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Design and Analytical Calculations of Double Wishbone for Formula Student Race Car

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Abstract— **The functional objective of the suspension system is to provide relative motion between the sprung and the unsprung mass. This is achieved by a spring to absorb the shocks and some kinematic linkages holding them together with particular degrees of freedom. Further for better handling the kinematics of the system is designed and optimized. This research focuses on the simulation, design, and analytical calculations of the kinematical linkage called A-arm. The double-wishbone suspension geometry was analyzed in Lotus Shark software to procure the most appropriate location of the hardpoints. A-arm was modeled in CATIA V5, Autodesk Fusion 360, and analyzed in ANSYS 16.0. Software results are correlated with the analytical calculations to obtain a feasible solution.**

Keywords—Double wishbone, A-arm, thickness calculation, rod end selection, vehicle dynamics, threaded joints, etc.

I. INTRODUCTION

The FS car's suspension controls the angle, position, and velocity of each wheel to maintain a high value of mechanical grip while transmitting the forces generated by tires to the chassis. A reliable method to determine the forces produced by the road loads in suspension members opted. In this paper, a detailed analytical calculation was explained for deciding the A-arm diameter using a set of calculations. The component is designed to be cost-effective, durable, and lightweight. As the double-wishbone undergoes tension and compression forces, the yield and buckling need to be calculated, followed by the threaded joints calculation for the rod ends as the force being transmitted through these to the chassis.

Fig1. Double wishbone

II. PROBLEM STATEMENT

The double-wishbone suspension system is independent. This design allows the race car engineers to control the wheel's motion throughout suspension travel, controlling the parameters such as camber angle, caster angle, toe, roll center height, and scrub radius.

They are a force transmitting and kinematic part, so dynamic stability is the most crucial factor. Designing by using lightweight material along with it being cost-efficient is the recent discovery many teams are trying to develop. In the era of upgrading trends, many teams are trying to evolve in a world full of research related to advanced materials.

In this fast-changing and technologically developed world, there is much-advanced software in use, but all that software is programmed on the necessary calculations. To correlate those results with calculations and further substantiate those software results, we need analytical calculation. Because sometimes that software is constrained to some limits. The paper explains the method to find the thickness of the a-arm by pen and paper calculation. These calculations require critical skill, and also, they are an essential set for sound stress engineers in the automobile industry.

Some teams want better design irrespective of cost, and some want a mediocre design with the economical product. But many teams are trying to get the intermediate between them, so to improve this, we need our work to be well developed from all angles and aspects. Hence analytical form is considered to be more cogent.

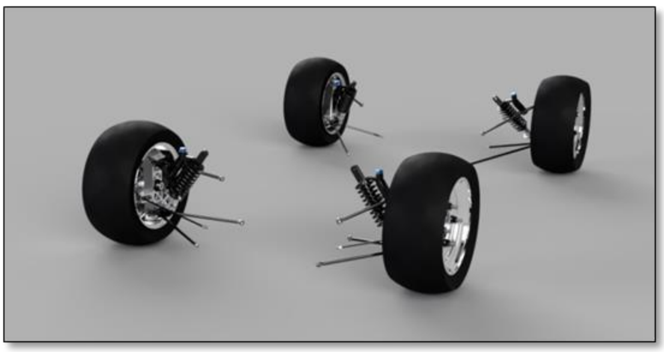

Fig2. Suspension Assembly

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III. OBJECTIVES

1. A lot of forces during cornering, acceleration, and bump are applied directly during dynamic conditions, so it is essential to produce a compact, lightweight, and reliable suspension system to increase vehicle performance.

2. To reduce the unsprung mass.

3. Combining analytical calculations result with FEA as it provides a good starting point for the design.

4. To understand how much an advantage FEA gives during the design process, it's first necessary to understand how analysis and design optimization are performed without FEA.

5. To determine loads, restraints, and material properties, pen and paper calculations played a vital role.

6. The FEA result's verification and validation with an independent set of calculations are crucial to provide confidence in the analysis results.

