HANDOUT 1 ESTIMATING TIRE LOADS:

THREE WHEELER IN STATIC, CORNERING, BRAKING & BUMP

1. static vertical loads ("1g")

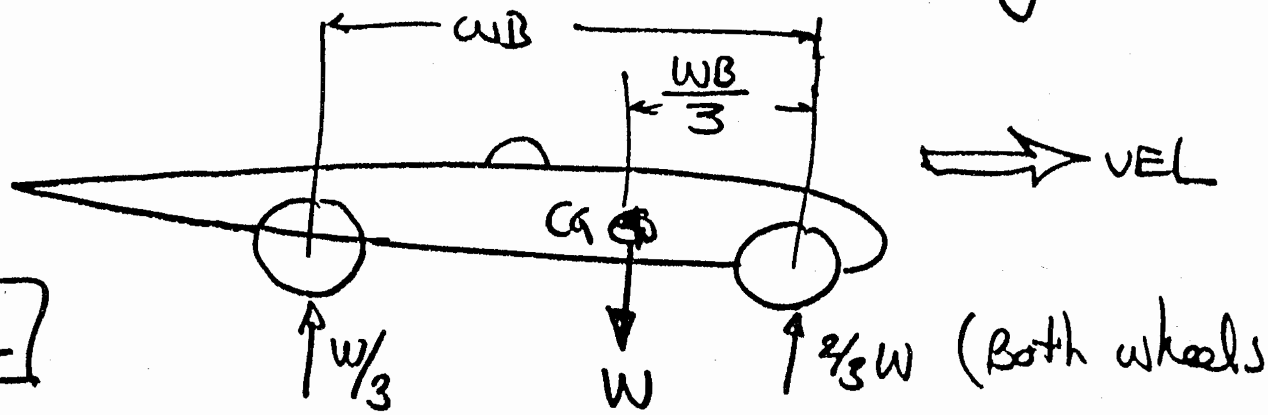


FIG 1

Assume CG at $\frac{1}{3}$ of WB from front axle, 2 wheels at front, 1 at rear,

Then "1 g bump" ⇒ each wheel supports $W/3$ (1)

We will multiply this load by some factor to accomodate Bump & shock LOADS (later) THE LAST 4 PAGES of the HANDOUT ARE from the G.M. Sunraycer Workshop Report and describe factors to use to inflate the static LOADS - WE HAVE USED THESE factors IN PAST SOLARCARS - AND NOTHING HAS FAILED--- YET!

CORNERING
• Assume the tires can produce side loads proportional to the vertical load ~ coeff of friction = μ_c.
• Assume side load is expressed as a fraction of g cornering load as $f_c * W$, where

$$f_c = V^2/gR \qquad \left\{ \begin{array}{l} V = \text{vel, ft/sec} \\ R = \text{Turn Radius, ft} \\ g = 32.2\ \text{ft/sec}^2 \end{array} \right. \qquad (2)$$

The max f_c is limited by the μ_c that the tires can generate for bike tires, $\mu_c \simeq 0.6$ (for cars, $\mu_c \simeq 0.9$)

Figure 2 shows a FBD (with i = inside and o = outside) of a vehicle in a right hand turn, with vertical and side forces at the bottoms of the tires. Consider $f_c = 1$ ("one g turn") The vehicle will slide, but pretend its constrained, (like hitting a curb). It can be shown (you do it) that it will tip when:

$$f_c \geq \frac{TR\,(WB - LG)}{2\,(WB)(HG)} \qquad (3)$$

"Tip" occurs when W_i, the weight on the inside front wheel becomes ZERO.

