HANDOUT 2

TRACING LOADS FROM TIRES to CHASSIS

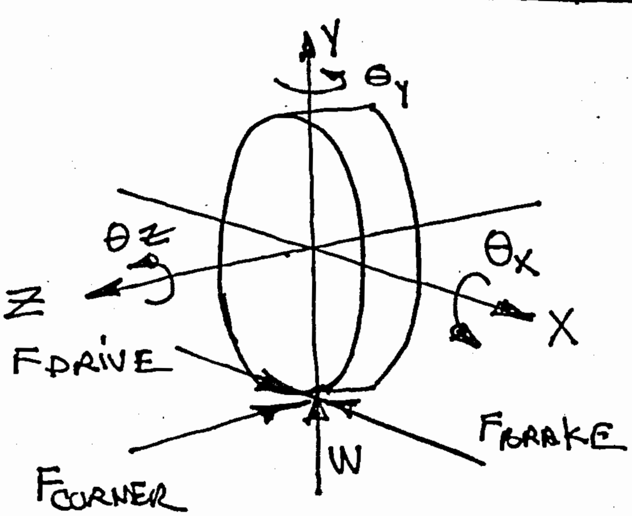
Handout 1 DEALT with ESTIMATING LOADS at the bottom of tires in CORNERING, ACCELERATION, BRAKING & Bump — HERE WE TRACE those loads through the suspension for a Double-A arm suspension. WE use the methods of statics in 3 dimensions where needed, AND will develop some simple but useful relationships. Also we introduce some of the terminology of suspension with a few comments on what is "good."

⇨ BEFORE CLASS, EXAMINE the 2001 FSAE CAR (OR BII)
AND EXAMINE the components of suspension:

- "KINGPIN" or "upright"
- steering ARM
- HUB / Bearings / axle
- SPRING Link / Rocker
- VARIOUS clearances at Bump / droop / steering angles
- Upper & Lower A-arms
- TIE Rod
- pivots: Rod ends & spherical Brgs
- Front Brake Assy
- Rear Brake assy

Since we are operating under the umbrella of "DESIGN", I'll make a few remarks about the function of any suspension. Figure 1 shows a tire & wheel with the ground loads and possible motions. The function of a suspension is to transfer these forces to the chassis, provide MEANS of applying F_{DRIVE} and F_{BRAKE}, and control motion.

Direction	Motion Function	Forces – Path to Chassis
Y	Bump & Rebound Travel: provide specified values	Transmitted via spring-shock units & susp. members
X	Fore-aft movement: controlled to be zero or small	Transmitted via susp. members.
Z	Lateral motion: controlled to be zero or small	Transmitted via susp members.
Θ_Y	"Steer" motion: provide specified range at front, and controlled to be zero or small changes during bump and rebound ("Bump steer")	Transmitted via steering arm & tie rod and susp. members.
Θ_X	Camber change: controlled to be small during bump & rebound	Transmitted via susp. members.
Θ_Z	Rolling motion: must do this! usually with hub, bearing & axle options	Drive & brake torques provide F_{DRIVE} & F_{BRAKE}; reactions via susp. members



Also: some forces may not act at the center of the contact patch and cause addl. torques about the X and Y axes

FIGURE 1 – Forces & motions

PACKAGING ISSUES - AND initial DESIGN choices. THE NEXT FEW PAGES ARE FROM "RACE Car Vehicle Dynamics", by Milliken & Milliken, SAE, 1995, PAGES 624-627. The focus is on front SUSPENSIONS. Some comments are in the MARGINS

17.5 Front Suspensions

Introduction

Many types of front suspensions have been used over the years. They include various beam type axles with steering via kingpins at each end of the axle, the parallel trailing arm type such as the VW, the Morgan sliding pillar type, and the Chevrolet Dubonnet. In recent history, passenger car designs have come down to basically two types: the MacPherson Strut and the SLA (Short-Long-Arm). ← Double A-ARM

This chapter will deal only with the last two mentioned as these make up the majority of front suspensions that will be encountered. The other types suffer from either high bending loads, poor geometry, high friction, or a combination of these problems. The best way to discuss each type is to go through the design process step by step. For each step a decision has to be made that is often a compromise. By discussing these decisions, hopefully a feeling for the limitations of the design will develop.

Front Suspension Design Issues—General

The first task in designing a front suspension of any type is to establish the packaging parameters that are fixed, or absolutely cannot be changed for whatever reason (see Figure 17.17). These should be listed so that they are not overlooked. The next task is to package the wheel, tire, brakes, and bearings. This is done in car position, so the track width has to be known. If it is not yet established, it should be made as wide as practical. This sounds evasive, but there are trade-offs in everything, even things as simple as choosing the track width. For example, what do the rules allow? What is the predominant race

wheel, TIRE, BRAKES, BEARINGS, TRACK

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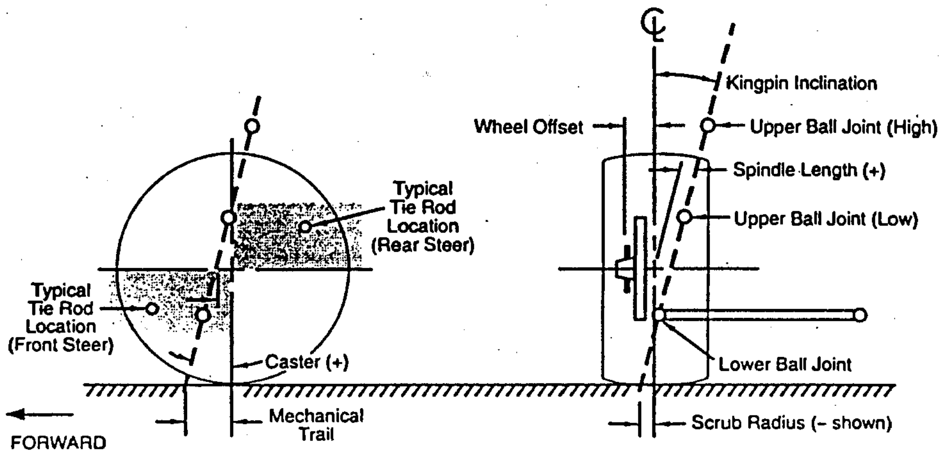


Figure 17.17 Front suspension packaging.

track type on which the car will run? Is top speed, thus low frontal area important? Are slow-speed tight street circuits of concern? All these issues can affect the decision on the basic track width!

TIRES & WHEELS

Tire size and rim diameter and width must be settled. The specific wheel manufacturer needs to be known and a cross section of the wheel is desirable for optimizing the use of that wheel. Tire sizes are usually limited by the sanctioning body rules. In general, use all the tire they will let you get away with. Another point is to always design for the latest sizes being developed by the suppliers; this guarantees that the latest compounds and constructions will fit your car. Remember, the tire is the single most important chassis component on the car.

BRAKES, BEARING LOAD COMMENTS on P.20

The wheel offset is worked out in conjunction with fitting the brake caliper to clear the inside surface of the wheel. Once the caliper is located, this automatically locates the brake rotor. With the rotor location comes the absolute farthest outboard location for the lower ball joint. Wheel bearings need to be looked at soon, as ideally they should be located such that the tire center is between the two rows of balls or rollers (to minimize loads on the bearings).

UPRIGHT/KINGPIN/SPINDLE

Now that the lower ball joint cross car boundary (lateral position) has been set, the height of the lower ball joint comes next. In production cars it must be above a 5-in. wash rack clearance requirement, but on race cars it should be made as low as possible for structural reasons. Usually there is no rule but some practical considerations such as deflated tire ground clearance might be in order. If it is totally inside the wheel all it has to do is clear the wheel and the brake rotor under all travel and load conditions.