IV. ABBREVIATIONS

Abbreviations	Explanation
Ax	Acceleration along X-axis
Ay	Acceleration along Y-axis
W_f	Weight on front
W_r	Weight on rear
H_{CG}	CG height to the ground
$W_{Longitudinal}$	Longitudinal load transfer on a wheel
$W_{Lateral}$	Lateral load transfer on a wheel
FOS	Factor Of Safety
P_{act}	Actual Load
P_{cr}	Critical Load
E	Modulus of elasticity
d_c	Core Diameter
d	Nominal Diameter
d_i	Inner diameter
d_o	Outer diameter
σ_t	Maximum tensile stress
KL	Equivalent length of column end condition

V. METHODOLOGY

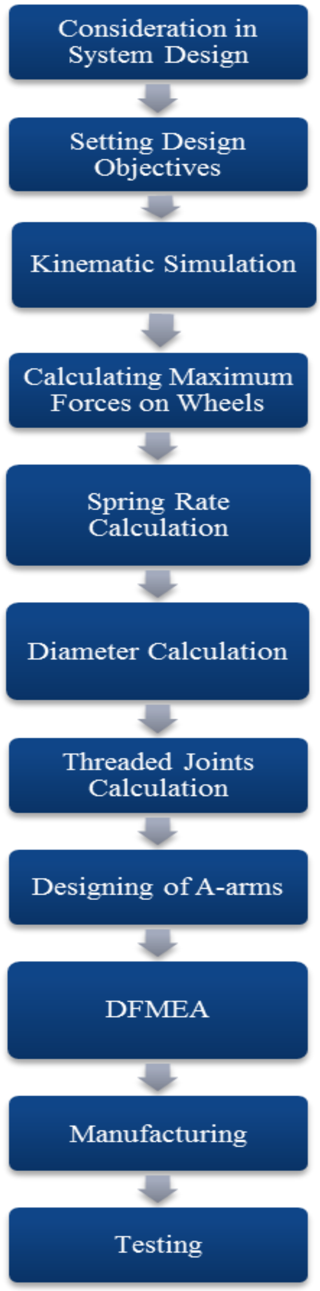

Fig3. Design Flowchart

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VI. DESIGN PHILOSOPHY

Choosing suspension geometries and components involves a wide range of choices and compromises. An analysis of the tire, suspension component, chassis, and road interaction is required to decide the trade-offs that will result in an optimum configuration for the type of vehicle.

The necessary steps in designing a vehicle's suspension are:

- Selection of the suspension type to be employed;
- Selection of the wheels;
- Establish the vehicle's dimensions: wheelbase and track width(s);
- Model the suspension geometry;
- Designing components.

TABLE I. KINEMATIC SIMULATIONS

Parameter	Front	Rear
Scrub Radius	56 mm	70 mm
Caster Trail	12.64 mm	0 mm
Roll Rate	132 Nm/deg	124 Nm/deg
Bump Coefficient	0.083 deg/mm	0.093 deg/mm
Roll Gradient	0.43 deg/g	0.43 deg/g
Wheel Rate	15.22 N/mm	11.73 N/mm
Jounce	30 mm	30 mm
Rebound	30 mm	30 mm

TABLE II. VEHICLE PARAMETERS

Parameter	Value (in mm)
Wheelbase	1545
Track (front)	1000
Track (rear)	1100
Static camber	-2°
Castor	4°
Static toe	2° (in)
Kingpin angle (front)	4°
Kingpin angle (rear)	0°
Roll center height (front)	7.42
Roll center height (rear)	32.55
Static weight distribution	45:55

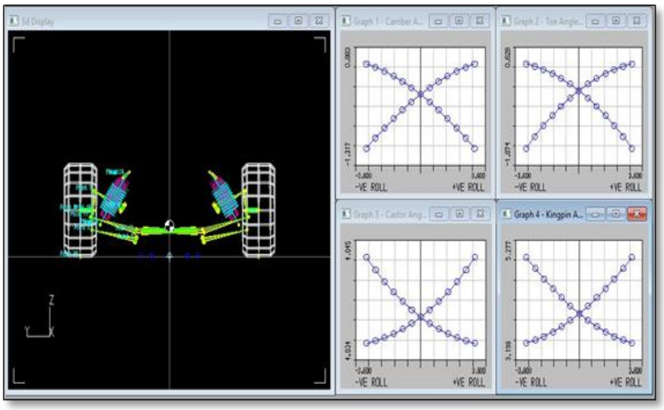

Fig4. Lotus simulation in roll condition

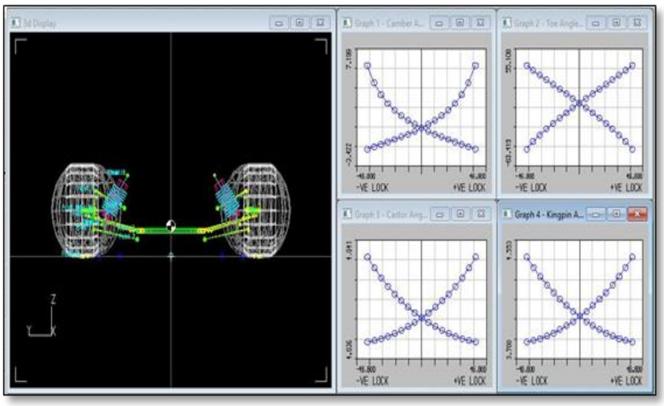

Fig5. Lotus simulation in steer condition

VII Load Transfer

The inertia of a mass is its resistance to change its state. For the change of state of a body, it must experience acceleration in its motion direction. Acceleration, braking, or cornering is nothing but the change of state of the vehicle, which offers resistance for a while and results in load transfer in a direction opposite to the force causing the change. No doubt, this load transfer persists for a short duration of time, but its impact on the car's performance is much more crucial.

