Torque Loss and Stress Relaxation in Constant Torque Springs

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Abstract

Constant torque springs are manufactured from spring steel strip that in some applications is stressed beyond the yield strength to achieve maximum torque-to-weight ratio. An adverse consequence of a high state of stress is torque loss resulting from stress relaxation, which typically occurs over a prolonged period of time, accelerated by thermal cycles or continuous elevated temperatures. This poster paper discusses a case of torque loss resulting from thermal cycling of a spacecraft hinge spring manufactured from Type 301 corrosion resistant steel strip, cold worked to the extra hard condition.

The equations governing the design of the constant torque (Neg'ator®) spring are reviewed. Included is a discussion of ongoing work to better understand the design, manufacturing, and stress relieving of constant torque springs, particularly in regard to stress relaxation and delayed cracking from sustained high stress levels in aerospace environments.

Introduction

Constant torque springs are sometimes stressed beyond the yield strength, by design, in order to obtain maximum torque-to-weight ratio. The springs are initially fabricated as a tightly wound coil of steel strip, and may have multiple laminates. The material is stress relieved, possibly before and after forming into the coil, at temperatures up to 425°C. When installed on the hinge, the spring is reverse flexed from the take-up spool onto the spool driving the output shaft (Figures 1 and 2).

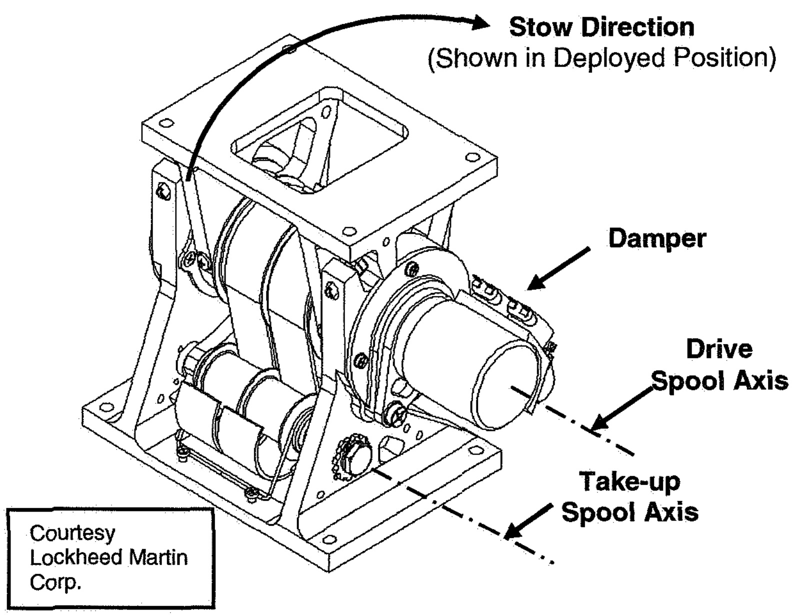

Figure 1.
Two Constant Torque Springs Installed in 90-degree Hinge

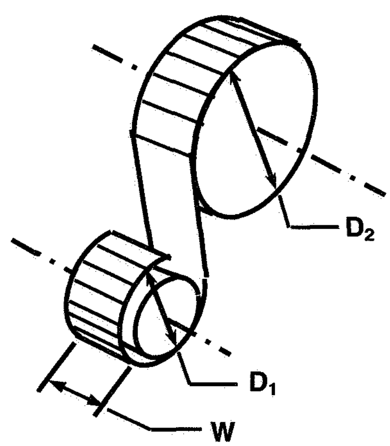

Figure 2.
Constant Torque Spring Critical Dimensions

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This work was initiated following loss of torque in spacecraft hinge mechanisms after a number of temperature cycles during thermal vacuum testing. The torque loss at the end of the deployment motion for one of the 180-degree hinges was 10 percent, as shown in Figure 3. (For a 90-degree hinge, as pictured in Figure 1, the torque loss was 7 percent). Note that the torque at the stowed position remains unaffected. Evidently, it is the slope of the deployment curve that is affected by the thermal cycling during this particular test. On tests of other hinges, however, the slope remained approximately the same, but the mean torque level decreased.

Empirical Analysis of Torque Loss

After considering a number of other possible explanations, it was concluded that the 10 percent thermal cycling torque loss was due to stress relaxation. The most likely other explanation, increase in interlaminate friction, was eliminated by a special test using only one laminate. (The torque loss was about the same amount). Since stress relaxation is a function of time, temperature, and stress level, it can be further surmised that the 37°C elevated temperature portion of the thermal cycles, combined with high bending stress levels, are the primary factors contributing to the observed torque loss.