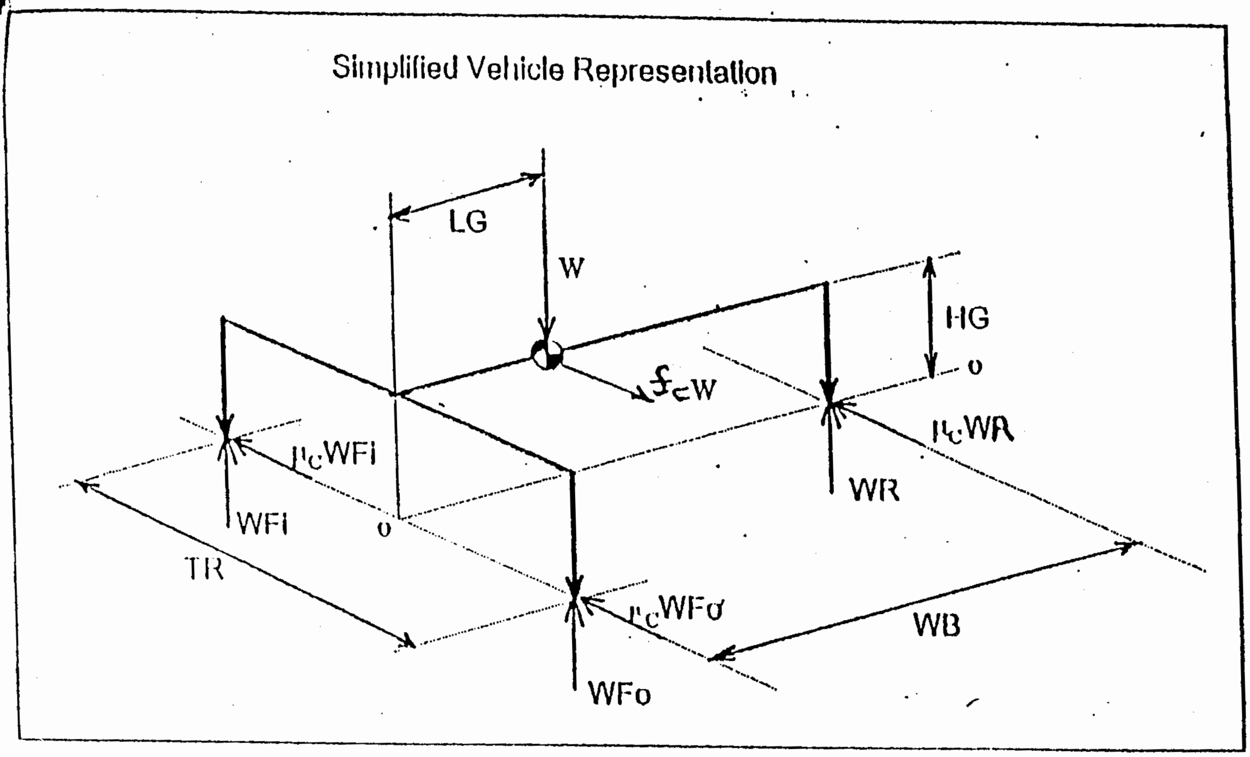


FIGURE 2 FORCE DIAGRAM FOR TIPPING ANALYSIS, RIGHT HAND TURN

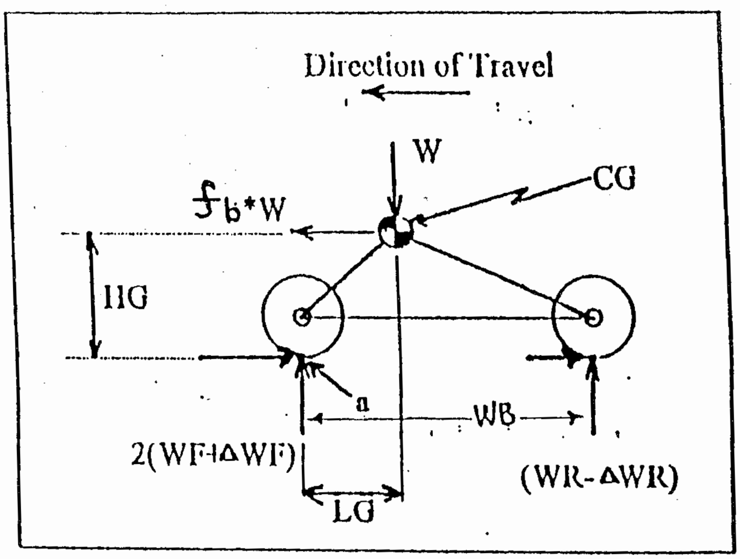


FIGURE 3 FORCE DIAGRAM FOR BRAKING WEIGHT TRANSFER

At this condition, then the vehicle weight is supported by the outside front wheel and the rear wheel:

$$W_o = \frac{2W}{3} \qquad W_R = W/3 \qquad (4)$$

$$W_i = 0$$

$$\mu_c W_o = \frac{2W}{3} \qquad \mu_c W_R = \frac{W}{3}$$

Braking Figure 3 shows a FBD of a vehicle Braking at f_B "g's". Weight ΔW_R is transferred to the front wheels, and each front wheel experiences an increase in vertical load of $\Delta W_F = \Delta W_R/2$. The weight transferred is: (you show)

$$\Delta W_R = \left(\frac{HG}{WB}\right)(f_B)(W) \qquad (5)$$