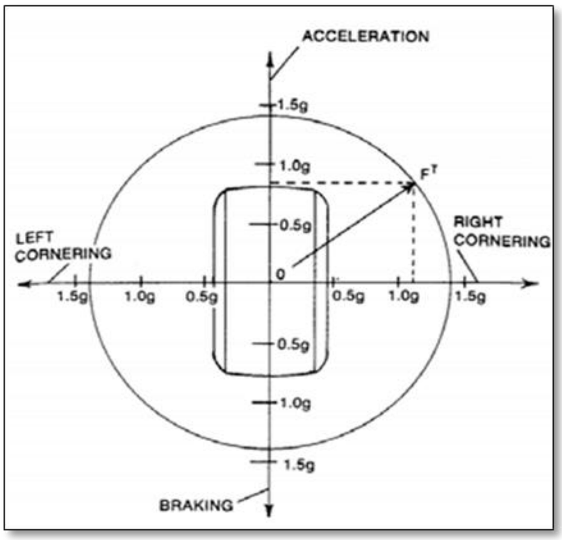

Fig6. Traction Circle

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The max longitudinal acceleration Ax = 1.2g
The max lateral acceleration Ay = 1.4g
However, the maximum acceleration experienced by the vehicle is during cornering accompanied by braking and given by:

$\sqrt{1.2^2+1.4^2} = 1.84g$ (1)

Thus, maximum acceleration = 1.84g

Now, the weight of the car is 250 kgs.
$W_f: W_r = 45: 55$
$\therefore W_f = 112.5$ kg = 1103.62 N
$\therefore W_r = 137.5$ kg = 1348.87 N.

Weight on single wheels in steady-state is:
$W_{f\,(single)} = 56.25$ kg = 551.81 N
$W_{r\,(single)} = 68.75$ kg = 674.43 N

The total longitudinal load transfer can be calculated as
$\Delta W_{longitudinal} = (W * H_{CG} * Ax)$/wheelbase
$\Delta W_{longitudinal} = (250*135*1.2)/1545$ (2)
$\boldsymbol{\Delta W_{longitudinal}}$ = **26.21 kg = 257.15 N** (3)

The total lateral load transfer can be calculated as
$\Delta W_{lateral} = (W*H_{CG}*Ay)$/trackwidth
$\Delta W_{lateral} = (250*135*1.4)/1000$ (4)
$\boldsymbol{\Delta W_{lateral}}$ = **47.25kg = 463.52 N** (5)

However, the max load transfer can occur during cornering and braking. This will be the diagonal load transfer on the front outer wheel.

$\Delta W_{diagonal} = \sqrt{(\Delta W_{longitudinal}^2+\Delta W_{lateral}^2)}$
$\Delta W_{diagonal} = \sqrt{(26.21^2+47.25^2)}$ (6)
$\boldsymbol{\Delta W_{diagonal}}$ = **54.03 kg= 530.06 N** (7)

Thus, the maximum load on a single wheel is:
$\Delta W_{max} = W_{f\,(single)} + \Delta W_{diagonal}$
$\boldsymbol{\Delta W_{max}}$ = **110.28 kgs = 1081.84 N** (8)

VIII SPRING RATES

Front spring

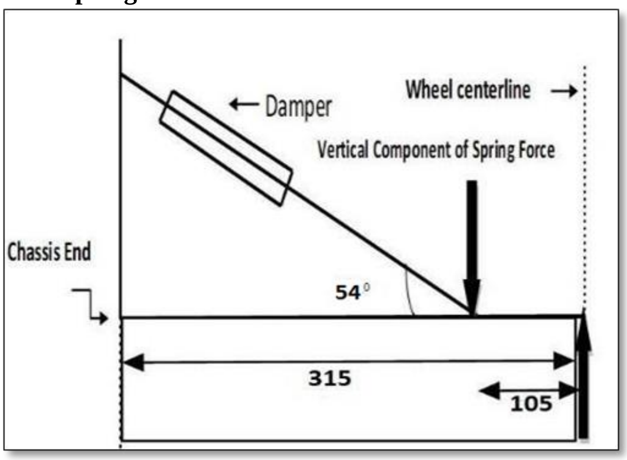

Fig7. Free Body Diagram

Sprung mass = 250 kg (approx.)
The factor for static to dynamic conditions: 2.1
According to the mass distribution of 55:45 (Rear: Front)