"structural reasons": see EQNS 9 & 10

The decision about the kingpin angle in the front view is the next order of business. The issues here become scrub radius, spindle length, and kingpin angle. They are interrelated

16

AKA "Kingpin offset"

and a compromise is needed. If you want a certain scrub radius you now have two points established, i.e., the lower ball joint and the ground contact point of the kingpin (set by the scrub radius)—the kingpin angle becomes fixed automatically. If you want a certain kingpin angle then the scrub radius will not necessarily be what you want. Basically, on rear-wheel-drive cars push the lower ball joint out as far as possible and run a fairly low kingpin angle, less than 8°, and accept the scrub radius that results. If you are dealing with a front-wheel-drive car you **must** minimize the spindle length and have a negative scrub radius. This may result in a kingpin angle as high as 16°, but you will have to accept it or find another clever way around it.

Lower ball joint: see eqn(2) make "c" small

Kingpin angle affects the performance of the car when the wheels are steered. One concept that should be understood is that the more the kingpin angle the more the car is lifted when it is steered. This is one source of steering returnability, the weight of the car returns the steering to center. The amount the car is lifted is also a function of the spindle length where a longer spindle means more lift.

angle ← [WORD MISSING IN TEXT]

The camber of the wheels when steered is a function of the kingpin angle and the caster angle. With no kingpin angle (and no caster angle) there is no camber change with steer lock. As kingpin is added (but still no caster) the wheel will "lose" camber with steer lock, or in other words it will change in a direction giving positive camber on the outside wheel.[60] As caster is added this modifies the effect of kingpin. With positive caster and no kingpin angle, the wheel gains negative camber on the outside wheel and positive camber on the inside wheel. Thus caster can add favorable camber angle to the effects of kingpin angle. In other words, the reason that low kingpin angles are desirable is that kingpin angle subtracts from the negative camber gain due to caster on the outside wheel.

GOOD

The decision on a rack location depends on several packaging factors such as engine location and orientation, front-wheel drive vs. rear-wheel drive, whether it is to be high- or low-mounted, etc. In addition there are performance reasons for choosing the rack location. First we must assume that every structure is a spring and should be treated as such. As an example the rack mounting stiffness versus the upper or lower control arm mounting stiffness to the chassis will not necessarily be the same. Therefore, when a cornering force is applied, any difference in the lateral displacement of the ball joints in relation to the tie rod outer pivot will cause a steer angle. To assure stability it is better to have lateral force deflection toe-out (lateral force understeer) rather than toe-in. We can assure that this happens by the proper location of the rack. If a high-mounted rack is required it must be behind wheel center and if it is low-mounted then it must be ahead of wheel center as shown by the shaded areas in Figure 17.17.

Steering Rack Location

must? Depends on loading

Structural requirements for the suspension design must always be considered when packaging each element of the total system. Control arms that have one leg straight across from the ball joint are superior in system stiffness to arms that are splayed. Establishing

Like on our solar cars

60 That is, the wheel further from the turn center.

linkage ratios for the spring, shock, and stabilizer bar as close to 1:1 as possible will provide more direct load paths thus improving system stiffness while providing a lighter overall design.

Front Suspension Design Issues—SLA

The Short-Long Arm (SLA) suspension is the choice of designers without question for its ability to meet desired performance objectives with minimum compromise.

The design starts with the basic package as described above. The details of the track width, the wheel size, the tire, the brakes, etc., bring about the location available for the lower ball joint. The upper ball joint is located either via kingpin angle requirements or by scrub radius requirements. There is a little more freedom with the SLA that is not available to the strut design and that is the choice of a short knuckle or a tall knuckle.

The *short knuckle* means the upper ball joint is located basically within the diameter of the wheel. With high offset and large-diameter wheels the kingpin angle can be kept small (while achieving small spindle lengths and scrub radius) by tucking the upper ball joint into the wheel.

Like our FSAE cars (Front)

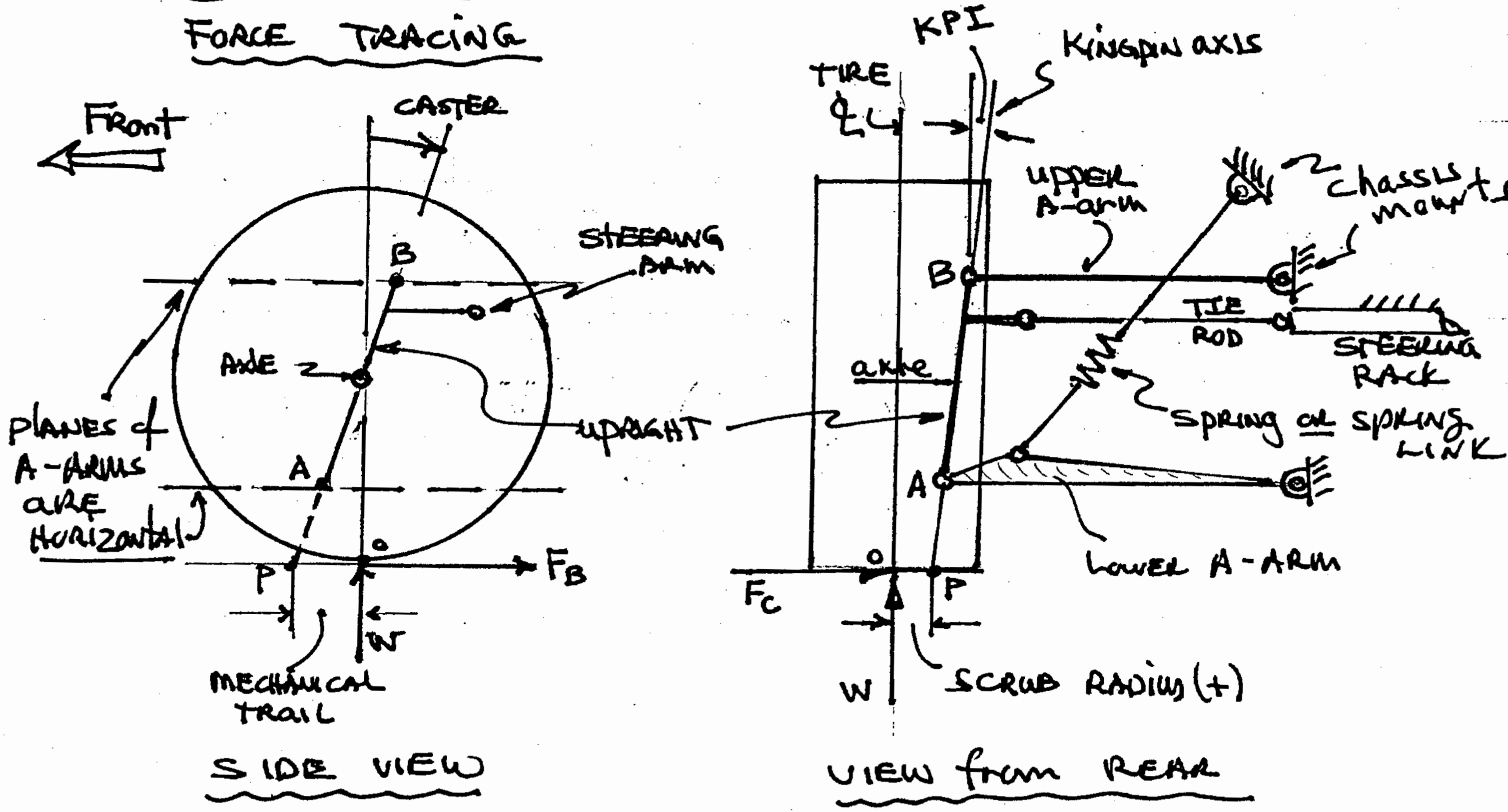
To reduce the loads on the control arms and other suspension components, it is desirable to have a long kingpin length, that is, separate the upper and lower ball joints as much as possible. Depending on details of the installation, the short knuckle may yield less than optimum kingpin length. The other alternative is the *tall knuckle* where the upper ball joint is above the tire. In the tall knuckle design the ball joints naturally have a very large span and thus reduce reaction loads. This option also allows reasonable kingpin angles while achieving desired spindle length and scrub radius. Another advantage for the tall knuckle is that build errors will result in smaller geometry errors than with short knuckle designs. Some negatives to the tall knuckle, of course, are the added structural requirements of the knuckle, and the limitation of never changing tire size or width without widening the track and increasing the spindle length and scrub radius after the design is completed.