The Figure 3 data also shows $\pm$ 7 percent hysteresis and an initial negative slope of 15 percent of the maximum torque. It is also surmised (tentatively) that these adverse effects can be attributed in some measure to high bending stresses.

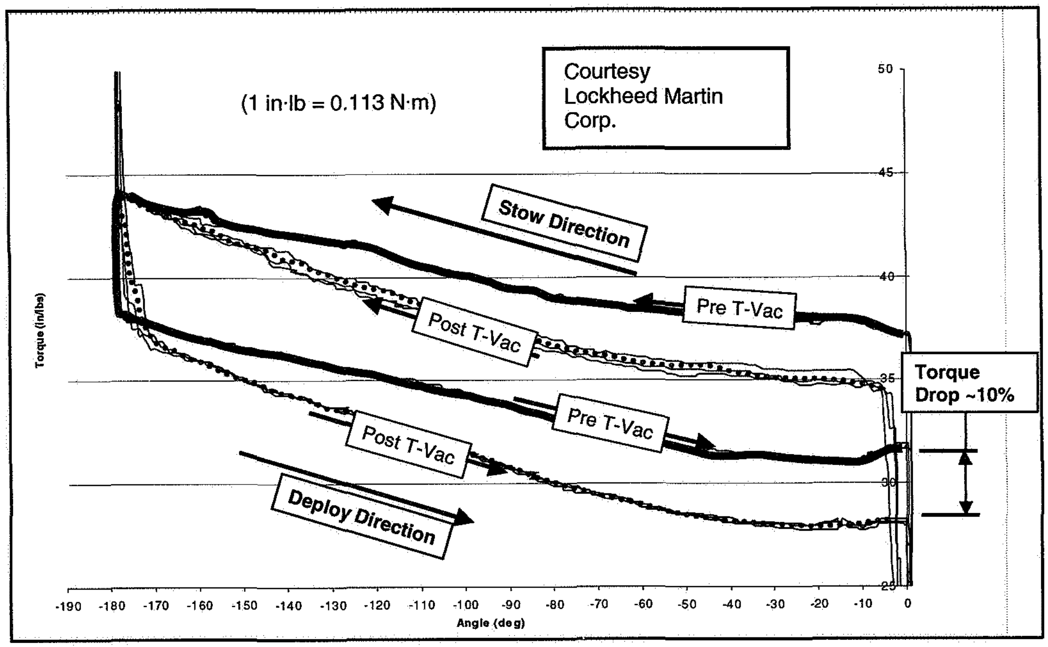

Figure 3. Torque vs. Angle for 180-degree Hinge

As the torque loss analysis proceeded, calculations were made that indicated that the springs were stressed beyond the yield strength (discussed in the following section). This conclusion is supported by examination of springs used for development testing, shown in Figure 4. Yielding of the spring stock from the high operating stress and strain levels after being reverse flexed is evident from observed distortion of the springs. The material is Type 301 corrosion-resistant steel, 0.0229-cm (9-mil) thick, cold worked to the extra hard condition (60 percent cold reduced). It is seen from Figure 4 that after being removed from the hinges, the springs are no longer in a tightly wound coil, but are expanded out of shape into a loose spiral. It can be concluded that being stressed beyond yield, combined with thermal cycling, contributes to stress relaxation and the associated reduction of torque.

Courtesy
Lockheed Martin
Corp.

Figure 4. Springs Showing Various Degrees of Yielding

To find out if performing multiple thermal cycles has resulted in the springs achieving a state of stress stability, a special test was performed where the torque output was measured after every three thermal cycles. It was found that the torque became essentially constant between 9 and 15 cycles at a level that still allowed sufficient torque margin. Additional thermal cycles are planned to confirm this.

Stress and Strain Analysis

The practice of operating constant torque (Neg'ator®) springs at high stress, sometimes beyond yield, derives from general usage where the design stress level is based on low cycle fatigue requirements and the number of operational cycles. The invention of this type of spring is credited to Frank A. Votta Jr. In his technical paper[1], he based the number of allowable cycles on a formula for (what he called) stress factor, f_S, shown here in Equation (1).

$$f_S = t\left(\frac{1}{D_1} + \frac{1}{D_2}\right) \tag{1}$$

f_S = Stress Factor
t = Spring Strip Thickness
D_1 = Fabricated Diameter (Fig.2)
D_2 = Drive Spool Diameter (Fig. 2)

This is identical in form to the usual formula from strength of materials for calculating bending strain in the outer fibers.

$$\varepsilon_b = t\left(\frac{1}{D_1} + \frac{1}{D_2}\right) \tag{2}$$

ε_b = Bending Strain

The first term, t/D_1, is the bending strain from the change of curvature during the transition from the original coiled diameter D_1 to the straight section between the coils. The second term, t/D_2, is the strain from further changing the curvature by reverse flexing the spring from the straight transition to the diameter of the drive spool D_2.