The value of f_B is limited by the coeff of tire friction in braking, μ_B, and the number of wheels braking.

With the front wheels only braking, then.

$$\max f_B = \mu_B \frac{(WB - LG)}{(WB - \mu_B HG)} \qquad (6)$$

(show this)

To examine a "worst case", consider 1 g braking: $f_B = 1$, AND pretend the front wheels can support this with the appropriate μ_B! To further make it "worst", consider all the weight from the rear being transferred to the front, so $W_R = 0$, so each front wheel supports $W/2$, AND each braking Force at the front wheels becomes $W/2$. .... (7)

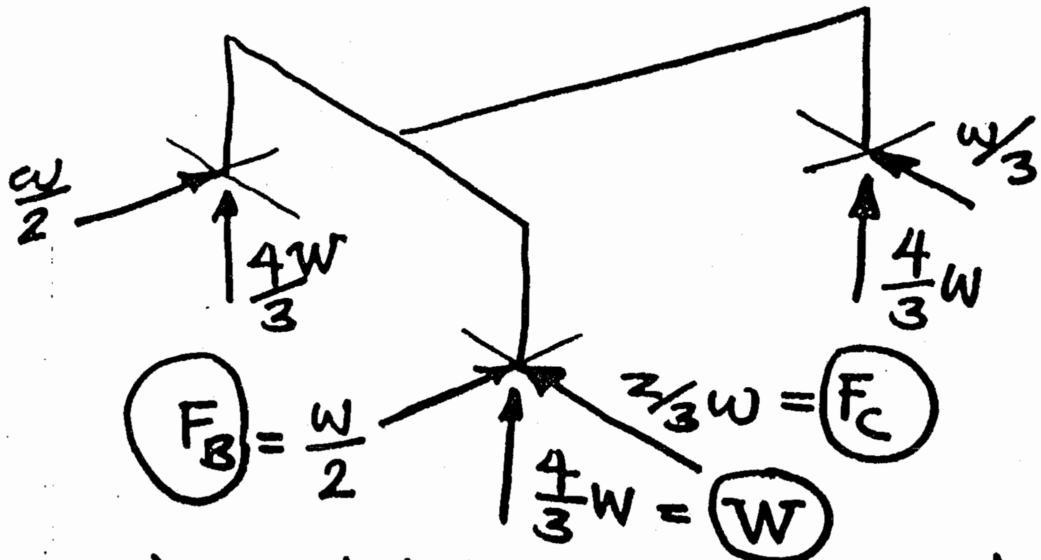
Bump - the static vertical loads are multiplied by a "factor" to create Bump loads. GM Sunraycer uses 4G Bump — so will we. So, Bump loads at the bottom of each tire are:

$$\frac{4}{3} W \qquad (8)$$

Figure 4 shows the individual loads for: 4G Bump
1G Turn (Right hand)
1G Brake

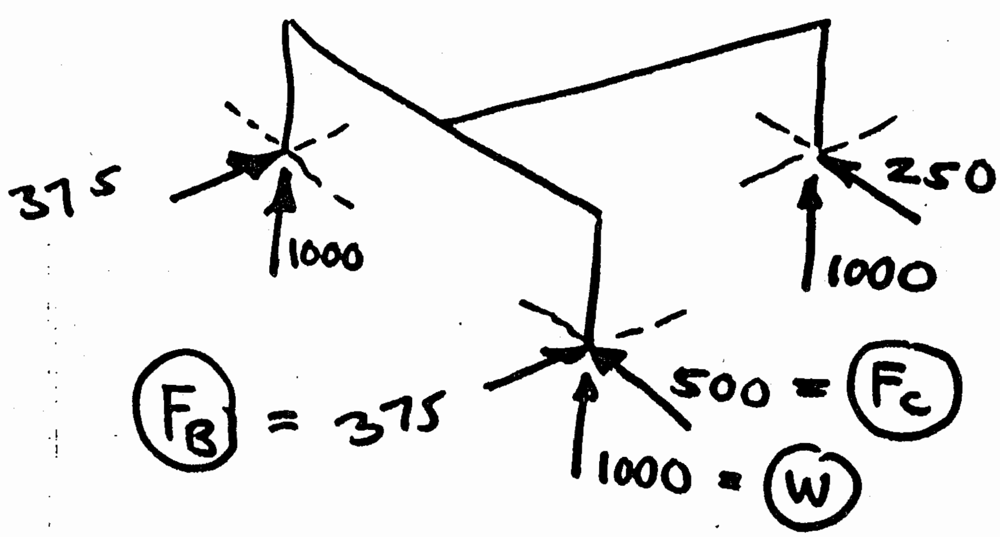
[The circled Symbols were used in last week's notes,]
SEE NEXT PAGE

FIG 4



Assume the static weight is our target of W = 750 Lbs — Then:

FIG 5



The loads are combined in order to do stress analysis and size suspension components, chassis mounts, rod ends, etc, using the following combinations.

1. 1G Bump ~ static condition: $\frac{W}{3}$ = 250 Lb, (Aurora 3)
2. 4G Bump
3. 1G Bump & Brake
4. 4G Bump & 1G Brake
5. 1G Bump & Corner
6. 4G Bump & 1G Corner
7. 1G Bump, Brake, Corner
8. 4G Bump, 1G Brake and Corner

The forces at the bottoms of the tires will be traced through the suspension members using free body Diagrams. This is illustrated in Handouts 3, 4, 11 and 17.

In Aurora II Lower A-arm rear leg, which was parallel to the axle line, had the highest loads in cases 2 and 7, and in opposite directions.

The following pages are from the Sunrayer Report Notes — SEE my comments on the last page.

(8)

LECTURE 6-1: STRUCTURE (CHASSIS)

by Ray Morgan and Herman Drees of AeroVironment

Although the aerodynamic shape of the Sunraycer was not initially well defined, we needed to get started testing a chassis for reliability, structural integrity and fatigue because of the short time frame. As a result, we decided to keep the chassis design separate from the body rather than trying for unibody construction, in which the structural loads are carried through the body. The latter would also probably not have been a good idea for this vehicle, since the body was supporting the solar panel and we didn't want to put stresses on the cells themselves. To get things going, we began with the simplest possible design, using tubes, which would let us minimize the number of parts and, if we used aluminum tubing, save on tooling time.

