Service

Article #26

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"Austin 1800 Utility Torsion Bars"

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Introduction

One of the more unusual features of the Australian-designed Austin 1800 utility is the torsion bar arrangement at the rear suspension.

Because the vehicle is designed to carry a payload, the rear displacers were increased in size to match those at the front – unlike the saloon which used smaller diameter displacers at the rear.

In addition to larger diameter hydrolastic units, a torsion bar was also fitted to each of the rear suspension arms. Although we might at first think that the torsion bars serve to assist the hydrolastic springs in carrying the payload, an analysis of the way in which they are set reveals that much more is going on.

In this article, the action of the torsion bars and the way in which they are set is described in detail so as to reveal their true function.

1. Setting Procedure

The figure below shows the general arrangement and factory instructions for torsion bar initial setting.

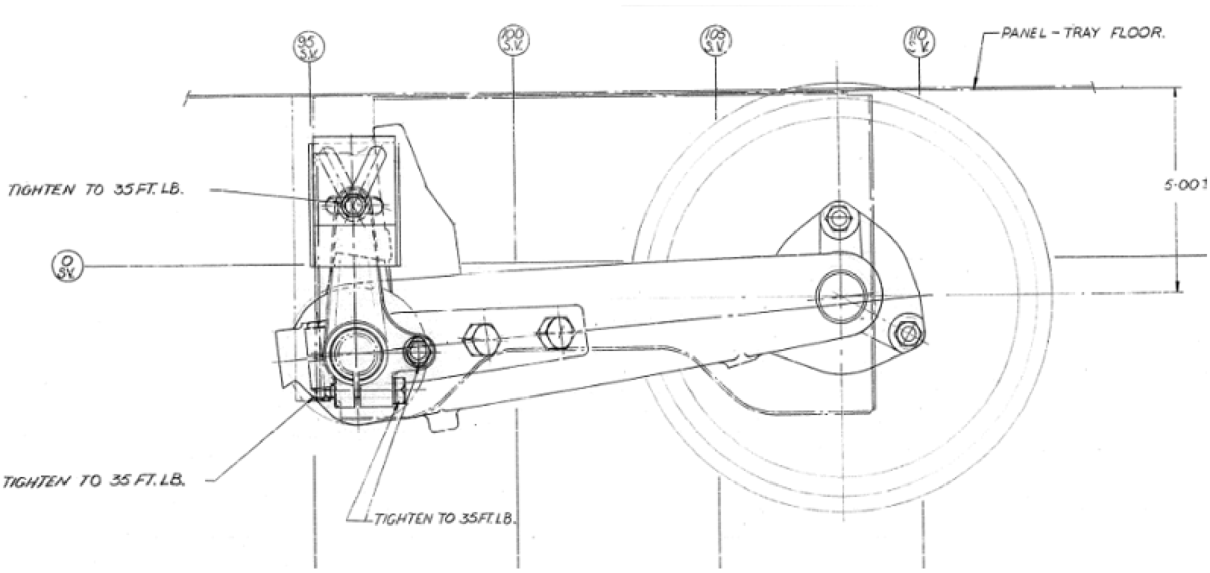
With the hydrolastic pressure released, the rear radius arm is positioned so as to give the clearance shown 5" from the centre of the wheel bearing to the load floor. This is the torsion bar neutral position. The torsion bar brackets are then adjusted for no torsion (unstressed) and locked into position.

The hydrolastic system is then pressurized in the normal manner.

But, when the hydrolastic suspension is pressurised, the clearance above is considerably greater than 5" – meaning that at a zero load condition, the hydrolastic rubber spring is being compressed by the torsion bar. This is a pre-stress that is applied to the rubber spring before any vehicle load or payload is applied.

One would have thought that the purpose of the torsion bar was to assist in taking the load, but instead, as far as the rubber spring is concerned, the torsion bar prestress adds to the load it had to bear - well, up to a certain point at least. Eventually, with increasing load, the torsion bar will be twisted back to its neutral position, and from then on, it will assist the rubber spring in taking the load.

One might ask the question why is this so?



TORSION BAR SETTING INSTRUCTIONS.
INITIAL PRODUCTION AND RECTIFICATION.

