Triumph TR3 & 4 Suspension Geometry

by

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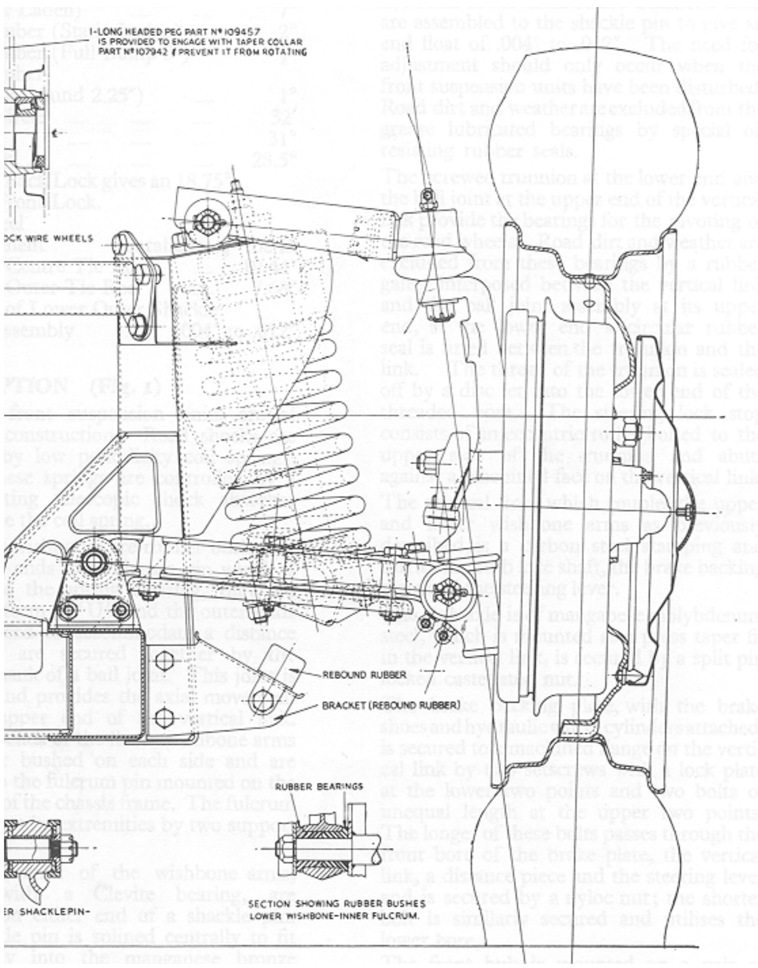
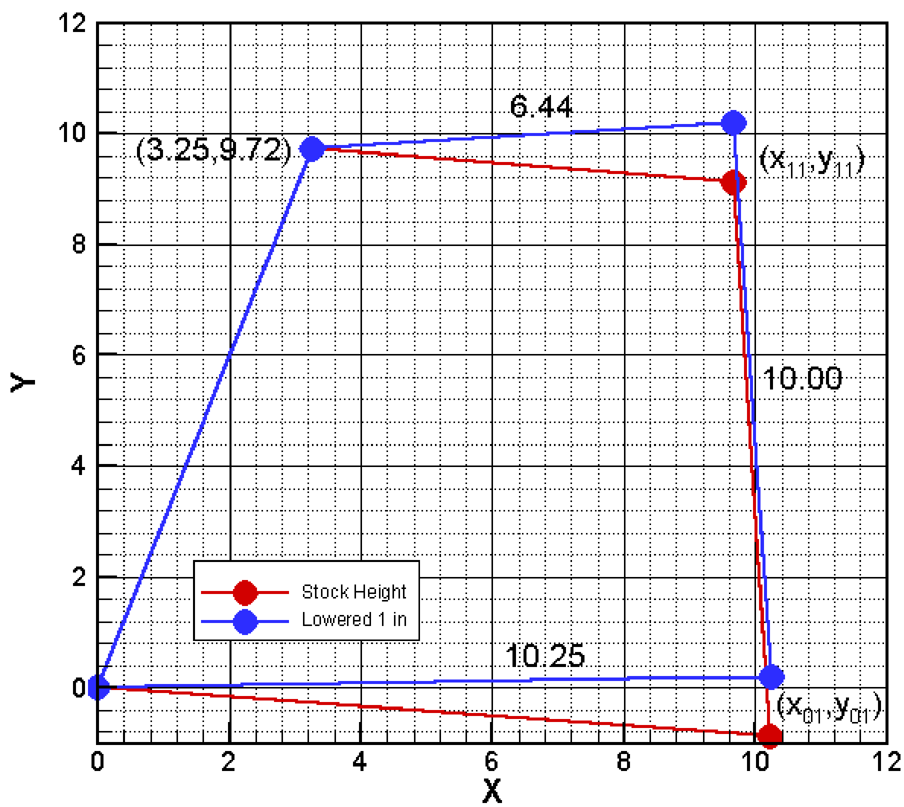


