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Optimization, Design and Analysis of Push Rod Actuated Double Wishbone Suspension System

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Abstract—*Suspension system is one of the most imperative sub-systems of an automobile. Its elementary function is to quarantine the driver from the road shocks and bumps. The Subordinate function includes load transfer lateral stability and provides adequate wheel travel fortifying ergonomics and driving comforts. The study describes optimization, design and analysis of a push rod actuated double wishbone suspension system for a formula styled race car. The aim is to compete in FSI (Formula Student India) competition. The suspension is designed for giving adequate stiffness, motion ratio, wheel rate, roll rate, camber variation, anti-dive geometry and various other parameters. Various stress and displacement analysis on upright, bell-crank and wheel hub and buckling load analysis on push rod were carried out for accomplishment of the optimized design.*
The geometry of the suspension has been modeled in Unigraphics NX 8.5 and MSC ADAMS. The finite element analysis of suspension components are done in SOLIDWORKS 2014

Keywords*: Push rod, double wishbone, roll, wheel rate, roll gradient, anti-dive geometry, factor of safety.*

1. INTRODUCTION

The intent of a suspension system is to quarantine the body and the respective parts from the variability of the road surface. Exemplary the vehicle should ride in level and without any vertical motion. And another supreme feature of suspension is that keeps the tires forced towards the ground during the ride and if there is no suspension system, tires tend to lift off the ground every time when they will pass over a bump and as a result the shock occurs as the tires leave the ground and hit back again, that will be felt by the driver. There are few aspects that were austerely abided by during designing of this kind of suspension system. The fundamental was to keep the suspension parameters such that they don't change from changing of wheel travel as much as possible. The ground clearance has to be optimum to clear any kind of hindrance on roads. The relocation of the Instantaneous Centers has to be kept low. The Roll Centre heights in front and rear were kept so that the Roll Axis slope is inclined forward this will help quicker load transfer. The Side View Swing Arm Geometry was considered as it provides the maximum load transfer to the shocks.

2. PUSH ROD ACTUATED DOUBLE WISHBONE SUSPENSION SYSTEM

Pushrods are actually diagonal bars connected to a vehicle body at one end and to the lower A-arm or knuckle at the other end. The springs and dampers are mounted on the chassis of the vehicle and positioned transversely. When a wheel goes over a bump then pushrod pushes the bell crank and this force acts on the bell crank from there on to the shockers and is absorbed by them. Due to double wishbone and push rod actuation control of wheel and shocker movement remains disarticulated from each other resulting in finer response and expeditious handling at variable speeds. Immutable link of the springs and dampers to the chassis shows that the springs need not to be very stiff, this provides an affluent ride while offering meticulous handling.

We opted to have a push rod so that we could adjust the shock travel rate self-governing of the wheel rate. We want shock to move as much as the wheel moves with respect to the ground. The more far a shock moves it becomes easier to control the wheel since the shock gets more time to work. We can adjust shocks in fast jounce, slow jounce and rebound so we opt the fast jounce quite soft and firm slow jounce. This will slow down the body roll as the vehicle gets into a turn so that the chassis isn't affected by quick and jerky movements. The A-arms are made in a way that the upper arm is short and the lower one is long so as the wheel moves up and down, the shorter upper A-arm moves on a much steeper arc than the longer one.

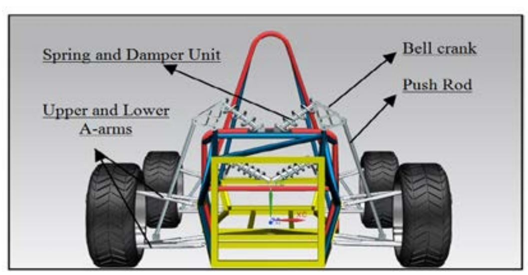

Fig. 1: Front View Depicting Basic Components

This pulls the top of the wheel in faster camber gain than the bottom. With proper negative camber gain the tire stays as flat as possible on the ground as the car travels while cornering.