Like the 99 BAhA CAR

With the upper and lower ball joint locations established, the tie rod outer point should also be set per the requirements established in Chapter 19 on steering geometry.

FRONT SUSPENSION

FORCE TRACING



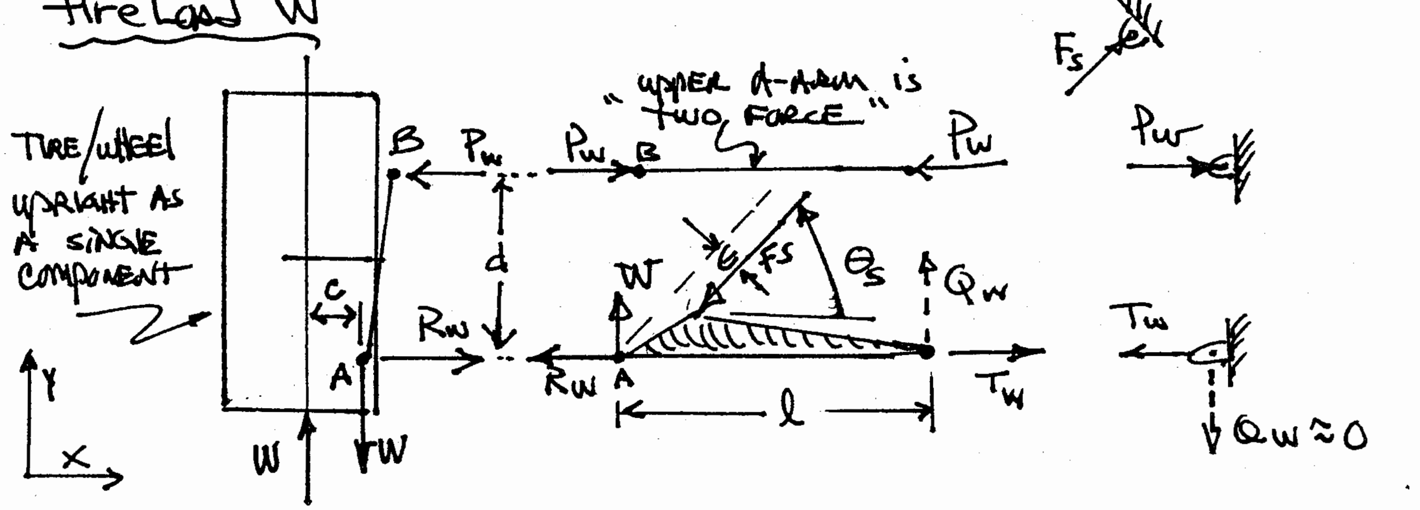
- A = Lower ball joint.
- B = upper balljoint.
- AXLE shown on LINE thru A-B in <u>SIDE VIEW</u>.
- A-ARMS are shown as horizontal (simplifies the analysis).
- KPI = KING PIN INCLINATION ANGLE (from vertical).
- Point O is Geometric center of Tire contact.
- Point P is where KINGPIN AXIS INTERSECTS Ground.
- Spring (or spring link) mounts on Lower A-arm.
- FORCES F_B, F_c, W act at point O.

WE will TRACE the F_B, F_c and W forces to upright pivots A & B, then FROM A & B to the A-ARM mounts and spring mount.

- All pivots will be treated as frictionless balljoints.
- Where needed, we will use statics eqns in 3 Dimensions, summing forces in "x, y, z" directions and torques about each axis.

- Some simplifications will be employed — like leaving out the steering forces in the initial analysis, then treating them separately, even though they interact.
- Also, the caster, KPI and scrub radius can cause the vertical load W to have a component that provides steering torque about the kingpin axis. We'll ignore this.
- Finally, we'll use superposition, by analyzing the effect on each tire load separately, then adding the results — this will provide insights as to which tire loads and dimensions contribute to the various forces.

Here is a FBD of the disassembled susp with tire load W



The upper A-arm is a two force member, so the line of action of its force is along the arm — we call the force P_W and chose its direction as shown. The lower arm is not two force due to the spring force F_s, so there can be forces at A in the x & y directions. The x direction force is defined as R_W, and the y direction force must equal and oppose the tire load W, since it's the only other vertical force.

Static eqns for the Tire/wheel/upright gives:

$\sum F_X$: $R_W - P_W = 0 \Rightarrow R_W = P_W$ (1)

$\sum T_A$: $P_W(d) - W(c) = 0 \Rightarrow P_W = W\left(\frac{c}{d}\right)$ (2)

NOTE: if c = small or ≈ zero, then $R_W = P_W \approx 0$

Examine the lower A-arm — note distance ϵ, between the line of action of F_s and a parallel line through A. Suppose we sum torques about A.

$\sum T_A$: $Q_W(\ell) = F_s(\epsilon) \Rightarrow Q_W = F_s\left(\frac{\epsilon}{\ell}\right)$ (3)

By design, we can aim the spring or spring link at pivot A (or close to A) so $Q_W \approx 0$ (4)

If $Q_W \approx 0$, then:

$\sum F_Y$: $W - F_s \sin\theta_s = 0 \Rightarrow$

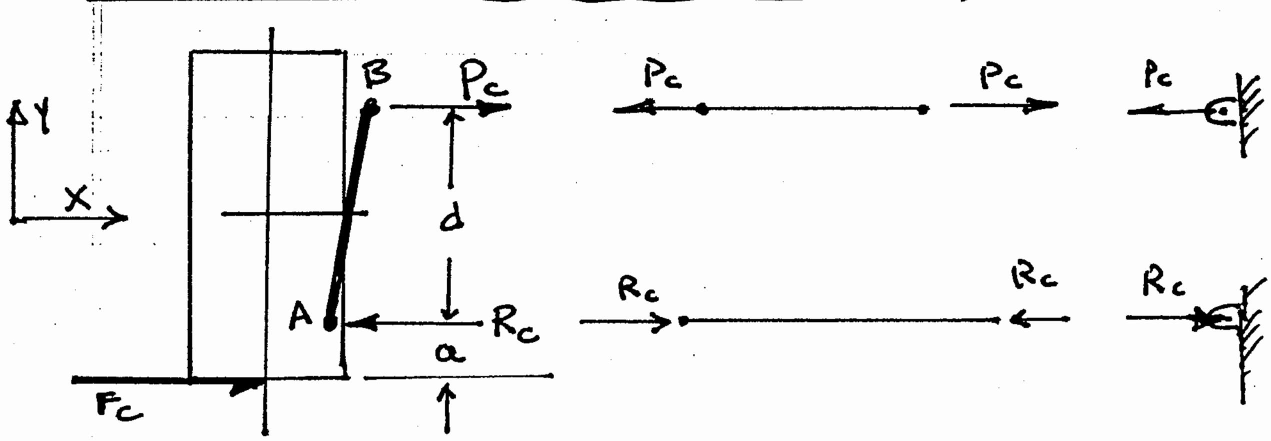
$F_s = W/\sin\theta_s$ (5)

AND FROM $\sum F_x$:

$$T_w - R_w - F_s \cos\theta_s = 0$$

$$T_w = R_w + F_s \cos\theta_s \qquad (6)$$

NOW EXAMINE CORNERING LOAD F_c



F_c DOES NOT INTRODUCE ANY vertical forces – a consequence of assuming horizontal (or nearly horizontal) A-arms.