Stress vs. Strain Characteristics of Spring Steel Strip

If one were to calculate the quasi-linear bending stress on the assumption that the modulus of elasticity in simple tension were constant and equal to the compression modulus, we would come to the impossible finding that the calculated bending stress would exceed the ultimate stress. The spring represented by Figure 3 has a calculated bending strain of 1.1 percent. Based on a modulus of elasticity of E = 193,000 MPa (28,000,000 psi) and a Poisson's ratio of 0.3, the linear formula for bending stress

(Equation 3) would give a value of 2330 MPa (338,000 psi), which is well above the characteristic ultimate tensile strength of 1860 MPa (270,000 psi) for extra hard Type 301.

$$\sigma_b = \frac{\varepsilon E}{(1-\nu^2)} \tag{3}$$

σ_b = Bending Stress (in Outer Fibers)
E = Modulus of Elasticity in Tension
ν = Poisson's Ratio

This formula represents the quasi-linear change in bending stress from the state of residual stress in the fabricated spring coil. (This method of calculating the bending stresses in a constant torque spring is summarily discussed in the Associated Spring Design Handbook[2], which also cites Votta's paper). The spring manufacturer can intentionally pre-stress the spring in a process called strain hardening, to induce residual bending stresses of opposite sign, and thus reduce the stresses when the spring is installed on the hinge or spring motor. (It is tentatively believed that this was not done for the particular springs involved in this case of stress relaxation).

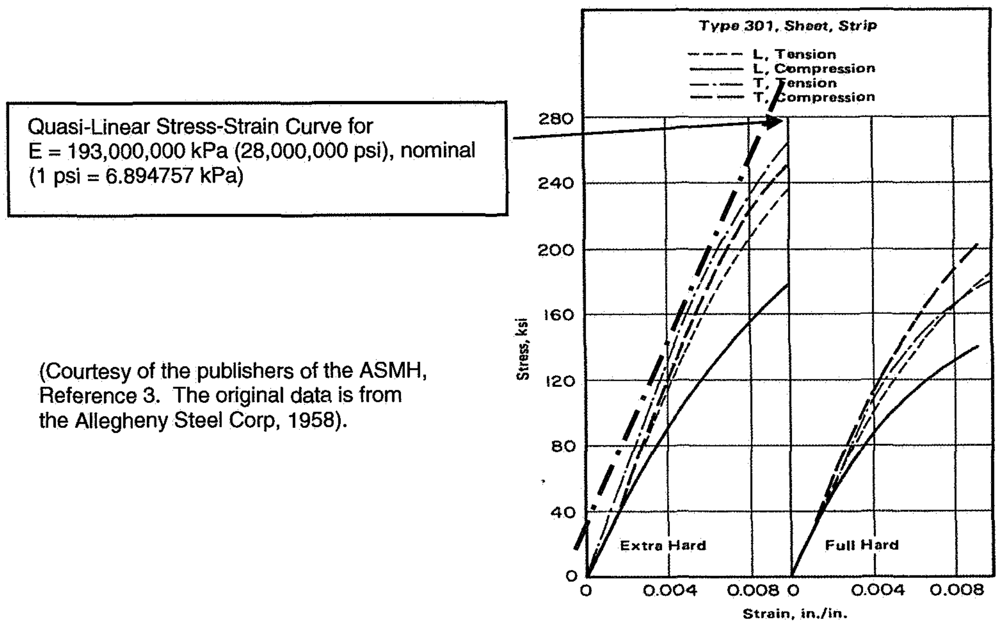

FIGURE 3.0212. STRESS-STRAIN CURVES FOR SHEET AND STRIP COLD ROLLED TO FULL-HARD AND EXTRA-HARD TEMPERS (11)

Figure 5. Stress/Strain Curves for Type 301, Extra Hard Condition

Tensile and compression test curves for extra hard Type 301, from the Aerospace Structural Metals Handbook[3] (ASMH), are shown in Figure 5. A hypothetical (quasi) linear curve is added to the graph. From comparison of calculated operating strains for constant torque springs with typical published tensile or compression test data (as shown in Figure 5), it can be concluded that the behavior of steel strip at high strain levels is different in bending compared to pure tension and compression. Type 301 extra hard shows higher values of elastic modulus in tension than in compression. The tensile and compression curves start to droop near peak values, such that the ultimate failure stress is less than would be predicted by the nominal assumption that the stress strain/curve is linear all the way to ultimate. Stress/strain curves published in the ASMH for extra hard Type 301 typically terminate at 1.0 percent strain, (although some of the numerical data in the ASMH state elongation to failure greater than 1.0 percent).