1. WITH THE HYDROLASTIC SUSPENSION SYSTEM DEPRESSURISED, SUPPORT THE RADIUS ARMS AT THE HEIGHT SHOWN ABOVE, WHICH IS THE NEUTRAL POSITION OF THE TORSION BAR. (I.E. THE TORSION BAR IS UNSTRESSED.)
2. FIT VERNIER BRACKET TO TORSION BAR SPLINES IN A SUITABLE ANGULAR POSITION, WHICH WILL ALLOW FITMENT OF THE CLAMPING BOLTS. (VERNIER BRACKET TO BODY).
3. FIT LOCKING PLATES, ALL BOLTS ETC. AND TIGHTEN. NOTE THAT LOCKING PLATE CAN BE TURNED OVER AND/OR MOVED UP AND DOWN TO ACCOMMODATE ANY POSITION OF THE TOP VERNIER BRACKET BOLT.

SERVICE.

1. WITH THE TORSION BAR VERNIER BRACKET REMOVED, LOAD THE VEHICLE AT THE REAR OR ADJUST HYDROLASTIC PRESSURES (WHICHEVER IS APPLICABLE) TO POSITION THE RADIUS ARMS AS SHOWN. THIS IS THE NEUTRAL POSITION FOR THE TORSION BAR.
2. AS FOR INITIAL PRODUCTION.
3. AS FOR INITIAL PRODUCTION.

Fig. 1 Setting procedure for torsion bar.

2. Analysis

2.1 Linear Spring

To understand the action of the torsion bar, we might model two cases. The first being the action of a torsion bar and a rubber spring both with a linear characteristic. In fact, the rubber spring has a non-linear response and we will get to that case shortly, but let's start with a simple case of two linear springs, interconnected, one acting in compression and the other pre-tensioned as shown in the figure below.

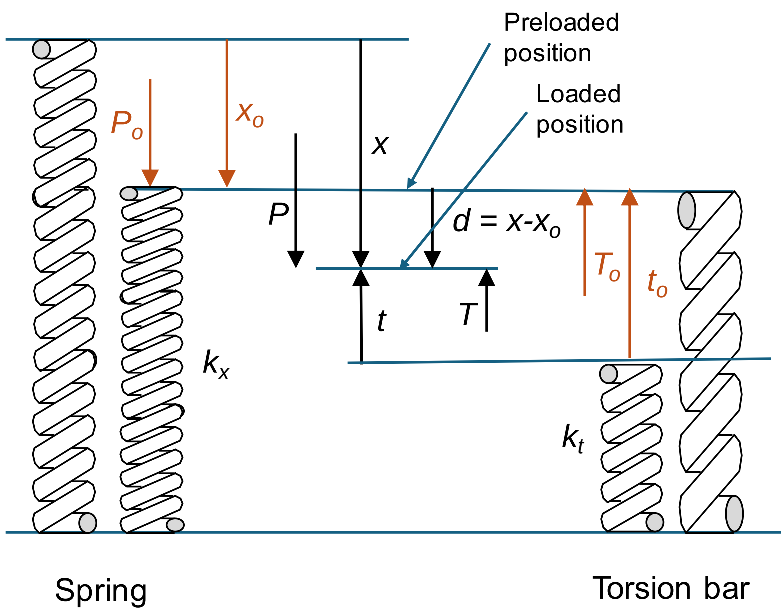


Fig. 2. Two springs in parallel, one the "rubber" compression spring, the other, the torsion bar.

For convenience, we will consider the weight of the vehicle as the "payload" *P* such that at zero load, the only force acting on the rubber spring (hereafter called the "spring") is the pre-tension *T* of the torsion bar. We will then examine what happens when load is applied (and this load could be the weight of the vehicle, plus any additional load that might be carried in the tray).

In the analysis to follow, "up" is taken as the positive direction, and "down" as the negative direction. The deflection of the spring is taken from the position at its free length.

As per the instructions shown in Part 1, the operator effectively sets an initial preload T_o.