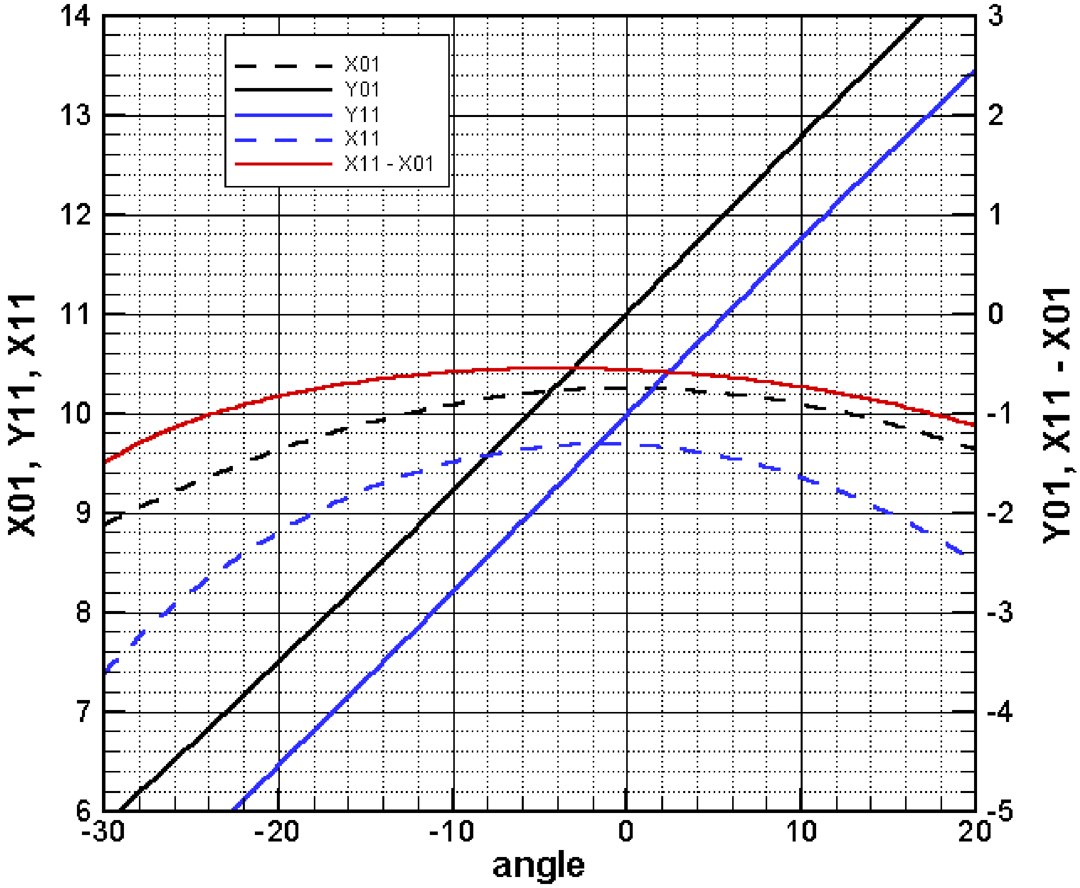
Above is a schematic of the Triumph TR3/4 suspension system taken from the Service Instruction Manual. The figure below is a simplified picture of the geometry at static conditions for a car with stock ride height and for one which has been lowered one inch. We have placed the origin at the inner lower pivot. The dimensions on the graph were determined by direct measurements and by scaling measurements from the schematic above. The relationships between the points on the graph can be determined from the following simple geometric relationships:

$$x_{01} = a_0 \cos\theta$$
$$y_{01} = a_0 \sin\theta$$
$$y_{11} = y_{01} + \sqrt{b_1^2 - (x_{01} - x_{11})^2}$$
$$x_{11} = x_{10} + \sqrt{a_1^2 - (y_{11} - y_{10})^2}$$

where: θ is the angle of the lower A arm
a_0 is the length of the lower A arm
b_1 is the length of the vertical link
a_1 is the length of the upper A arm
x_{01} and y_{01} are the coordinates of the upper inner pivot



The graph below was calculated using these relationships for various positions of the suspension arms. The Front Suspension Geometry Pro software requires the geometry to be specified by the coordinates of all the joints and pivots. This approach is cumbersome for performing "what-if" calculations, because all the joint and pivot locations are interrelated by the relationships above.



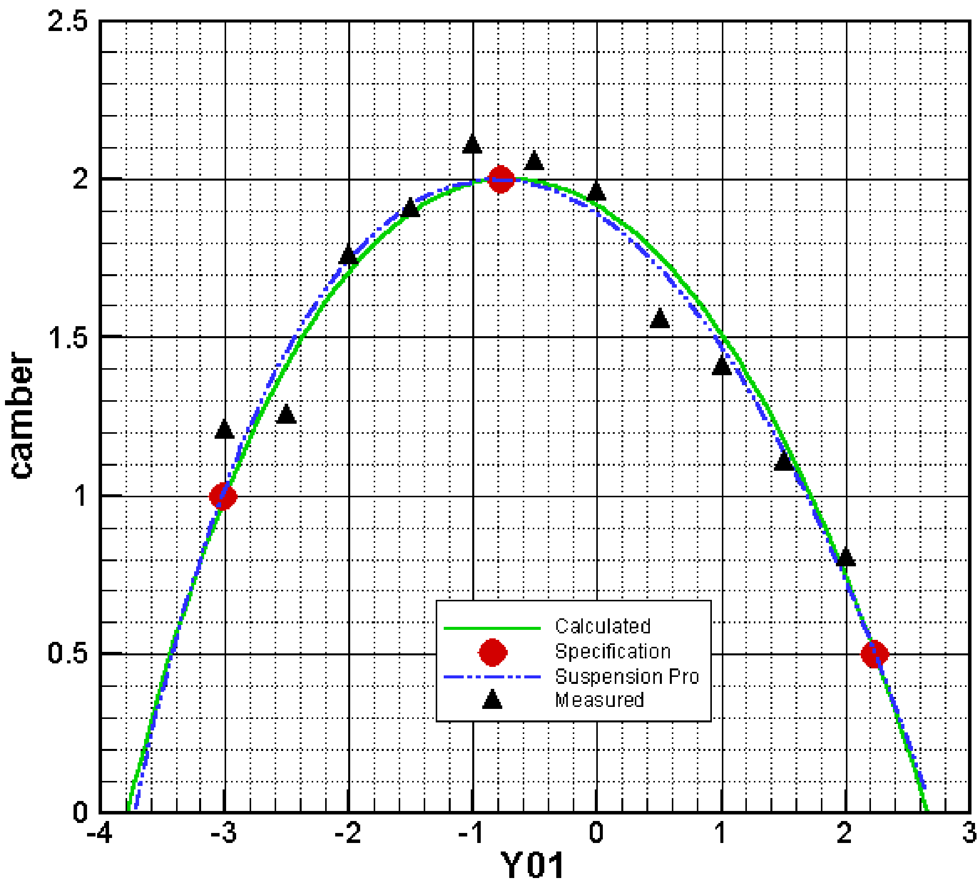
Stock Camber

One of the important aspects of the suspension geometry is the relationship between camber and suspension position. The tangent of the camber angle is proportional to the difference between x_{01} and x_{11} divided by the length of the vertical link. For a stock suspension system, the camber is accurately approximated by:

$$Camber = 5.7388(x_{11} - x_{01}) + 5.1623$$

The Service Instruction Manual lists the camber at three ride heights. To verify that we have determined the correct dimensions, the camber was calculated using a spreadsheet and the relationships above, and with the Front Suspension Geometry Pro software. As a further check, the camber change was directly measured on a TR3. The graph below compares all four sets of values. The values shown on the plot were shifted to achieve the agreement shown. These adjustments are described in the following.