A Bell crank is a mechanical device which is used to convert translational motion of one object (push rod) into translational motion of another object (shockers) managing at different angles. We have used a bell crank to stimulate the spring and damper unit via a pushrod. The dimensions of the bell crank may be altered to change the relative motion of the shockers with respect to the pushrod i.e. the motion ratio. This is essential as the motion ratio can be changed without changing the properties of the shockers. The dimension of the bell crank gives the motion ratio of the Suspension system which affects the spring stiffness and damping ratio. Fig. 1 above shows the Front View depicting the basic components.

3. SUSPENSION DESIGN CONSIDERATION

Front Suspension is very important and is designed first while designing a vehicle suspension system. The front of the vehicle hit the jerks first and makes the platform of the motion and loading characteristics. Thus it should provide enough wheel travel and damping effects to absorb the bumps and jerks.

Short and Long A-Arm Double A-Arm were used to provide essential camber gain curves in wheel travel and the shorter upper A-arm helps to bring about a camber curve which sustain most of the contact patch while rolling. During rolling, inner wheel droops while the outer wheel faces bump motion which in turn decreases the contact patches of the wheels. It was all attained by the Short-Long A-Arm geometry which helps to sustain the maximum contact patch even in rolling condition. The inner wheel facing droop have a positive camber change but the outer wheel faces bump and have negative camber. This will help in maintaining the wheels closer to the vertical, thus increases the contact patch and surely the traction. Roll Camber coefficient finalizes the angle of wheel with body roll and so determines the traction while rolling. We designed the suspension so as to assist all the packaging parameters and keep a roll camber coefficient at all times.

Based on the assumptions we made the front view swing arm geometry shown in Fig. 2. We made a line drawn from the ground contact patch going through the desired roll centre joining the instant center. Now from the instant centre one line is drawn back joining the location of lower ball joint and one line drawn to the location of upper ball joint. The length of the lower A-arm is as long as possible but is restricted by packaging parameter. The driver's legs have to be accommodated between the lower A-arms so to keep the CG height low. The length of the upper A- arm determines the curvature of the negative camber curve. If upper and lower A-arms are of the same length the camber curve will be a straight vertical line but if the upper A-arm is shorter than the lower A-arm, the curve will be tending towards the negative camber which is acceptable. Hence, shorter the upper A-arm the more concave the camber curve will be. So we decided to design geometry that will be having progressive camber in bump.

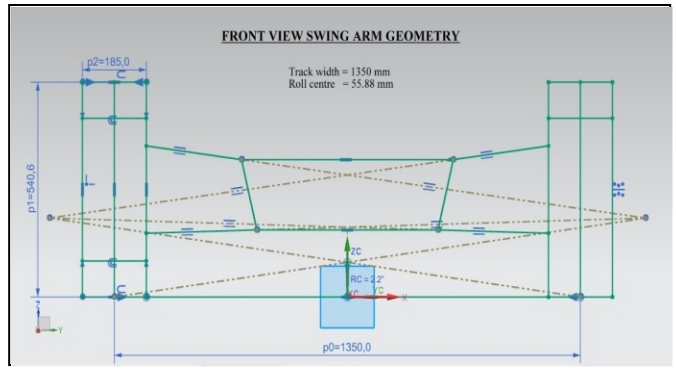

Fig. 2: Front View Swing Arm Geometry

Now the design of the side view swing arm geometry is based on the required anti-features i.e. anti-dive and anti-squat. But for front suspension of a rear wheel drive car the only anti feature is anti dive. This geometry gives us the desired angle of the side view A-arm. The length of the A-arm and Side View Swing Arm Length determines the amount of longitudinal wheel travel while bump & droop. Figure3 shows svsa geometry.