Mass per wheel (Front) = 56.25 kg
Angle of inclination of the strut = 54(from horizontal)
Point of attachment of strut = 210 mm from chassis end (from suspension geometry)
Reaction force acting from the ground on the wheel=
(56.25kg * 9.81) N =551.8125 (9)

Considering the A-arm hinges (A) as the point about which moment is taken;
The horizontal distance of reaction force from hinge point = 315 mm from suspension geometry
The horizontal distance of strut attachment point from hinge point = 210 mm
By taking moment about hinge points:
551.81 * 315 = Spring Force * sin 54 * 210 (10)
Spring Force = 1023.11 N
Considering the dynamic factor,
Dynamic force acting on the spring = 2046.22N (11)

Hence,
Spring Stiffness = Dynamic Spring Force / Deflection
= 2046.22/ 55.8
= 36.67 N/mm ≈ 37 N/mm (12)

By using a similar procedure,
For Rear

Spring Force =1384.98N (13)
Dynamic Force acting on spring =2285.22
Spring Stiffness = 40.8 N/mm (14)

Fig8. Double wishbone suspension system on car

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IX DESIGN OF A-ARM

As calculated above, the maximum forces on the wheels are:

A. For Front

F_z =110.28*9.81=**1081.84 N** (15)
F_x = 1.2*9.81*(56.25+13.10) =816.388 N (16)
Force on single A arm =816.388/2 = **408.19 N**
F_y=1.4*9.81*(56.25+23.62) =1096.93 N (17)
Force on single A arm = 1096.93/2 = **548.46 N**
Calculated -Spring Force =**1023.11 N.**

The analytical calculation for:

Threaded Joints

$P_{act} = \sqrt{[(F_x^2+F_y^2+F_z^2)^2 + (\text{spring force})^2]}$
$P_{act} = \sqrt{1279.76^2+1023.11^2}$ = 1638.45 N
(18)
S_{yt}=215 MPa FOS=2
For permissible tensile stress
$\sigma_t = S_{yt}/FOS$ =215/2 =107.5 N/mm^2
(19)
$\sigma_t = P_{act}/[(\pi/4)*d_c^2]$
(20)

Now, $d_c = \sqrt{[(P_{act}*4)/\pi*\sigma_t]}$ = 4.40 mm
(21)
d=4.40/0.8 = 5.5 mm [(h=0.8d) the factor determining threads will not fail in shear.]
d=5.5≈ 6mm

Designation	Nominal or major dia d/D (mm)	Pitch (p) (mm)	Pitch diameter d_p/D_p (mm)	Minor diameter (mm) d_c	D_c	Tensile stress area (mm^2)
M 6×1	6	1.00	5.350	4.773	4.917	20.1
M 6×0.75	6	0.75	5.513	5.080	5.188	22.0
M 8×1.25	8	1.25	7.188	6.466	6.647	36.6
M 8×1	8	1.00	7.350	6.773	6.917	39.2
M 10×1.25	10	1.25	9.188	8.466	8.647	61.2
M 10×1	10	1.00	9.350	8.773	8.917	64.5

Fig9. Basic dimensions of screw threads (fine series)

- Fine threads are used where the parts are subjected to dynamic load and vibration.
- Hollow thin-walled parts where coarse threads are liable to weaken the wall considerably
- These are used in parts which are used for the purpose of adjustment.

Hence from the standard reference table of fine series, we have chosen **M6 x 1**
Accordingly, the POS 6 rod end was selected.

Considering this as the inner diameter, the outer diameter was calculated by the following procedure:

Analytical Calculation for diameter calculation:

For Front

$FOS = P_{cr}/P_{act}$ (22)
$P_{cr} = (\pi^2 EI) / (KL)^2$ (K=2 Because the condition is one end fixed, other free) (23)
$(FOS*(KL)^2*P_{act})/\pi^2*E = (\pi/64)*(do^4-di^4)$ (24)
E = 205*10^3 MPa

Mechanical Properties	Metric	English
Hardness, Vickers	170	170
Ultimate tensile strength	440 MPa	63800 psi
Yield strength	370 MPa	53700 psi
Modulus of Elasticity	205 GPa	29700 ksi
Bulk Modulus	140 GPa	20300 ksi
Shear Modulus	80 GPa	11600 ksi
Machinability	78%	78%
(Source: www.matweb.com; ASM International, 1990).		