FORCES P_c and R_c are introduced as shown.

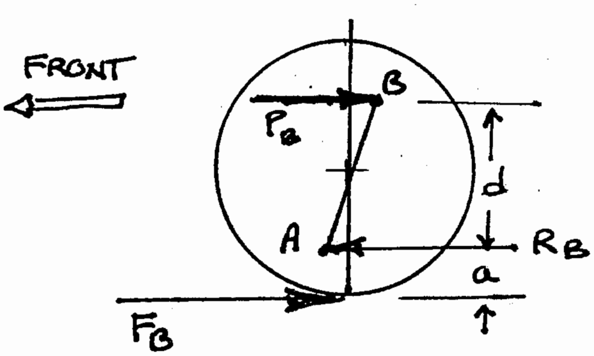
$$\sum F_x = 0: \quad F_c + P_c - R_c = 0 \qquad (7)$$

$$\sum T_A = 0: \quad F_c(a) - P_c(d) = 0 \qquad (8)$$

FROM 8: $$P_c = F_c\left(a/d\right) \qquad (9)$$

FROM 7 & 9: $$R_c = F_c\left(\frac{a+d}{d}\right) \qquad (10)$$

AND BRAKING LOAD F_B ~ TREAT wheel/BRAKE/upright AS the component



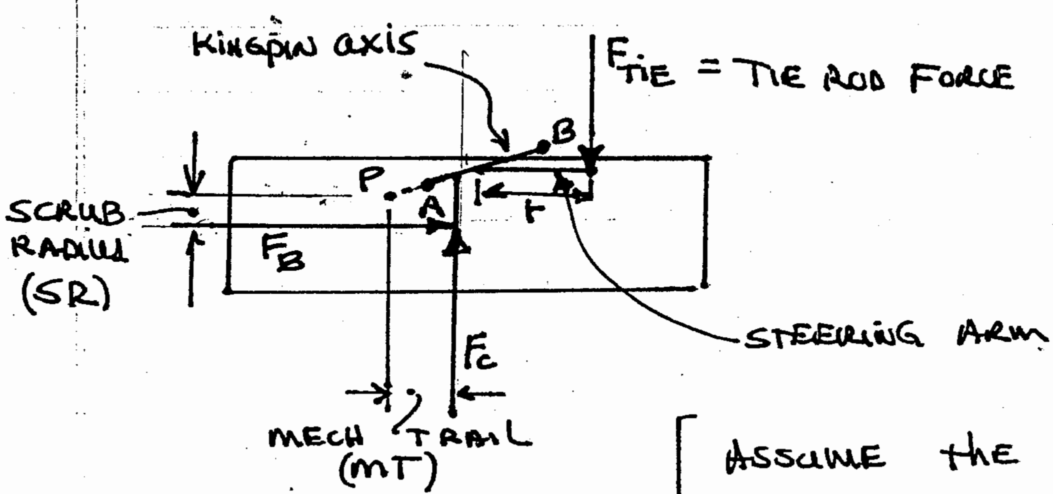
BY ANALOGY with EQNS (7–10):

$$P_B = F_B\left(a/d\right) \qquad (11)$$

$$R_B = F_c\left(\frac{a+d}{d}\right) \qquad (12)$$

Now Look at the steering Loads

Top view of wheel/upright:



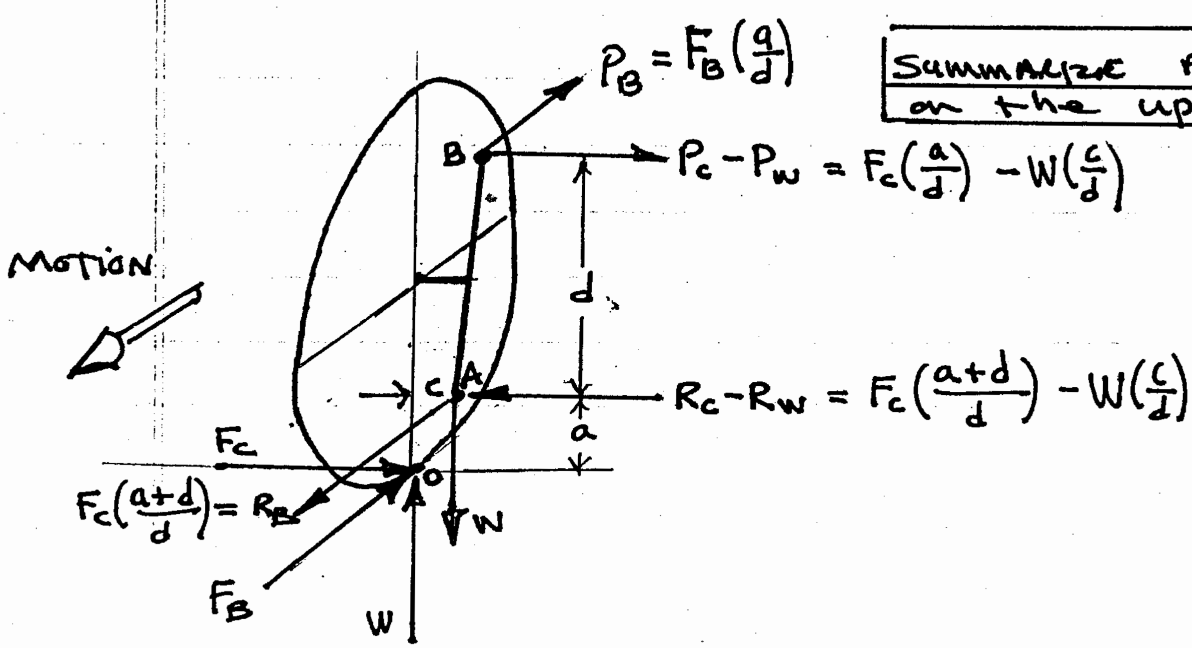
$$F_{TIE} = \frac{F_B(SR) + F_c(MT)}{r} \qquad (13)$$

Assume the caster and KPI angles are small, then sum torques around the kingpin axis

NOTE that F_{TIE} was left off of the analysis for cornering load F_c and braking F_B. F_{TIE} will slightly alter the lateral forces. Whether these forces increase or decrease depends on the steering arm location with respect to the kingpin axis (in front or behind)

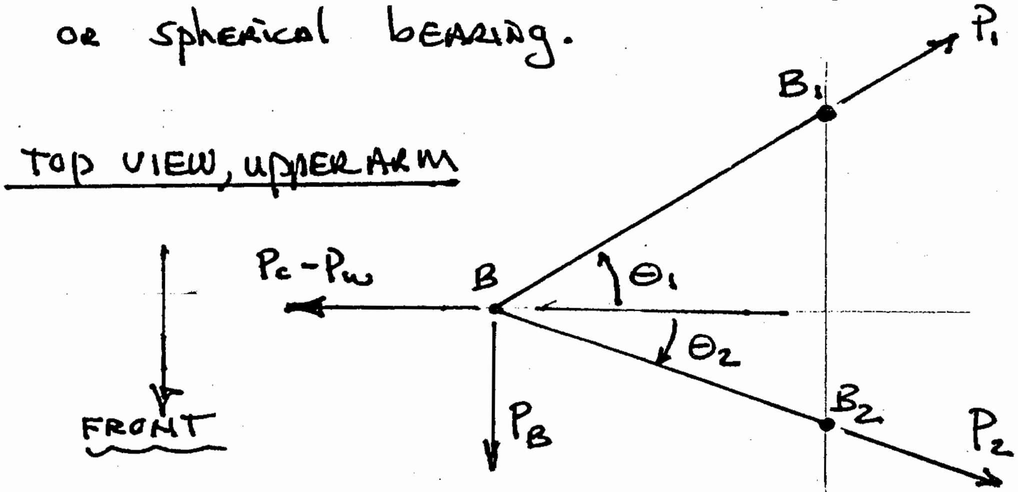
EQN 13 can be used to estimate the load on the steering rack.