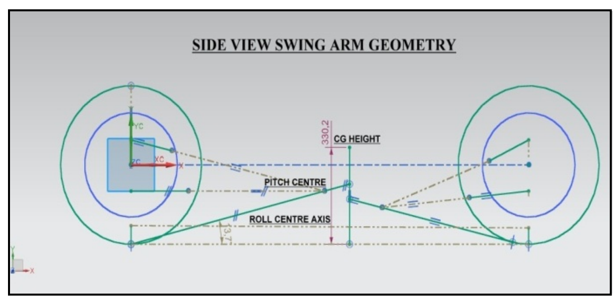

Fig. 3: Side View Swing Arm Geometry

4. DESIGN OPTIMIZATION

Short and Long A-Arm were used as they provide ideal camber curves in wheel travel and the shorter upper A-arm instigates a camber curve maintaining maximum contact patch while rolling. During rolling motion the inner wheel faces droop while the outer wheel faces bump motion which decreases the contact patch in the wheels and this was achieved by the Short-Long A-Arm geometry. The inner wheel which faces droop suffers a positive camber change but the outer wheel faces bump and gains negative camber change. This will help in maintaining the wheels as close to vertical position as possible. Dynamic Analysis of suspension system was carried out in ADAMS and variation of dynamic camber change is shown in Fig. 4.

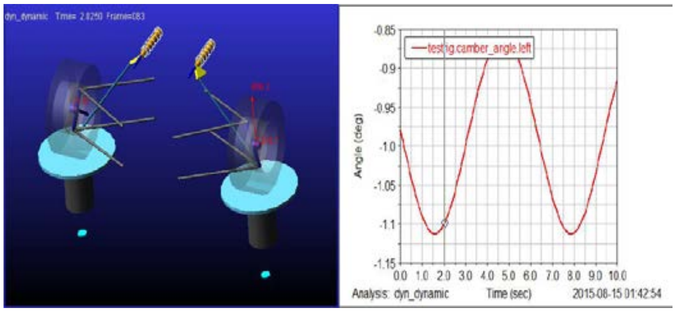

Fig. 4: Dynamic Camber Change

Roll-Camber coefficient of vehicle decides the angle of the wheel with respect to roll of body hence determines the traction while rolling. We designed the suspension so as to keep a roll camber coefficient at all times. The Camber Coefficient graph is plotted in Fig. 5.

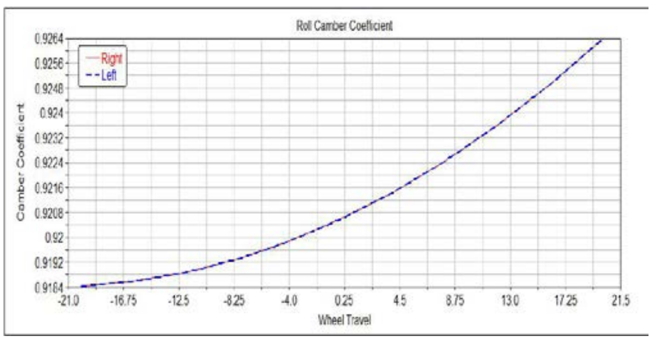

Fig. 5: Roll Camber Coefficient

So as the vehicle tilts in corners the camber angle will decrease. And during this period the bottom touching tread of the tire is quiet away from the center line of the vehicle than the top one. The reason for this is that the complete tread of the tire holds the road equally in corners due to tire sidewall. Camber Gain graph is plotted in Fig. 6**.**

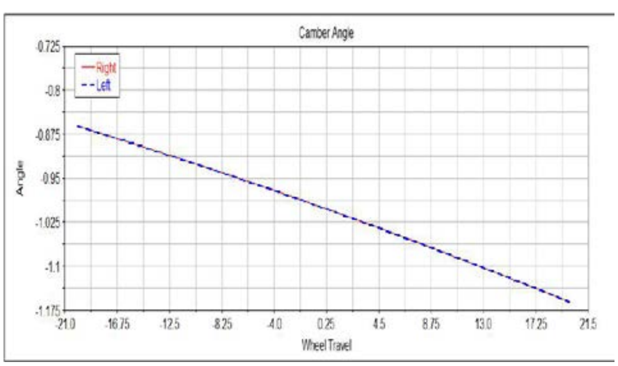

Fig. 6: Camber Variation with Wheel Travel

Since the position of the roll centre is determined by the location of the instant centers. High instant centers will lead to a high roll centre & vice versa. When the vehicle steers, centrifugal force acting on CG transmitted to the roll centre and goes down to the wheels where the lateral forces are acting. So we can see that higher the roll centre height much smaller will be the rolling moment and this rolling moment must be countered by the springs. If the roll centre is located above the ground the lateral force generated by the tire generates a moment about the instant centre, which pushes the wheel down and carries the sprung mass. Fig. 7 shows the Roll center variation with the wheel travel.