Tubes are also ideal for carrying torsional loads. In an automobile like you drive down the street, torsion is not of great concern, but, for handling, a race car needs a torsion-rigid structure, so the suspension stiffness may be tuned between the front and rear. (As it turned out, in the streets of Darwin, regular passenger cars couldn't keep up with the Sunraycer, because they didn't have the handling qualities it did.)

The equation for torsional stiffness alone suggested the chassis should consist of a tube with as big a radius as could fit into the body. The torsional stiffness of a tube is equal to the product of G, the shear modulus (a property of the material) and J, the polar moment of inertia, which for a tube is approximately $2\pi r^3 t$, where r is the radius and t the wall thickness. The mass, m, of the tube is $2\pi r t \rho$, where ρ is the material's density. The stiffness per unit mass is thus

$$\frac{GJ}{m} = \frac{2\pi r^3 t G}{2\pi r t \rho} = r^2 \frac{G}{\rho}$$

G/ρ is the specific shear modulus of the material. Therefore for maximum stiffness and minimum weight, the solution is to increase the radius. However, the bigger the diameter, the thinner the wall, so that eventually the tube would not support its own weight unless it were inflated. The way around this "minimum gauge" problem is the space frame structure, in which many small tubes are brought to nodes, simulating a structural shell. After a few design iterations on paper, we did some modeling using finite element analysis to determine the nodes.

Then we started looking at the buckling allowances. If one pushes on the end of a tube it can fail in one of two ways: either the wall gets crushed (as would happen with a cardboard tube) or the tube bows out to one side. In the latter case, if the force continued, the tube would break. That's called the Euler column buckling strength, which obeys the following equation:

$$\text{allowable load on a column} = \pi^2 E\, I\, C/\text{length}^2$$

(9)

where E is the Young's modulus, I is the moment of inertia ($I = \pi r^3 T$ for thin-walled tubes, where T is the wall thickness and r the radius). C is the end fixity coefficient, which is determined by how the ends are constrained. In a welded frame, C is about 1.5.

In the Sunraycer, all of the tubes except the one in the driver's seat and the four long tubes framing it were buckling critical. Those were designed for bending loads instead, because we intended to put the steering wheel column and pedals on the former and we didn't want to the latter to collapse if people leaned on them when getting into the car. Because aluminum has poor fatigue strength allowance, we wanted to keep the working stresses fairly low, below 10,000 psi, even though the aluminum might be good to 70,000 psi. Although the Sunraycer was designed to the limit loads for a race car, the structure is such that the stresses never get above about 8,000 psi, which allows about 10^7 cycles in fatigue. The car's entire chassis weighs 15 lb. (The whole car, with the driver, grosses 575 lb.) We could have optimized the tubing size, using tubing about two inches in diameter and about 0.010 wall thickness (like a beer can), but that would be difficult to weld, and if anyone leaned on a tube, it would crumple. If everything were optimized for driving loads only, the frame would have weighed under 10 lb.

The material we used, 6061-T6 aluminum, was fairly easy to weld. We considered heat treating the whole frame after welding it together, but samples sent to the heat treater came out warped, so we didn't want to take the risk. Welding anneals the material; the stress allowable for the annealed state is about 12,000 psi, but hardens again after aging. Actually, during welding, the long sidemembers warped due to the heat and took a bow. We were a little concerned that, in a collision, they would buckle from compressive loads, since they were already bowed, so we added some thin tubes to stabilize them at their centers.