SUMMARIZE FORCES ON THE UPRIGHT (14)



FORCES AT A & B act upon the outer END of the A-arms, and can be traced through the arms as follows. Assume EACH arm pivot is a Rod end or spherical bearing.

TOP VIEW, UPPER ARM

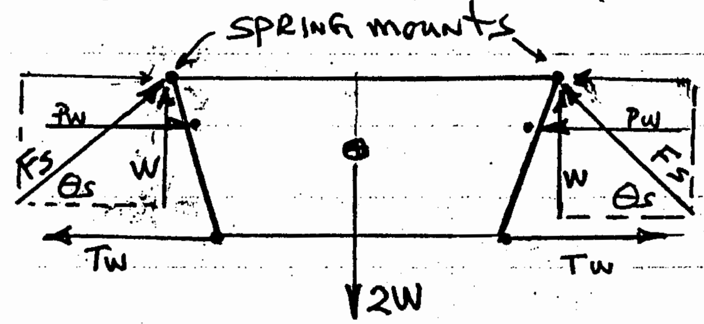


Introduce FORCES P_1 & P_2 - THE DIRECTIONS shown. Assume the A-ARM is designed so each leg can be treated as a two-force member. Angles θ_1 & θ_2 are design choices.

Using statics equations, forces P_1 & P_2 can be expressed in terms of P_c, P_B, P_w and θ_1 and θ_2.

What Holds the car up?

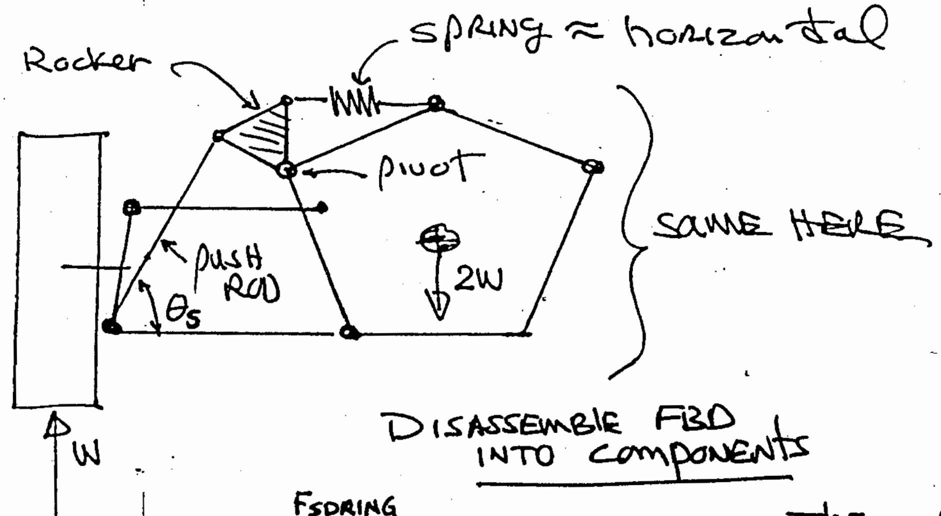
EXAMINE THE CHASSIS REACTIONS USING THE PREVIOUS FRONT SUSPENSION ANALYSIS



(W = WT PER WHEEL)

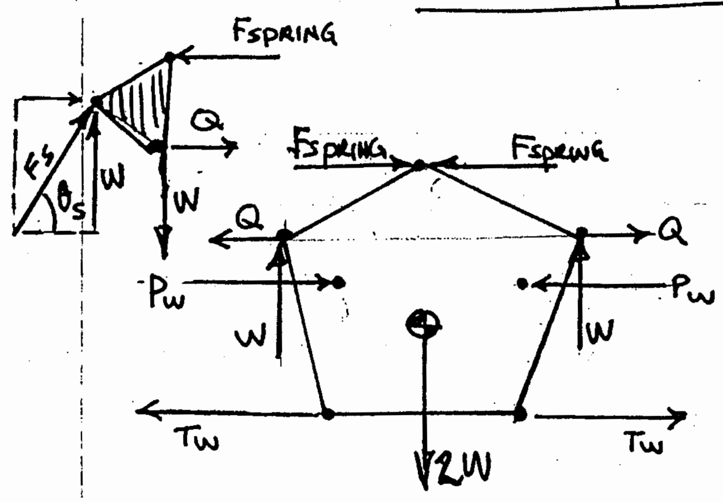
With A-ARMS "about horizontal, the only vertical forces are acting at the spring mounts. The vertical component of F_s is $F_s \sin\theta = W$ (EQN 5).

Same idea for Pushrod susp as in FSAE CAR.



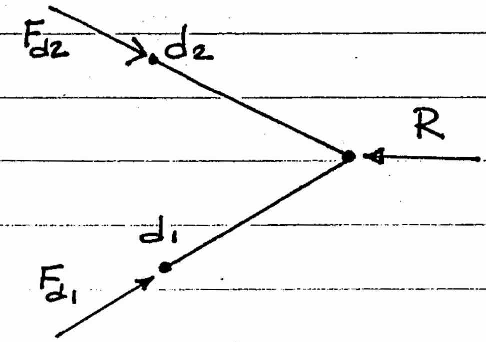
FBD of Pushrod Actuated springs. FORCE F_s is "IN" the pushrod

DISASSEMBLE FBD INTO COMPONENTS



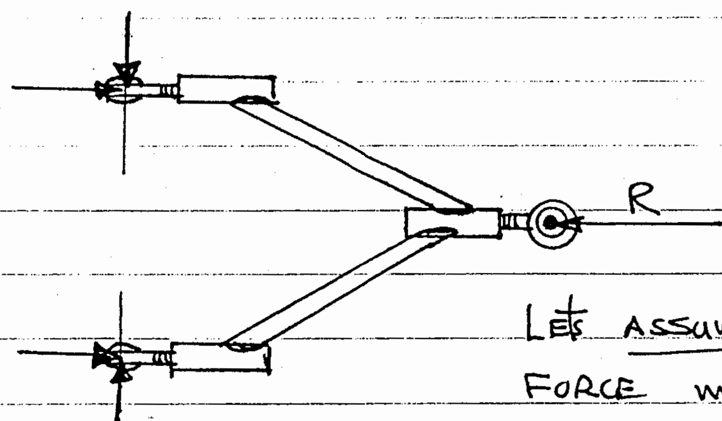
The ROCKER pivots hold the car up, supporting W on each side. You'll SEE some MARGINAL supports for these pivots on FSAE CARS.