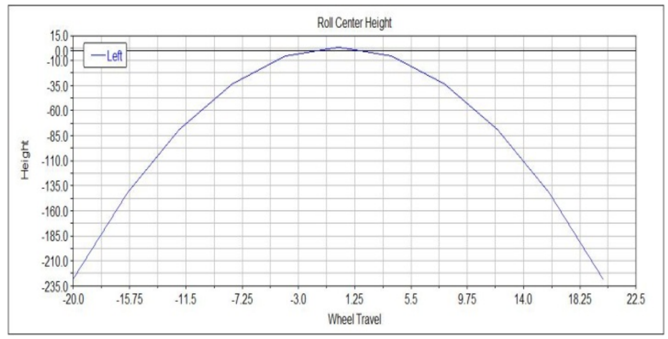

Fig. 7: Roll Center Variation with Wheel Travel

Kingpin angle is the angle viewed in end view between the vertical and the steering axis. Positive kingpin is the angle when the kingpin axis tilts in towards the center of the vehicle whereas negative inclination is just the opposite. Another key parameter is the scrub radius which is the horizontal measurement between the center of the tires contact patch & the kingpin axis. Stated is that positive offset will be defined when the scrub radius is located outboard from the kingpin axis. Variation of Scrub Radius with the wheel travel is shown in Fig. 8.

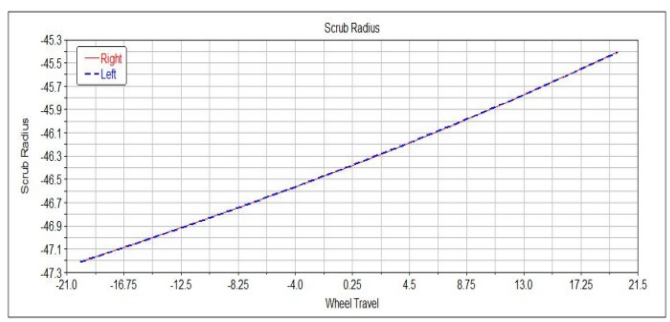

Fig. 8: Scrub Radius Variation with Wheel Travel

Wheel Rate is the actual rate of a spring acting at the tire contact patch. This value is measured in lbs/inch or N/mm, just as spring rate. The wheel rate can be determined by using the formula below.

Wheel rate (WR) = spring rate/ MR^2

Variation of wheel rate with wheel travel is shown in Fig. 9.

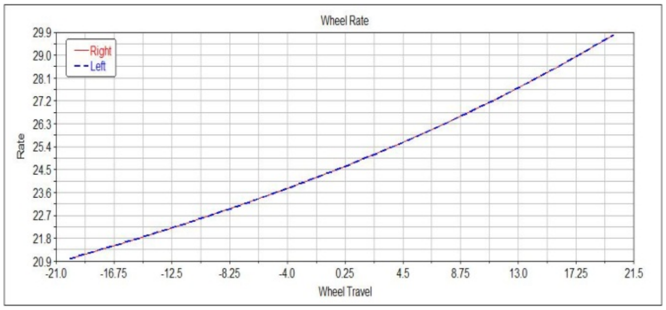

Fig. 9: Wheel Rate Variation with Wheel Travel

Analysis has been done on various suspension components to examine Stress analysis, Displacement analysis, Factor of Safety value on applying the respective forces. Analysis of lower A-arm has been shown in Fig. 10 as the lower A-arm is facing more forces than the upper one on calculating critical load it came 2500N.

Results were in favor and the component was under safe limits.