Since in this case, both springs are linear springs, then the following equations apply – where via Hooke's Law, the load is linearly proportional to the displacement *x* of the end of the spring via spring constants k_s and for the displacement *t* of the torsion bar, k_t.

$$P_o = k_s x_o$$

$$T_o = k_t t_o$$

Since at this condition, the only load applied to the spring P_o is supplied by the "tension" T_o of the torsion bar such that:

$$P_o = -T_o$$

Thus,

$$x_o = -\frac{k_t}{k_s} t_o$$

At some payload *P*, the total force *F* now acting on the spring is the sum of the *P* and that from the torsion bar *T*.

$$F = P - T$$

$$= k_s x$$

And

$$T = k_t t$$

where

$$t = t_o + (x - x_o)$$

And so,

$$k_s x = P - k_t\big(t_o + (x - x_o)\big)$$

Collecting terms and making *x* the subject of the formula, we have the deflection of the spring as:

$$x = \frac{P - k_t t_o + k_t x_o}{k_s + k_t}$$

From the initial condition (pre-stress), the load point displacement for the payload P is thus:

$$d = x - x_o$$

Fig. 3 below shows the characteristics of the spring response for different scenarios. The numbers shown are arbitrary, but the vertical axis represents force, and the horizontal axis, displacement of the free end of the spring.

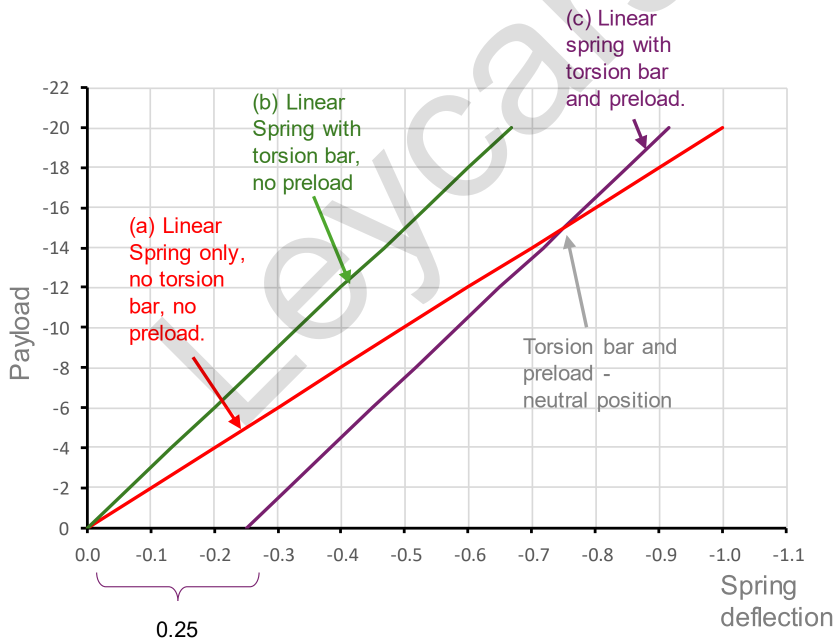


Fig. 3. Spring response for (a) Linear Spring, no Torsion Bar and no preload, (b) Linear Spring with Torsion Bar – no preload, (c) Linear Spring with Torsion Bar and preload.

As would be expected, (a) and (b) show the increase in stiffness of the combination when the torsion bar is included. The existence of a preload serves to shift the response to the right, but the slope of the response is the same as before.

Note that as *P* increases, at some corresponding displacement, the torsion bar reaches its neutral position and at that point, the load applied to the spring is the same at which would occur for the spring alone, the torsion bar contributing nothing to the load applied to the spring.

The conclusion here is that although the torsion bar stiffens the suspension, the preload does not make any difference and only serves to lower the height of the vehicle a constant amount.

2.1 Non-Linear Spring

We now come to the more interesting case of a non-linear spring. This is more realistic for the hydrolastic suspension system used in the vehicle since the rubber, and the shape of the cone at the end of the pushrod, effectively stiffens the response as a function of displacement. That is, the spring "constant" is a function of dsplacement. We might represent this as a square law such that:

$$P = k_s x^2$$

The torsion bar remains as a linear spring.

At the initial preload state, then:

$$P_o = k_s x_o^2$$

$$T_o = k_t t_o$$

As before, the only load applied to the spring P_o is supplied by the "tension" T_o of the torsion bar such that:

$$P_o = -T_o$$

Thus,

$$x_o = -\sqrt{\frac{k_t}{k_s} t_o}$$

At some payload *P*, the total force *F* now acting on the spring is the sum of the *P* and that from the torsion bar *T*.

$$F = P - T$$

$$= -k_s x^2$$

The minus sign is necessary here because the downwards direction (*x*) is negative as is the force *F*.