A-ARM DETAIL - WE'VE SHOWN A-ARMS schematically as lines with joints at the ENDS, implying that EACH LEG could be viewed as a two-FORCE MEMBER, EITHER IN TENSION OR compression like this:



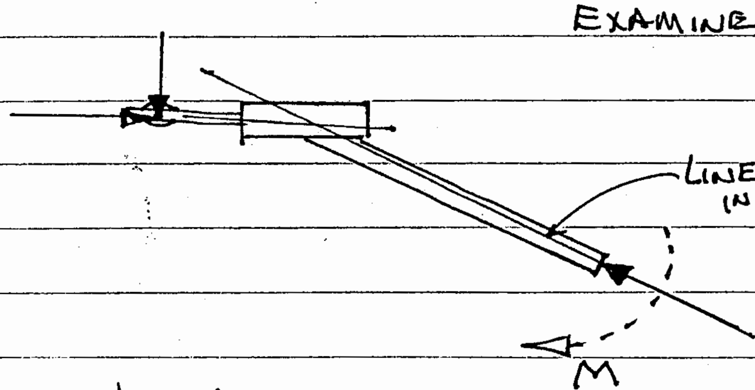
IN ACTUAL construction, it takes some design to achieve the TWO-FORCE MEMBER CONCEPT.

A COMMON A-ARM construction Looks like:



• ROD ENDS cannot produce moments about any pivot axis, only FORCES —

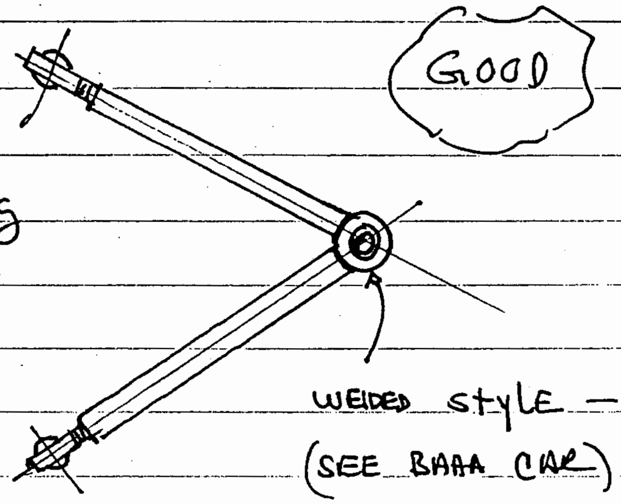
Lets assume the LEGS ARE TWO FORCE MEMBERS and see if Equilibrium could be achieved. EXAMINE the upper mount:



But This FBD cannot be equilibrium unless A BENDING MOMENT, M, IS INTRODUCED.

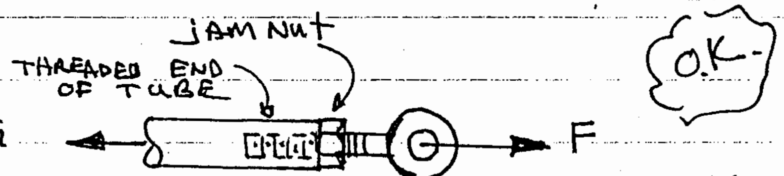
SO, This A-ARM DESIGN will introduce MOMENTS into the LEGS AND WELDS.

A GOOD A-ARM DESIGN will have the outer joint axis IN-LINE with the legs. This can be achieved by mounting A spherical Bearing (the "INSIDE" of A ROD END) IN A housing which is WElded OR machined into the legs. Then the legs ARE TWO FORCE MEMBERS.



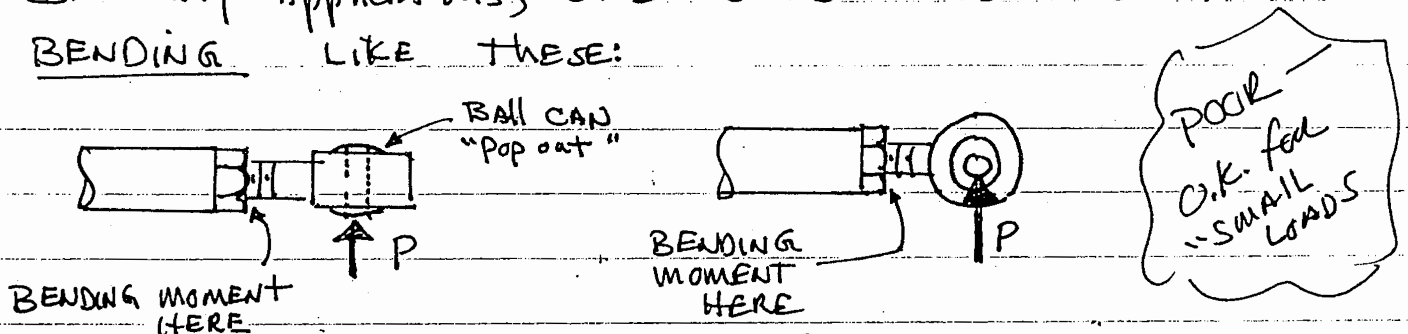
LOADING ROD ENDS

THE DESIGNED LOADING is This:



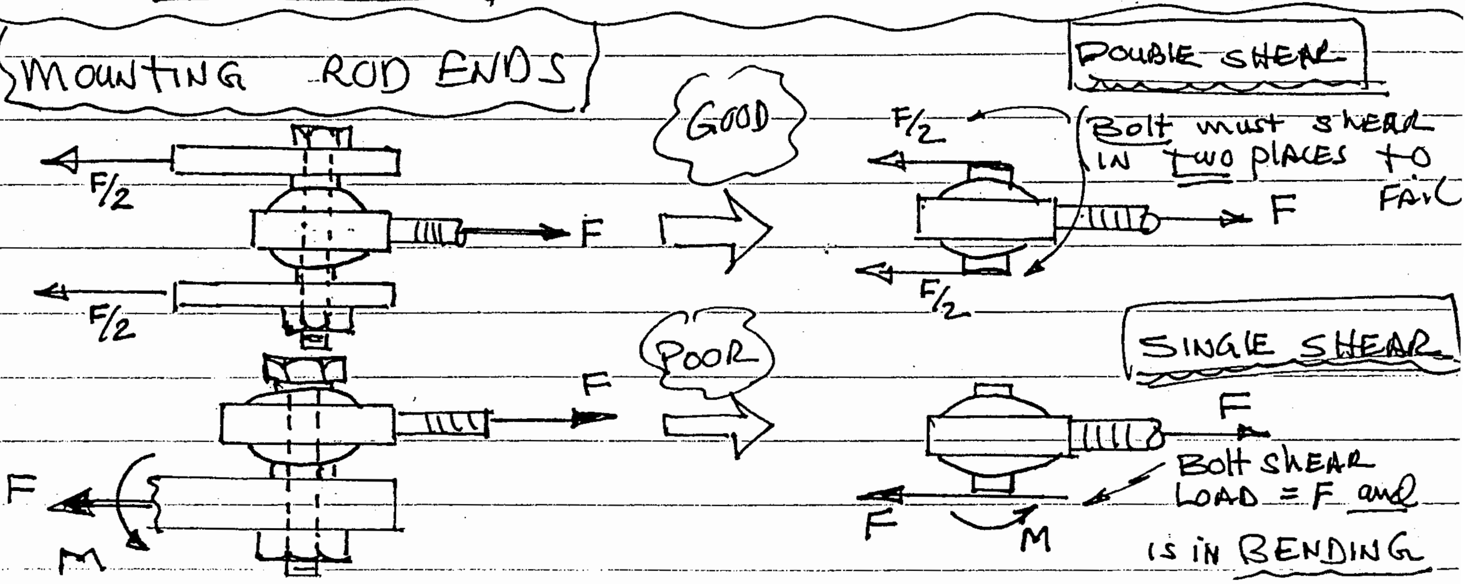
RATED LOAD EXPECTS This LOAD DIRECTION.

