by drawing different parts of required model using ‘add frozen’ option so that it should be treated as assembly of parts by the software.

Defining Contacts of Model -: Once we provide geometry input, next step is defining contacts of model. Before provideing contacts manually, it is necessory to delete contacts autometically created by software. First of all, we need to specify a part of geometry as a Ground Body i.e. reference body with respect to which we are going to carry the analysis. Now all other static parts (having no movent in simulation) are defined as fixed body with respect to ground body. Proper degree of freedom should be allocated and

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remaining all other should be restricted so that we get our desired motion from the model. After defining all contacts in such fasion, input part should be configured to check wheather our assembly behaves (moves) in desired manner or not.

Applying force/ displacement to input part-: Force or displacement is provided to input link as per the output requirement. In solution we can ask for total deformation of the assembly. To get the dynamic forces at every joint, it is necessory to drag that specific joint in solution section.

Results –: Total deformation of the assembly is first output of the analysis. Dynamic forces (force variation with respect to time) is the important result of rigid dynamics. Total cycle time of moving assembly is divided in number of small intervals. At any specific time, we get forces in all direction of that perticular joint. Graph can also be viewd showing variation of forces with respect to time in all the three diections.

5.2 RBD of Steering Mechanism

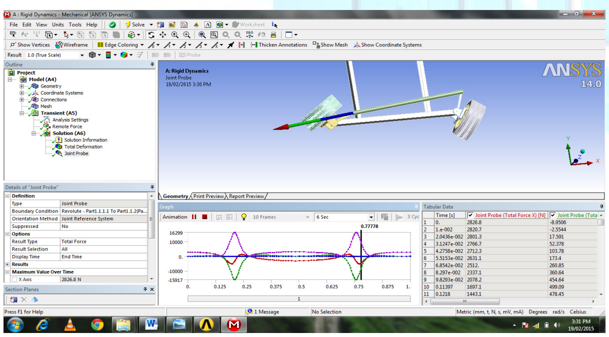
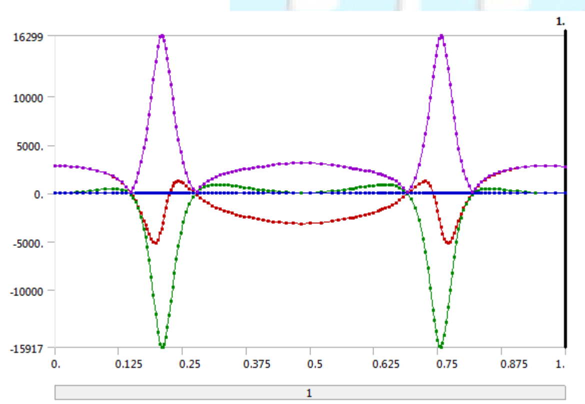
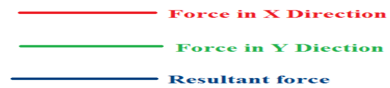


Fig 7 : Result of RBD of Steering Mechanism





Graph : Forces acting on ball joint

The maximum forces acting on ball joint is 16.23kN using rigid dynamic tool in Ansys whereas by analytical calculations its value comes to be 17.04kN.

6 Static Analysis of Components

Maximum forces acting on ball joint in the steering mechanism obtained from above rigid body dynamic analysis are applied to seperate components. Tie rod and tie rod arm has two joints at two extream locations and at each joint two maximum forces are acting in mutually perpendicular directions. In order to servive in running condition, links have to withstand under these forces. So, static analysis of each link is performed.

The material used for tie rod is AISI-1019M and that of tie rod arm is 40Cr4(TYPE-B) having yield strengths of 780MPa and 290MPa respectivelly. While performimg static analysis, precausion should be taken that generated stresses at any part of any component should be less the yield strength of that respective material.

In all, maximum two types of forces are acting on tie rod and tie rod arms (i.e. normal and bending are acting).

1) For Tie rod-:
15917N in bending direction -
5149N in normal direction.

2) For Tie rod arms-
- 5149N in bending direction
- 15917N in normal direction.

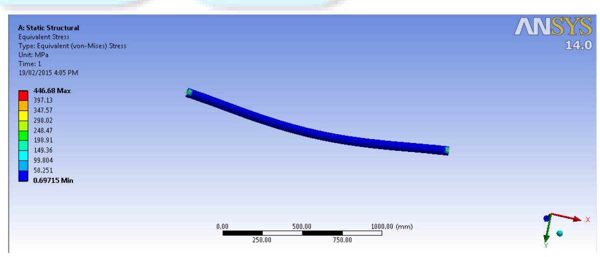


Fig 8 : Stress due to bending load on tie rod

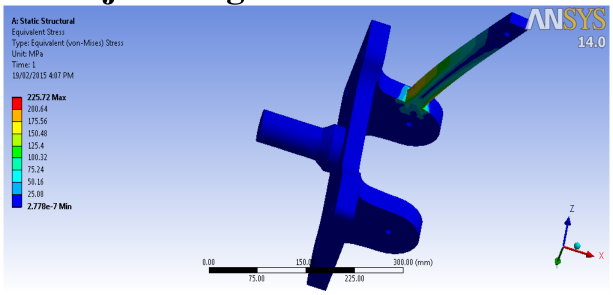


Fig 9 : Stress due to bending load on tie rod arm

7 Conclusion

Generally mechanisms are tested under static conditions and more focus is not given for dynamic forces. For any mechanism, testing under dynamic forces has to be done in order to approach towards more safe design.'Rigid Dynamic' tool in Ansys workbench helps us to find dynamic forces at all joints in all the three directions. Unlike static analysis, in rigid body dynamic simulation, variation of forces with respect to time are obtained which will be very useful for further analysis. By considering these forces, seperate analysis should be done to check wheather components servive under these dynamic forces or not.

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