First, the calculations cannot give the camber at static conditions; they give only camber changes with suspension travel. For this reason, the calculated curves were shifted to match the specification of 2 degrees of camber at stock ride height. Second, the calculations made with the Front Suspension Geometry Pro (FSGP) software differ slightly from the spreadsheet results. The agreement in the graph was achieved by shifting the results from the FSGP by 0.09 inches. The value of y_{01} for a stock setup at static conditions is not known precisely. By matching up the results, FSGP indicates that y_{01} is about -0.86. The spreadsheet calculations give a value of about -0.77, while a measurement from the schematic gives a value of about -0.89. Also, the shapes of the calculated curves are slightly different. Although these differences are small, I would like to know the cause of these discrepancies.



The triangles in the figure were measured on a TR3 with suspension that was stock except for springs which lowered the car about 0.3 to 0.4 inch. Camber measurements were made using a digital level or digital protractor.

The measurements are probably accurate to within about 0.2 to 0.3 degrees. Initially, the static camber was determined to be +0.3 degrees on the left side and +1.3 degrees on the right side. The difference between the left and right side measurements indicates the spring towers could be askew. The camber curve was measured on the left side. During this procedure a camber of +1.6 degrees was measured at static ride height. The difference between these left side measurements was apparently due to the weight of the car and play in the suspension system. To further confuse the picture, the static camber measurements became +2.1 and +1.2 degrees on left and right after replacing the suspension bushings and trunnions. In the figure, the values plotted are 0.4 degrees greater than the actual measured values.

The variation of camber with suspension travel is important during corning. In order to maximize the tire patch as the body rolls in a turn, the camber needs to decrease in the bump direction (outside tire) and increase in the droop direction. The variation in camber with suspension travel is called *camber gain*. For stock suspension and stock ride height, the camber gain is zero, i.e. the camber does not change for small variations about the static ride height.