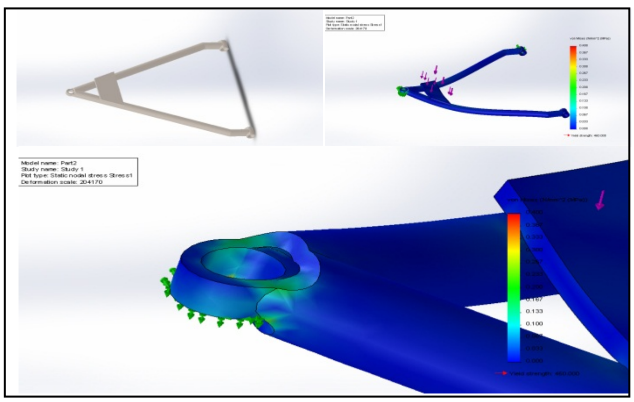

Fig. 10: Stress Analysis- Lower A-arm

The upright design also benefited greatly by using FEA to find and change locations where stresses were either too high or low enough that material could be taken away. After the initial design was complete a series of simulations were performed for final design changes under 2g braking in Fig. 11.

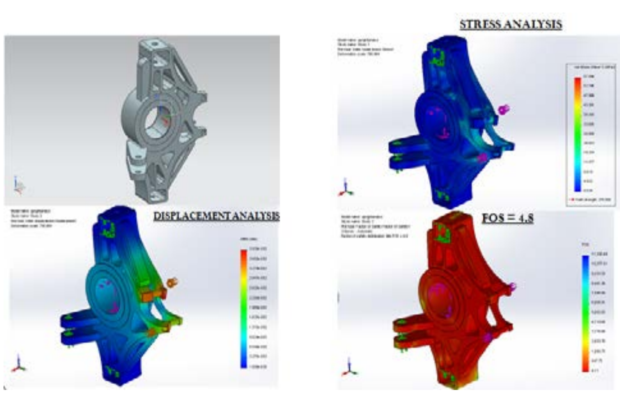

Fig. 11: Upright Analysis

Bell Crank is an important component needs to be analyzed as they are subjected to large amount of stresses. Also for the bell crank, stress analysis is done by using FEM. From the results of this analysis it is observed that results shows maximum failures stress concentration occurring at maximum bending surface. Comparison between numerical, FEM and experimental data it was observed that results obtained are in close agreement with each other shown in Fig. 12.

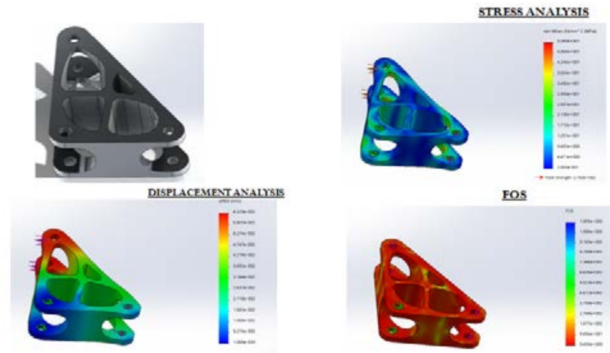

Fig. 12: Bell Crank Analysis

Push Rod has a disadvantage that they may buckle under stresses. So to check, buckling analysis is also carried out under critical load calculated of 1375.7N. Result is shown in Fig. 13.