The design loads were a compromise between those for a race car and those for a standard passenger vehicle. Assuming the full gross weight of the vehicle, these loads were as follows:

4 G Bump	i.e., the acceleration of gravity was pushing up on all four wheels
2 G Twist	i.e., with all the vehicle's mass supported on two diagonally opposed wheels, the frame could be twisted to twice the force of gravity
1 G Corner	i.e., coefficient of friction of 1 for the tires, with full load transfer to the outside wheels
1 G Braking	i.e., coefficient of friction of 1 for the tires, with full load transfer to the front wheels

The options for suspension included swing axles, double A-arms, and MacPherson struts. Swing axles are the simplest, but they don't keep the wheels perpendicular to the street, as we desired for optimum handling. The double A-arm does keep the wheel perpendicular, but has more parts than the MacPherson strut, which we chose to use for the front. For the rear suspension, we used a trailing arm. The MacPherson struts are made of chromoly steel, the same kind of steel good bicycles are made of, heat treated after welding. Although they take enormous side loads, the A-arms that hold them to the body are placed where the moment is largest, so that they carry the loads fairly easily. Initially, we made the struts of aluminum, but the first time we tested a skidding brake, they bent, so we rebuilt them of steel. The member that slides up and down the suspension is made of tool steel.

The A-frames that attach the MacPherson struts to the car are also used for steering. We placed the steering axis of each front wheel in front of the tire path, so that when we let go of the steering wheel, the car's motion always returns toward a straight line, just like in standard automobiles. In the rear, a long steering link accommodates roll steer, i.e., the tendency of the wheels to turn in the direction the car leans during a turn. We set the roll steer at the level with which our professional race driver was comfortable, giving the roll steer of the wheels at about one-tenth of the degree per degree of roll of the car. To avoid excessive steering input forces, the steering axis is in line with the tire path. If it had been offset, a moment would be generated on the steering linkage, making it more and more difficult to turn.

To drive the rear wheel, we used a big toothed pulley driven by a cog belt and a small pulley. It has about a four-to-one step-up. Initially, we used a chain, which was very efficient, but switched to a cog belt because it didn't need grease and wouldn't pick up dirt. The motor is stationary on the frame, but the suspension moves up and down, so the apparent length of the drive shaft changes. We allowed it to slide back and forth through the bearing block, which was probably fairly risky, but we put a lot of grease into it and did not have any wear problem.

After the car was assembled and had been driven to get the bugs out, Hughes aircraft put about 20 strain gauge bridges on the suspension pieces and 6 three-axis accelerometers on the body. The recorder and data processing system were portable and ran off the car's battery. We wanted to find out the actual G loads. We had assumed that the greatest load would be 4 g and had selected 6 G as a safety factor. During testing on real roads and over simulated cattle grates made of wood, we found spikes in the frame to as much as 20 G, but these were of short duration and thus not of concern. We were also worried that the cattle grates might result in resonance, but they did not. After 1000 miles of testing at the GM proving grounds near Phoenix, we applied a penetrant in critical areas of the tubes, but found no cracks.

During the race, we made a thorough inspection of the chassis and drive components for loose hardware and cracks in the frame after each race day. None were found, except for a small crack in a tube, on which the power relays were mounted. This was discovered on Day 4 of the race and although we did not think that the location of the crack was critical, we reinforced the tube with a splint made of aluminum just in case.

