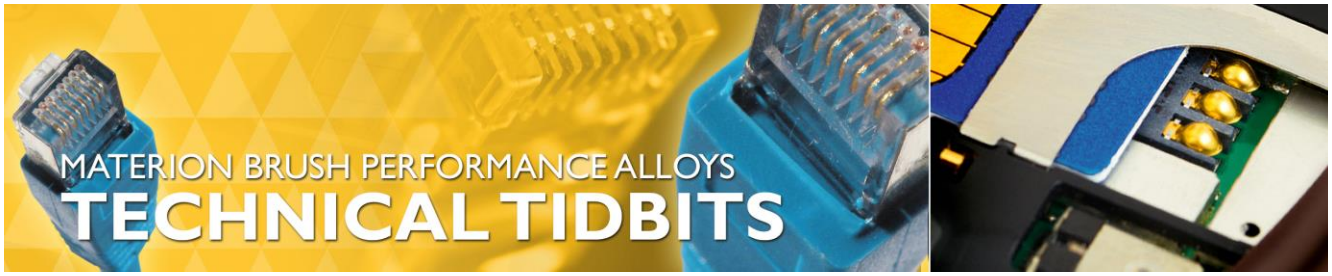

MATERION BRUSH PERFORMANCE ALLOYS
TECHNICAL TIDBITS

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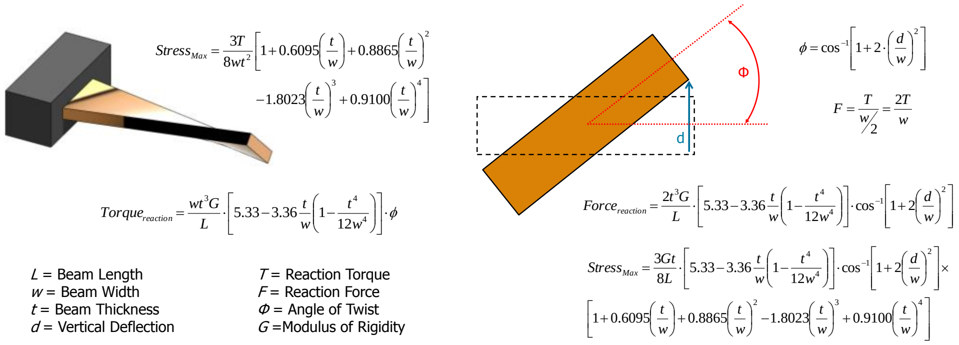
Do the Twist! – Contact springs that are functionally equivalent to the torsion bar.

- **Torsion Bar**
- **Louvered Contact**
- **Modulus of Rigidity/ Torsional Modulus/ Shear Modulus (*G*)**
- **Hertz Stress**

Spring Types Part 3 - Torsion Bars

This series of six or so editions of Technical Tidbits will discuss various types of springs used in electrical contacts or sensors, and group them into six broad categories of similar function (cantilever beams, simply supported beams, torsion bars, Belleville washers, coil springs, and bellows & diaphragms). This month we will focus on the torsion bar.

The last two months explored cantilever and simply supported beam spring types. Both had similar equations governing their stress-deflection and force-deflection behavior. Both of these beams are loaded in bending. Next we will look at a spring type that is loaded by twisting along the axis, known as the **torsion bar**. The left side of figure 1 shows a bar loaded in pure torsion, and the relevant equations for maximum stress and contact force as a function of angle of twist (Φ) and applied torque (T). In electrical contacts, torsion bars usually take the form of **louvered contacts** in the female (socket side), which start in the twisted state, and are deflected along one edge by the male blade into a less twisted state. The equations in the upper right of Figure 1 show how to obtain reaction force from the torque and angle of twist from the applied deflection, so you can calculate reaction force and stress from deflection.



$$Stress_{Max} = \frac{3T}{8wt^2}\left[1 + 0.6095\left(\frac{t}{w}\right) + 0.8865\left(\frac{t}{w}\right)^2 - 1.8023\left(\frac{t}{w}\right)^3 + 0.9100\left(\frac{t}{w}\right)^4\right]$$

$$Torque_{reaction} = \frac{wt^3G}{L}\cdot\left[5.33 - 3.36\frac{t}{w}\left(1 - \frac{t^4}{12w^4}\right)\right]\cdot\phi$$