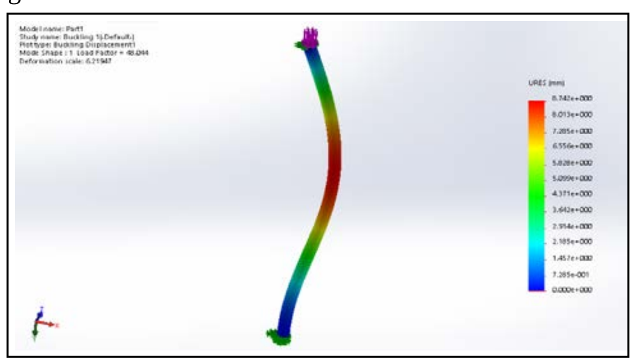

Fig. 13: Push Rod Buckling Analysis

5. FINAL RESULT SHEET

Table 1: Springs and dampers Specifications

Vehicle weight	330 kgf
Motion Ratio	Front- 0.8 Rear- 0.7
Frequency	Front- 3.1 Hz Rear - 2.8 Hz
Spring Stiffness	Front-11.16 N/mm Rear- 11.81 N/mm
Target Wheel Rate	50.8 mm
Wheel Rates	Front- 17.43 N/mm Rear- 24.116 N/mm
Roll Gradient	0.7 deg/g
Damping ratio	0.7
Roll Centre Height	Front- 55.88 mm Rear- 63.5 mm

```
Info for Suspension Analysis

  Analysis Name :  dyn_dynamic
  Analysis Type :  Dynamic Suspension Analysis
  A/Car Version :  Adams 2014
  Built on      :  Aug 26 2014
  Submitted by  :  Mayur Patil
  On hostname   :  Samsung
  Date          :  2015/07/20,05:09:41

***************************  ANALYSIS RESULTS  ***************************

    Successful completion

**************************  ANALYSIS PARAMETERS  *************************

       number of steps :     400
        vertical setup :  wheel_center_height
    type of vertical excitation :  displacement
    left actuator function :  20*sin(time)
    right actuator function :  20*sin(time+PI)
       steering motion :        0.0

*************************  SUSPENSION PARAMETERS  ************************

    total_sprung_mass  :    125.6340
    cg_height          :    330.2000
    wheelbase          :   1600.0000
    drive_ratio        :      0.0000
    brake_ratio        :      0.5500
    wheel_radius       :    270.3000
    tire_stiffness     :    150.0000

*****************************  TESTRIG INFO  *****************************

  Testrig Name  :  __MDI_SUSPENSION_TESTRIG


****************************  ASSEMBLY INFO  *****************************

  Assembly Name  :  Front_assembly
  Assembly Class :  suspension
  File Name      :  <FSI_MD>/assemblies.tbl/Front_assembly.asy


  SUBSYSTEM NAME                     MAJOR ROLE             MINOR ROLE

  fsi_front_susp                     suspension             front

****************************  SUBSYSTEM INFO  ****************************

 Info for subsystem:  testrig

   File Name     :  *cannot find subsystem file*
   Template      :  *cannot find template file*
   Comments      :
      Template   :  Suspension Analysis Test Rig
      Subsystem  :  none
   Major Role    :  analysis
   Minor Role    :  any


   HARDPOINTS:
     hardpoint name              symmetry        x_value    y_value    z_value
     --------------              --------        -------    -------    -------
     global_part_reference       single             0.0        0.0        0.0
     steering_input_rotation     single           200.0      200.0        0.0
     steering_input_slider       single           200.0        0.0        0.0
     steering_input_translation  single           200.0     -200.0        0.0

 Info for subsystem:  fsi_front_susp

   File Name     :  <fsi_MD>/subsystems.tbl/Front_assembly.sub
   Comments      :  *no comments found*
   Major Role    :  suspension
   Minor Role    :  front


   HARDPOINTS:
     hardpoint name              symmetry        x_value    y_value    z_value
     --------------              --------        -------    -------    -------
     global                      single           250.0        0.0     -175.0
     arblink_to_bellcrank        left/right      -385.0     -203.2      472.7
     arb_bushing_mount           left/right       377.0     -127.0      -73.4
     bellcrank_pivot             left/right      -359.6     -254.0      434.6
     bellcrank_pivot_orient      left/right      -359.6     -254.0      333.0
     lca_front                   left/right      -766.0     -219.1      21.85
     lca_outer                   left/right      -512.0     -716.0       27.4
     lca_rear                    left/right      -258.0     -219.1      21.85
     prod_outer                  left/right      -461.2     -689.0        2.8
     prod_to_bellcrank           left/right      -435.8     -254.0      447.3
     shock_to_bellcrank          left/right      -385.0     -203.2      472.7
     shock_to_chassis            left/right      -131.0     -203.2      472.7
     tierod_inner                left/right      -415.0     -304.8       77.4
     tierod_outer                left/right      -415.0     -650.0       77.4
     uca_front                   left/right      -766.0    -200.05     263.15
     uca_outer                   left/right      -512.0     -641.4      283.0
     uca_rear                    left/right      -258.0    -200.05     263.15
     wheel_center                left/right      -512.0     -700.0      155.2
```

Fig. 14: Dynamic Suspension Analysis Report

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