Fig10. Mechanical properties of AISI 1018 steel

$(2*(2*210)^2*1638.45*64)/(\pi^2*205*10^3*\pi) = do^4-di^4$ (25)
Considering **d_i=6mm**
d_o = 9.18 ≈ 10 mm

R5	R10	R20	R40
1.00	1.00	1.00	1.00
			1.06
		1.12	1.12
			1.18
	1.25	1.25	1.25
			1.32
		1.40	1.40
			1.50

Fig11.1. Preferred series

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1.60	1.60	1.60	1.60
			1.70
		1.80	1.80
			1.90
	2.00	2.00	2.00
			2.12
		2.24	2.24
			2.36
2.50	2.50	2.50	2.50
			2.65

Fig11.2. Preferred series

By using the preferred series R5 which is 1,1.6,2.5,4,6.30,8,10, the diameter 10 was selected.

Hence, d_o=9.18 ≈ 10mm

Accordingly, an A-arm endplate was designed. To accommodate the spherical bearing, a bush (with accurate tolerance) was inserted in the laser cut part. This ensured proper fitment of the spherical bearing.
Slots on the laser cut part ensured accurate positioning of the damper mounts.

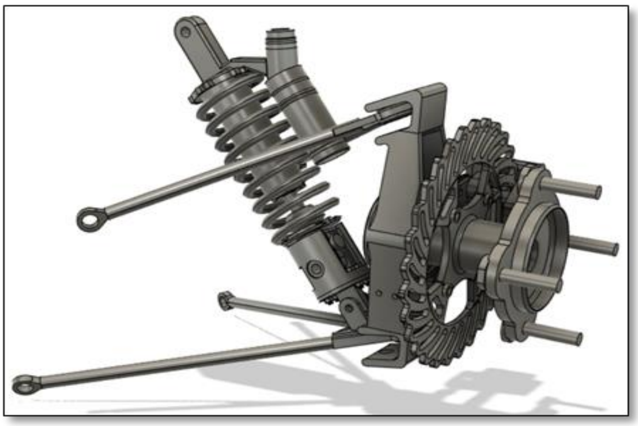

Fig12. Suspension Assembly

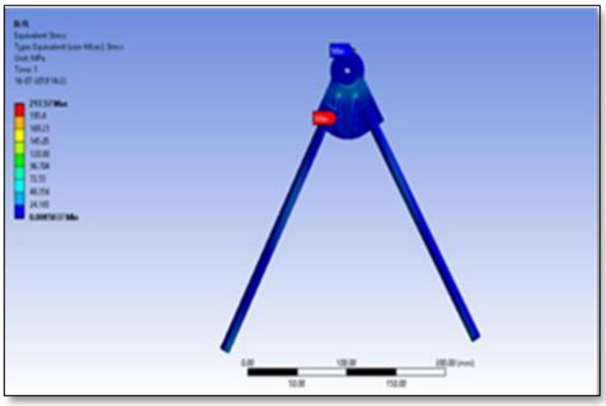

Fig13. Max Von-Misses stress = 217.57 MPa

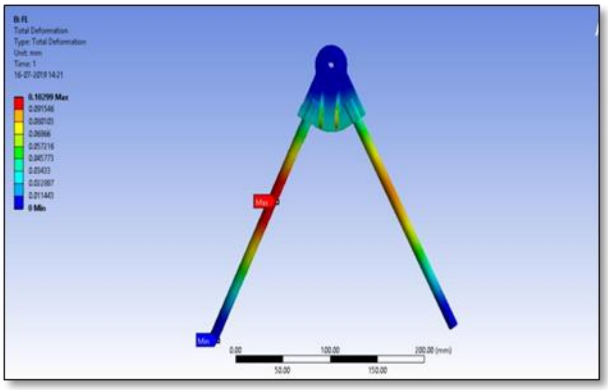

Fig14. Max deformation = 0.102mm

X. Conclusion

The purpose of this paper was to find the diameter of the double-wishbone by analytical calculations. The force is calculated using basic concepts. This paper gives a clear idea of how the forces are taken into consideration. Material is selected based upon calculated forces. Double-wishbone is designed using suspension points and dynamic force applied considering the factor of safety. Design is validated by using Ansys 16.0 software. This design is fabricated and tested in the Formula Student race car in all dynamic conditions. No failure occurred at the time of testing, it can be concluded that forces calculations and design are up to the mark.

XI ACKNOWLEDGMENT

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