Also,

$$T = k_t t$$

Where as before:

$$t = t_o + (x - x_o)$$

And so,

$$k_s x^2 = P - k_t\big(t_o + (x - x_o)\big)$$

Collecting terms and making *x* the subject of the formula, we can solve the resulting quadratic equation to obtain:

$$x = \frac{-k_t - \sqrt{k_t^2 - 4(-k_s)x_o(-k_t t_o + k_t x_o)}}{-2k_s}$$

In this equation, a minus sign in put before k_s to account for the negative direction of both x and F as said above.

From the initial condition (pre-stress), the load point displacement for the payload P is as before:

$$d = x - x_o$$

Fig. 4 below shows the characteristics of the spring response for different scenarios including a comparison with the case of a linear spring (Case (c) as before).

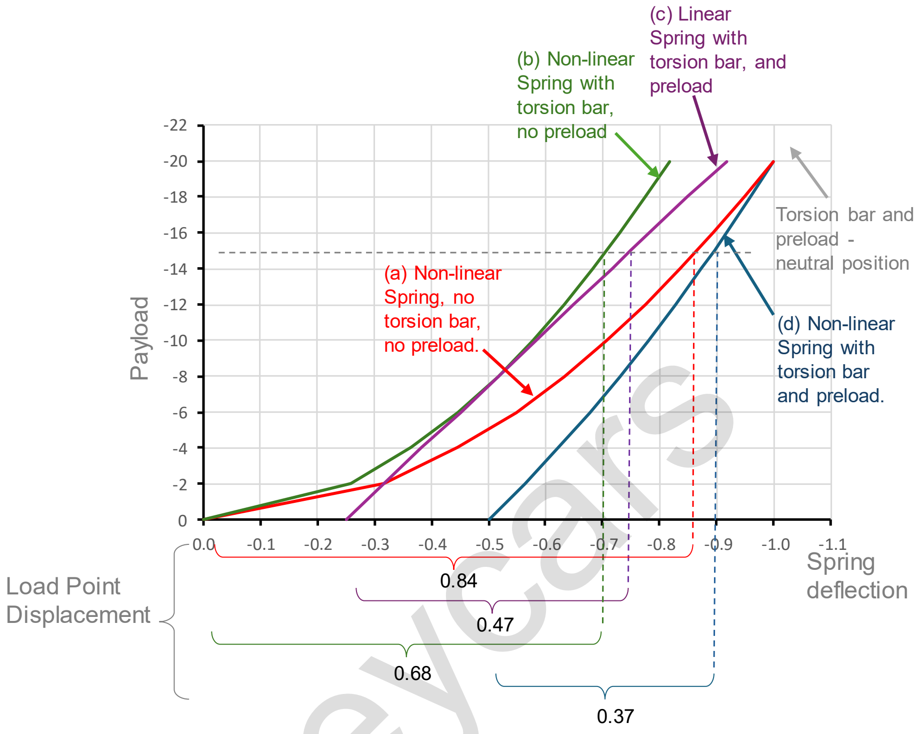


Fig. 4. Spring response for (a) Non-Linear Spring, no Torsion Bar and no preload, (b) Non-Linear Spring with Torsion Bar – no preload, (c) Linear Spring with Torsion Bar and preload (d) Non-Linear spring with torsion bar and preload.

Also shown in Fig. 4 is the load point displacement for each scenario for a selected arbitrary load.

As can be seen from Fig. 4, for a given arbitrary load of 14 force units, the load point displacement is the least for the case of a non-linear spring with a preloaded torsion bar. This would indicate that the overall stiffness of the spring-torsion bar system is stiffer compared to the other cases.

3. Discussion and Conclusion

In Part 2.1, in the case of a linear spring and torsion bar, the presence of an initial preloading made no difference to the stiffness of the spring combination. In Part 2.2, in the case of a non-linear spring and torsion bar, the presence of a preload increases the stiffness of the spring system as evidenced by the lower value of load point displacement for a given load.

This occurs because the initial preload point of the spring system is set further up into the stiffer part of the rubber spring non-linear response.

The actual shape of the rubber spring characteristic is not modelled since this information is not available, and so a simple square law is treated for the sake of illustrating the effect.

The purpose of setting a preload is to take advantage of the non-linearity of the rubber spring in the hydrolastic unit that reducing the change in attitude of the vehicle between the loaded and unloaded conditions. This is necessary in the case of an Austin 1800 Utility with interconnected hydrolastic suspension since the headlight aim would otherwise be unacceptable – a circumstance already in evidence in the saloon (with smaller rear displacers) when carrying rear seat passengers and luggage.

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May 2024