IN MANY applications, ONE SEES ROD ENDS IN BENDING LIKE THESE:



The LOAD P is around 10% of F, AND the THREADED SHANK must be SIZED AT the ROOT DIAM. to take the MOMENT

MOUNTING ROD ENDS



WHEEL BEARING LOADS – some observations from Fred Pahn in "How to Make Your Car Handle", p. 128. His "positive & negative offset wheels" really refer to the location of the tire centerline with respect to the bearing placement. His diagrams show the result of superposition of reactions due to weight (W) and cornering force (C). I added dimensions r and BS.

OBSERVE: $(BS)(C_i) = (r)(C)$

SO: $C_i = \left(\frac{r}{BS}\right) C$

Typical values: $r = 10''$, $BS = 2''$

so $C_i \approx 5C \Rightarrow$ cornering force is amplified at the bearings.

Also observe: $C_o = C_i$

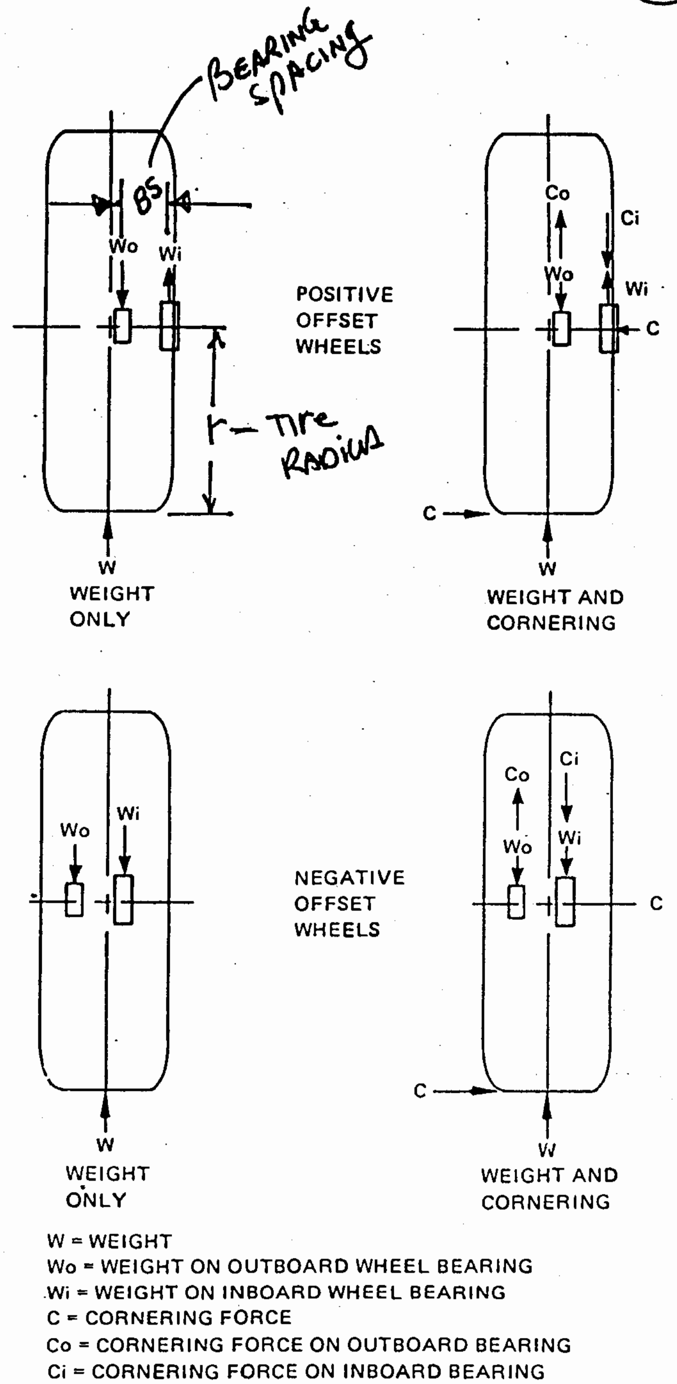


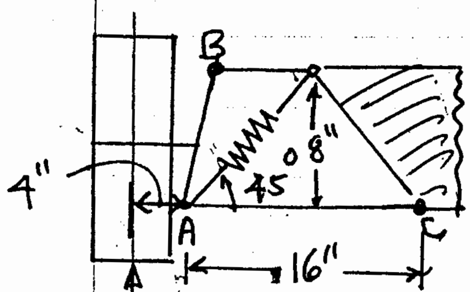
Figure 35/Wheel bearings carry weight of vehicle, "W", during straight-line running. They also carry cornering forces, "C", in turns.
With negative offset, weight loading is well distributed between inner and outer bear However, cornering loads upset this balance by increasing force on the inside bearing and decreasing force on the outside bearing. This makes cornering much harder on th inside bearing than straight-line driving.
With positive offset, the situation is reversed. In straight-line driving, bearing loads ar uneven but the addition of cornering forces tends to even them out. Thus cornering loads are less severe than those of straight-line driving.
In these sketches, offsets are exaggerated for clarity. Small changes in offset don't af bearing loads enough to worry about. In any event, wheel-bearing life will probably l affected more by your driving than anything else.

Since our FSAE car corners at $f_c = 1.3$, then $C > W$, and its amplified. So bearing loads due to cornering far exceed those due to vehicle weight, W_o & W_i. ← Discuss Bump Loads too.

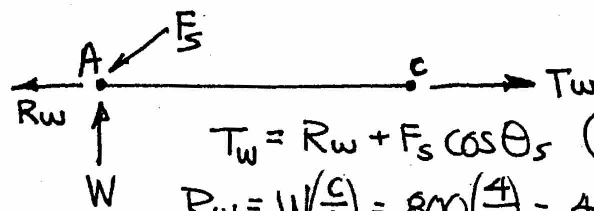
Comments on DEFORM Topics

EXAMINE two spring mount locations on a lower A-arm of a front suspension, AND find the A-ARM stresses.

CASE 1 SPRING AIMED at lower Ball joint



$W = 800$ Lbs — 200Lb static & 4G Bump



$T_W = R_W + F_s \cos\theta_s$ (EQN 6)

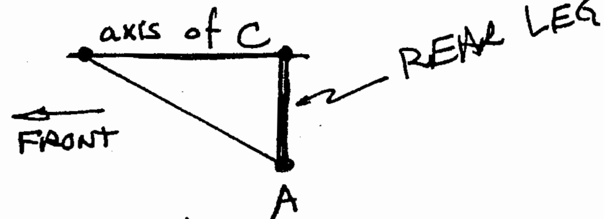
$R_W = W\left(\frac{c}{d}\right) = 800\left(\frac{4}{8}\right) = 400$ Lb

$F_s = W/\sin\theta_s = \frac{800}{.707}$

$(\sin\theta_s = \cos\theta_s = .707)$

so $T_W = 400 + 800 = 1200$ Lb

suppose the REAR LEG of the A-ARM is 90° to the vehicle ℄: so it takes All the T_W load.