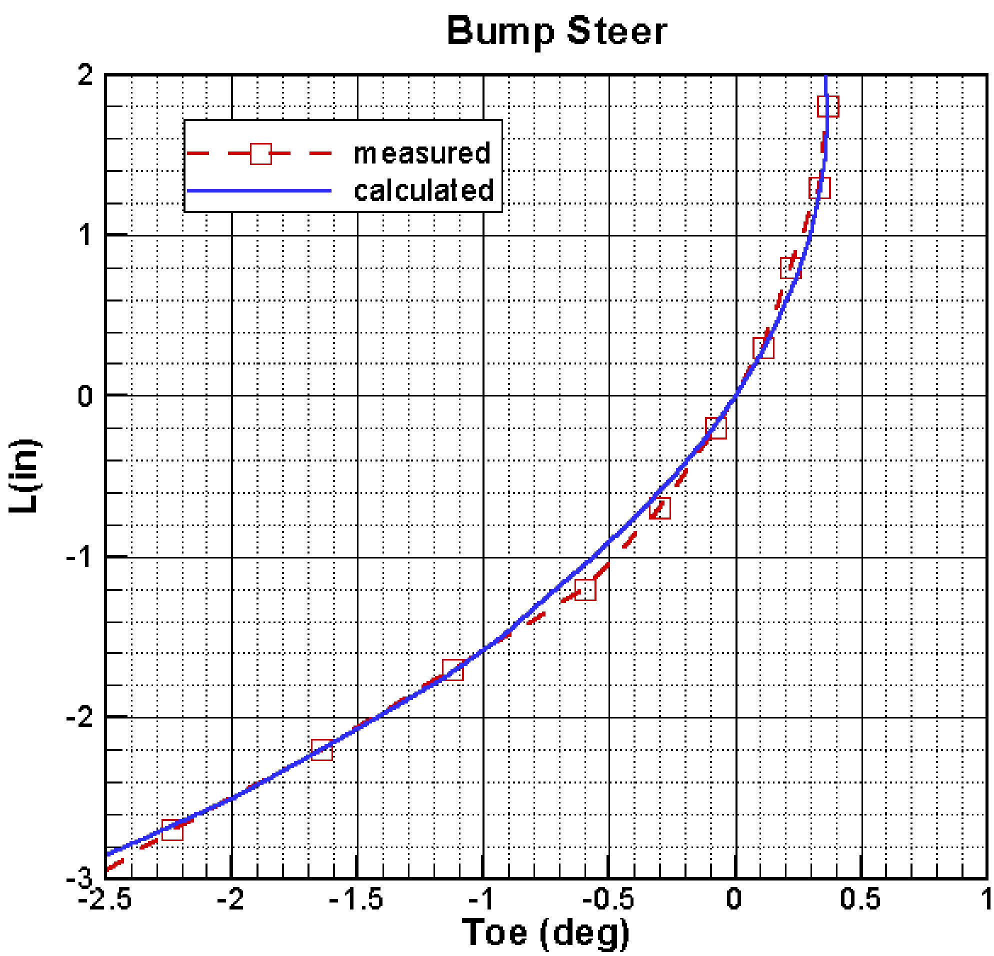
Camber Modification

Calculations were performed to investigate various suspension modifications. First, lowering the car will change both the static camber and the camber gain. If the car is lowered 1.5 inches (y_{01}= 0.7), the static camber will be reduced from 2 degrees to about 1.65 degrees. Since the camber at a bump of 1 inch (y_{01}= 1.7) is now about 1.0 degrees, the camber gain is 0.65 degrees/inch.

Most Triumph racers find that a static camber of -1.5 degrees works well. The two most common modifications used are: (1) shorten the upper A arm or (2) move the upper fulcrum inward. The equation for camber can be used to calculate the change needed. For example, to achieve -1.5 degrees for a car that initially has +1.6 degrees, the upper A arm should be shortened about 0.54 inches, i.e. (3.1/5.74). Alternatively, the upper fulcrum could be moved inward this same amount. The software indicates little difference between these two alternatives. The shortened A arms produce a slightly improved suspension geometry. With shortened A arms the roll center is at a height of 0.51 in and the camber gain is 0.71 deg/in, while for moving the fulcrum the values are 0.22 in and 0.60 deg/in, respectively. These calculations were made for Hoosier 5.50x15 tires, which produce a spindle height of 11 15/16 in.

Bump Steer

The suspension software performs calculations of bump steer. However, it is very difficult to measure the locations of the steering tie rod ball joints accurately enough to predict the bump steer effects. For this reason, the bump steer was measured directly. The uncertain dimensions were varied in the software until the calculations matched the measured data. The results of this exercise are shown in the graph. The vertical axis is the suspension travel, positive for bump and negative for droop. The vertical travel of zero was the static conditions for the subject TR3A, which had only been lowered approximately 0.3 inches. The bump steer is severe. For example, if the nose of the car drops 1 inch during hard braking, the toe out will increase almost 3/8 inch. The measurements are consistent with