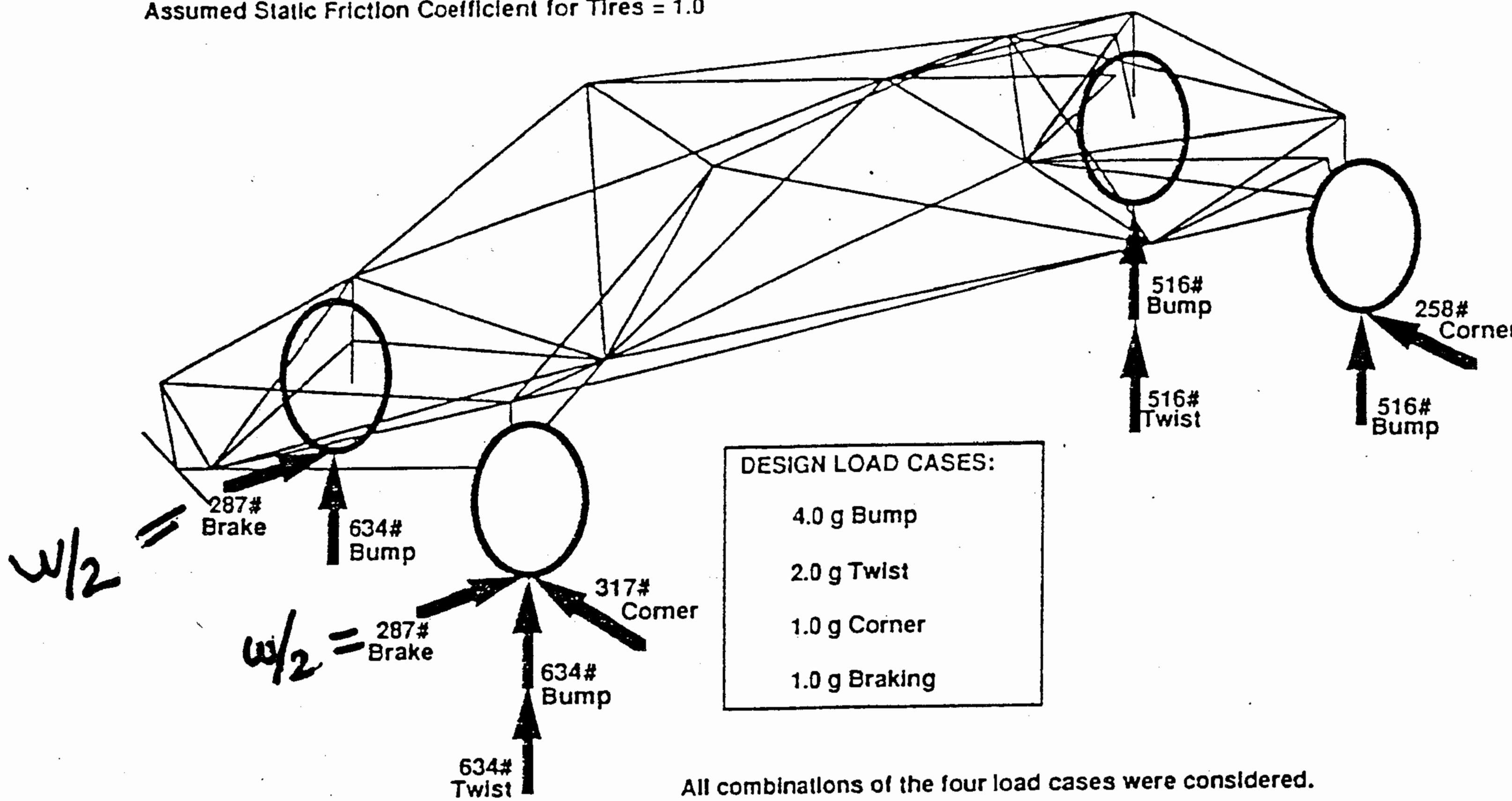
[The tubes of the chassis, which were nominally one inch in diameter and 0.035 inch in wall thickness, were welded together by John Mason of Mason Engineering of Simi Valley, California.]

SUNRAYCER CHASSIS DESIGN LOADS

Note $W_R = (.45)(575) = 258$ Lb

$W_F = (.55)(575) = 317$ Lb

W = Front to Back Wheel Weight Distribution: 55% - 45%.
Car Gross Weight = 575#
Assumed Static Friction Coefficient for Tires = 1.0



DESIGN LOAD CASES:
4.0 g Bump
2.0 g Twist
1.0 g Corner
1.0 g Braking

All combinations of the four load cases were considered.

Worst Case = 4.0 g Bump+1.0 g Corner+1.0 g Braking

Note
- only outside tires loaded in turning
- only front wheels doing Braking
- No interaction between *changes in vertical loads caused by cornering and braking ⇒ always 4G in Bump

11

VG 6