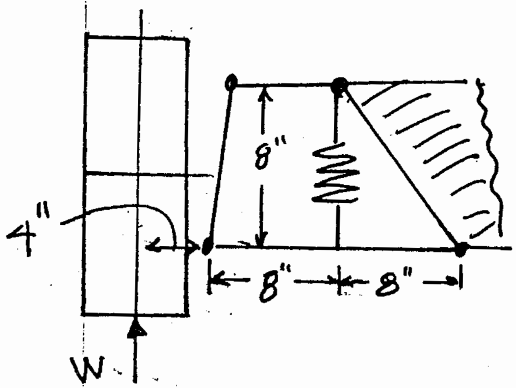


suppose we choose a steel tube 1" O.D. × .035 wall — The TUBE is IN TENSION with stress $= \frac{T_W}{AREA}$

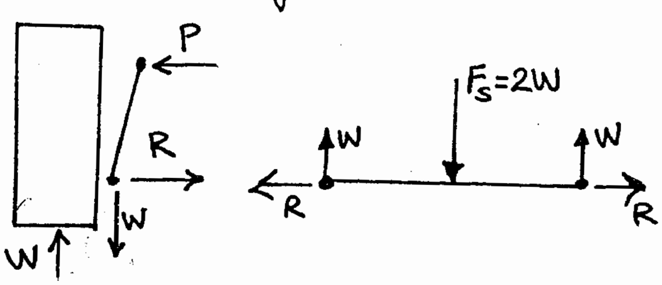
FROM THE TABLE, the AREA = 0.1061 sq"

so TENSILE STRESS $= \frac{1200 \text{ Lb}}{.1061} = 11{,}310$ psi

CASE 2 SPRING mounts IN MIDDLE of A-ARM



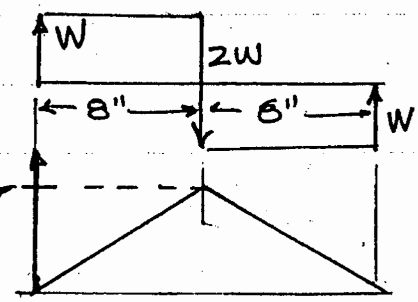
PREVIOUS ANALYSIS DOES NOT APPLY HERE, SO EXAMINE FBD's of wheel & Lower A-arm.



TENSILE LOAD $R = W\left(\frac{4}{8}\right) = 400$ Lbs and will be IGNORED. for the A-ARM ANALYSIS. The A-ARM is A simple BEAM with A

SHEAR DIAGRAM

AND MOMENT DIAGRAM



maximum moment = (8)W

BENDING STRESS EQUATION: STRESS = $\frac{Mc}{I}$ ← called "y" on Table

FROM TABLE $S = I/y$ (Table values)

so STRESS = $\frac{M}{S} = \frac{8W}{S} = \frac{8(800)}{S} = \frac{6400}{S}$, psi

for The 1" x .035 tube, $S = 0.0247$

so BENDING STRESS = $\frac{6400}{.0247}$ = 259,109 psi !

this far EXCEEDS the yield strength of common alloys – so WE would NEED EITHER:

1) "BIGGER" TUBING ~ INCREASE I/y.

2) BIGGER CROSS section near the middle, INCREASING I/y. (EXAMINE some ATV's)

3) change the design – aim the spring at point A !

Generally, if we can, we try to avoid bending situations if we want Light WEIGHT

MECHANICAL PROPERTIES—SHELBY AIRCRAFT TUBING—ROUND—Continued

STANDARD SIZES AS LISTED IN ARMY-NAVY AERONAUTICAL DESIGN STANDARDS—REVISED OCT. 1, 1942

Outside diameter inches	Thickness: Decimal of inch	Thickness: B.W.G. or fraction	Weight per foot, pounds W	Area of metal square inches A	Moment of inertia I	Section modulus S=I/y	Radius of gyration $r=\sqrt{I/A}$
1	.035	20	.3607	.1061	.0123	.0247	.3414
	.049	18	.4977	.1464	.0166	.0332	.3367
	.058	17	.5835	.1716	.0191	.0382	.3337
	.065	16	.6491	.1909	.0210	.0419	.3314
	.083	14	.8129	.2391	.0253	.0507	.3255
	.095	13	.9182	.2701	.0280	.0559	.3217
	.120	11	1.128	.3318	.0327	.0654	.3140
	.156	5/32	1.406	.4142	.0381	.0762	.3033
	.188	3/16	1.630	.4786	.0415	.0831	.2948
	.219	7/32	1.827	.5369	.0441	.0883	.2868
	.250	1/4	2.003	.5890	.0460	.0920	.2795
1 1/16	.049	18	.5306	.1560	.0201	.0378	.3588
	.065	16	.6928	.2037	.0254	.0479	.3534
	.095	13	.9821	.2888	.0341	.0642	.3437
	.109	12	1.111	.3265	.0376	.0708	.3393
	.120	11	1.209	.3553	.0401	.0755	.3359
	.156	5/32	1.511	.4449	.0470	.0885	.3251
1 1/8	.028	22	.3280	.0964	.0145	.0258	.3879
	.035	20	.4074	.1199	.0178	.0317	.3856
	.049	18	.5631	.1656	.0240	.0427	.3808
	.058	17	.6609	.1944	.0277	.0493	.3777
	.065	16	.7359	.2165	.0305	.0542	.3755
	.072	15	.8097	.2382	.0332	.0590	.3732
	.083	14	.9237	.2717	.0371	.0660	.3696
	.095	13	1.045	.3074	.0411	.0731	.3657
	.120	11	1.288	.3789	.0485	.0863	.3578
	.156	5/32	1.614	.4755	.0572	.1017	.3469
	.188	3/16	1.881	.5522	.0630	.1121	.3380
	.219	7/32	2.119	.6228	.0676	.1202	.3296
	.250	1/4	2.336	.6872	.0711	.1265	.3217
1 3/16	.028	22	.3469	.1020	.0172	.0289	.4101
	.049	20	.5961	.1753	.0284	.0479	.4029
	.065	16	.7796	.2292	.0362	.0610	.3975
	.083	14	.9795	.2880	.0442	.0744	.3916
	.095	13	1.109	.3261	.0490	.0826	.3877
	.109	12	1.256	.3693	.0542	.0914	.3833
	.120	11	1.369	.4024	.0581	.0978	.3798
1 1/4	.035	20	.4542	.1336	.0247	.0395	.4297
	.049	18	.6285	.1849	.0334	.0534	.4250
	.058	17	.7384	.2172	.0387	.0619	.4219
	.065	16	.8226	.2420	.0426	.0682	.4196
	.095	13	1.172	.3447	.0579	.0926	.4097
	.109	12	1.328	.3907	.0642	.1027	.4052
	.120	11	1.448	.4260	.0688	.1100	.4018
	.134	10	1.597	.4698	.0742	.1187	.3974
	.156	5/32	1.823	.5369	.0819	.1310	.3906
	.188	3/16	2.132	.6259	.0910	.1457	.3814
	.219	7/32	2.411	.7087	.0984	.1575	.3727
	.250	1/4	2.670	.7854	.1043	.1669	.3644
1 5/16	.065	16	.8664	.2547	.0497	.0757	.4417
	.120	11	1.529	.4496	.0807	.1230	.4238
	.156	5/32	1.928	.5676	.0966	.1472	.4125
	.188	3/16	2.259	.6627	.1078	.1642	.4032
	.219	7/32	2.559	.7517	.1169	.1781	.3944
1 3/8	.035	20	.5009	.1473	.0331	.0481	.4739
	.049	18	.6939	.2041	.0449	.0653	.4691
	.058	17	.8158	.2400	.0521	.0758	.4661
	.065	16	.9094	.2675	.0575	.0837	.4637
	.083	14	1.145	.3369	.0706	.1027	.4577
	.095	13	1.299	.3820	.0787	.1144	.4538
	.120	11	1.608	.4731	.0940	.1367	.4457
	.156	5/32	2.031	.5983	.1129	.1642	.4344
	.188	3/16	2.383	.6995	.1263	.1838	.4250
	.219	7/32	2.704	.7946	.1375	.2000	.4160
	.250	1/4	3.004	.8836	.1467	.2134	.4075
1 7/16	.049	18	.7269	.2137	.0516	.0718	.4912
	.095	13	1.363	.4007	.0907	.1262	.4758
	.156	5/32	2.136	.6289	.1310	.1822	.4564
	.250	1/4	3.172	.9327	.1717	.2389	.4291

(TABLE IS REPEATED WITH MORE SIZES IN HO 17)