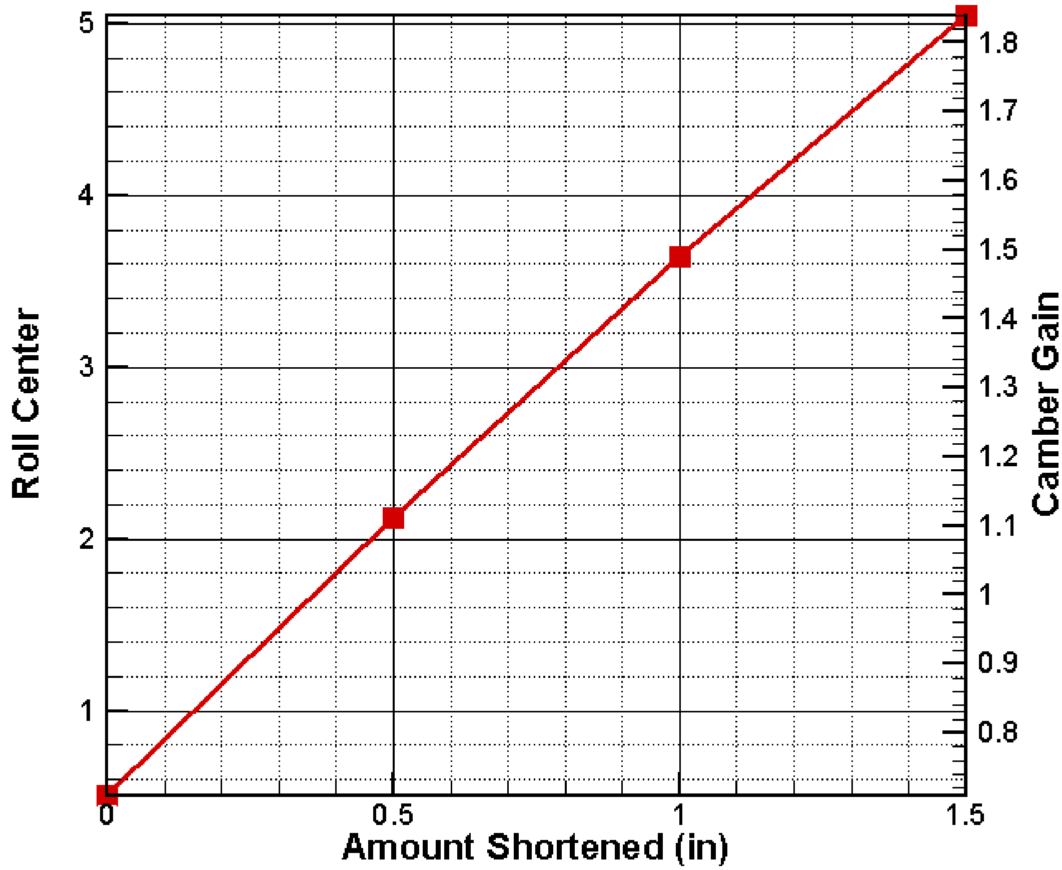
the observation that the car was unstable under hard braking. The software was then used to determine how to eliminate this problem. Bending the steering lever up 1/4 inch is sufficient to eliminate the bump steer problem. Lowering the car would also improve bump steer, because the toe variation is reduced in the bump direction.

The bump steer curve was remeasured after shortening the upper A arms, replacing the trunnions and suspension bushings, and lowering the car an additional 0.9 inches. Fortunately, the problem had been almost completely eliminated by these other changes.

Other Modifications

The software was also used to investigate other modifications to the suspension geometry. Two important effects of the suspension geometry are the roll center height and the camber gain. A higher roll center will reduce the amount of roll observed in a turn. The camber gain is important, because when the car rolls in a turn, we would like the tires to remain nearly vertical. The FSGP software recommends a camber gain of 1.8 deg/in, i.e. the camber increases 1.8 degrees when the suspension arm moves down (droop) one inch or the camber will decrease at that rate as the suspension arm moves up (bump). As indicated above, a stock TR suspension system is very poor with respect to these variables. Most TR racers prefer a large amount of static negative camber and large anti-roll bars for high roll stiffness.

For a car, which has been lowered 1 1/2 inch and has shorten upper A arms, the graph below shows the effect of lowering the upper A arm fulcrum. With no modification, the roll center is only about 1/2 inch above the ground. Lowering the fulcrum increases the roll center height significantly. Although FSGP recommends a camber gain of 1.8 deg/in the optimum value probably depends on the amount of roll stiffness. Lowering the fulcrum 1 1/2 inch approaches this value. This modification could be achieved by cutting down the spring tower. Of course, the spring would also have to be modified to avoid changes in the ride height, and the steering would also have to be modified to avoid excessive bump steer. If the tower were shortened 1 inch, the steering lever would have to be bent upward or the steering arm would have to be lowered about 5/8 inch. There could be other implications of this modification that I am not aware of.



Roll Center
5
4
3
2
1
Camber Gain
1.8
1.7
1.6
1.5
1.4
1.3
1.2
1.1
1
0.9
0.8
0
0.5
1
1.5
Amount Shortened (in)