L = Beam Length
w = Beam Width
t = Beam Thickness
d = Vertical Deflection

T = Reaction Torque
F = Reaction Force
Φ = Angle of Twist
G =Modulus of Rigidity

$$\phi = \cos^{-1}\left[1 + 2\cdot\left(\frac{d}{w}\right)^2\right]$$

$$F = \frac{T}{w/2} = \frac{2T}{w}$$

$$Force_{reaction} = \frac{2t^3G}{L}\cdot\left[5.33 - 3.36\frac{t}{w}\left(1 - \frac{t^4}{12w^4}\right)\right]\cdot\cos^{-1}\left[1 + 2\left(\frac{d}{w}\right)^2\right]$$

$$Stress_{Max} = \frac{3Gt}{8L}\cdot\left[5.33 - 3.36\frac{t}{w}\left(1 - \frac{t^4}{12w^4}\right)\right]\cdot\cos^{-1}\left[1 + 2\left(\frac{d}{w}\right)^2\right]\times \left[1 + 0.6095\left(\frac{t}{w}\right) + 0.8865\left(\frac{t}{w}\right)^2 - 1.8023\left(\frac{t}{w}\right)^3 + 0.9100\left(\frac{t}{w}\right)^4\right]$$

Figure 1. Torsion Bar with Rectangular Cross Section. A torsion bar may start out looking like a cantilever beam, but as it is twisted down the length instead of bent at the free end, the mathematics becomes far more complicated. Louvered contacts start out in the pre-twisted state (but unstressed), and are deflected along one edge by a flat blade towards the straight condition. Note that this would also add some bending stress to the design, but for this discussion we will assume pure torsion, in order to keep the math (relatively) simple.

One last look at the equations shows that the stress and force depend on the **modulus of rigidity** (G) (also known as **shear modulus** or **torsion modulus**) of the material. This is not available for most materials, but it is easily derivable from the elastic modulus (E) which is almost always known, and Poisson's ratio (ν), which for copper alloys ranges from 0.29 to 0.32, and can reasonably be assumed to be 0.3.

$$G = \frac{E}{2\cdot(1+\nu)} \approx \frac{E}{2.6}$$

The interesting thing if you look at the equations is figure 1 is that force is a cubic function of thickness and stress is a linear function of thickness, just as they were for cantilever beams. Length is also in the denominator in both cases. Unlike for cantilever beams, width is not in the numerator for the force equation, and also appears in the stress equation (this time in the denominator). Also, it is not the width per se that is important, but the ratio of thickness to width and deflection to width. Thick, narrow beams will provide much greater contact force (and stress) than thin, wide beams. This is because the beam width affects the moment generated by twisting, whereas it had no effect on the moment generated by bending.

The next issue of Technical Tidbits will compare one-piece and two-piece female terminal (socket) contacts.

Torsion Bar Springs (continued)

Most louvered contacts are not fixed at one end and free to twist at the other end, but they are fixed more like simply supported beams that are free to elongate axially as they are twisted towards flat. The big advantage of this type of spring is that it has a high contact force in a small area (i.e. high **Hertz stress**), so it can disrupt and wipe away most oxides or contaminating surface films during the mating process, resulting in good electrical contact. [Hertz stress is an interesting and somewhat controversial topic in the connector industry. There will be an issue or three on this topic in future Technical Tidbits.]

Looking at Figure 2 below, you can see really see how a louvered spring would essentially “grab and hold onto” a male pin, whether it is a flat blade (box terminal – center) or circular pin (cage-style socket contact – right side).

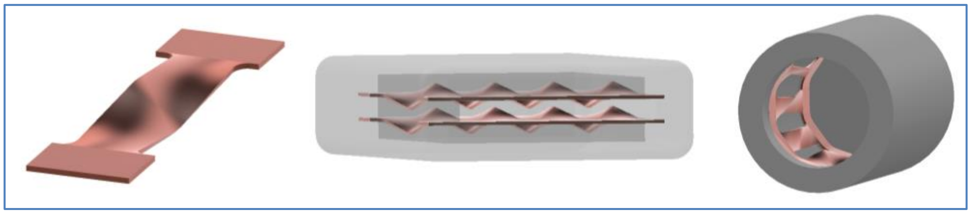


Figure 2. Louvered Electrical Contacts that are in the form of Torsion Bars.
The image at the far left shows a single louvered beam. Next is an automotive box terminal using louvered contacts (the housing is shown as semi-transparent to show the beam orientation). The far right shows a circular socket with louvered contacts inside.
Wire terminations and other housing features are not shown for the sake of simplicity.
The contacts, which would be plated in reality, are shown bare for clarity.

One caveat to note is that in a louvered contact, the male contact would deflect the near edge of the louver. The edge on the far side would not be loaded at all. (Pure torsion would require an equal and opposite load on the near edge and far edge.) So, in reality, the loading on a louvered contact is part torsion, and part bending. The true maximum stress will be the Von Mises stress generated by the shear stress from torsion combined with the bending stress. The contact force will be the linear addition of the bending reaction force and the torsional reaction force, using the principle of superposition.

The mathematical treatment of such a loading system would be quite complex. Luckily for us, there are plenty of finite element analysis packages available that would be able to calculate normal force, maximum stress, etc. much more quickly and easily than cranking through horrendously complicated algebraic equations. Insertion force is much more unpredictable via FEA than normal force, thanks to the complicated nature of surface contact calculations. This is why most performance specifications require that actual parts be tested.

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TECHNICAL TIDBITS

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