Another example that infers greater elongation to failure in bending compared to simple tension is the required bending test for a slightly less cold worked material, Type 301 full hard. (Stress/strain curves for

Type 301 full hard are also shown in Figure 5). Type 301 full hard is required by MIL-HDBK-5F[4] to be capable of being bent around a rod of diameter six times the thickness of the spring material, without the occurrence of surface cracking. Using Equation 2, the average bending strain calculated for 0.0254-cm (10-mil) thick strip bent around a 0.1524-cm (60-mil) diameter rod (neutral surface assumed at the 0.1778-cm (70-mil) mean diameter) is 14.3 percent, compared to the MIL-HDBK-5F requirement for minimum elongation to failure in simple tension of 8 percent. No similar bending requirement is given for the extra hard Type 301, as this level of cold work is not controlled by MIL-HDBK-5 or by ASTM specifications. Accordingly, it is prudent for the end user to provide specifications for the extra hard version, rather than rely entirely on the spring vendor's judgment and practice.

For a limited number of cycles at ambient temperature, strains beyond the yield strength (based on 0.2 percent offset) can be tolerated without failure. From a graph in Votta's paper, his recommended allowable stress factor (strain) for less than 5000 cycles is 2.0 percent, for what he calls "safe design", using 1095 carbon steel. In the Associated Spring Design Handbook[2], a graph is presented showing an "allowable" (quasi-linear) stress of 2,760 MPa (400,000 psi) for fewer than 260 cycles. In actual practice, however, the design of constant torque springs is based on empirical data, rather than Votta's formulas. In a couple of other major spring manufactures' design guides, Type 301 is rated at up to 2000 cycles for strains above 1.0 percent or greater. (Typically 50 to 100 cycles are considered adequate for life testing of deployables that only operate once in orbit, plus the cycles needed during manufacture and testing).

Normally, one would not be concerned over delayed cracking in typical spacecraft storage environments. However, for springs stowed at stress levels beyond yield over prolonged periods, this aspect merits some consideration. Type 301 is on the list of materials considered to be resistant to stress corrosion cracking; however, when cold reduced to the full hard or extra hard state, a phase transformation from austenite to martensite occurs, which would logically obviate a totally exempt status. Looking into this further, the storage atmosphere is typically at controlled humidity, and at the launch site the spacecraft is usually in an air-conditioned payload housing. One reference, by Phelps and Loginow [5], cites an exposure test to atmospheric corrosion at Kure Beach, N.C. of Type 301, 60 percent cold reduced, for periods of 240 to 370 days, during which the test specimens showed "excellent resistance to stress corrosion". However, these specimens were only stressed to 75 percent of the stated yield strength of 1640 MPa (238,000 psi). Data for springs stressed beyond yield, sustained over typical aerospace storage periods of five years, has not been located. On the other hand, neither has breakage of this type of spring been reported, due to delayed cracking (or for any other reason) in aerospace usage.

Torque Determination

In Votta's technical paper, he gives a formula for calculating the torque on a constant torque spring motor. The formula currently derived (Equation 4), based on strain energy, is almost the same as Votta's except for the factor in the denominator involving Poisson's ratio.

$$T = \frac{EWD_2t^3}{12\left(1-\nu^2\right)}\left(\frac{1}{D_1}+\frac{1}{D_2}\right)^2 \qquad (4)$$

T = Torque
W = Width (Figure 2)

This factor also appears in the quasi-linear stress formula, Eqn. 3, and is conventionally used to adapt the standard handbook formulas for beam bending of exceptionally wide beams. It results from an elastic state called "plain strain", which is applicable to a thin, wide beam (strip) in bending. (An interesting comment from A. M. Wahl, a noted authority in spring design of that era, was published in the "Discussion" portion of Votta's paper. Wahl stated that this factor should have been present in Votta's formula. Votta's response was that their test data showed that this factor did not apply because of energy lost in transverse [anticlastic] curvature of the straight length of spring between the two spools.)

Equation 4 implies that the torque should be constant with angular displacement. However, as seen from Figure 3, there is a fairly linear, negative slope to the deployment curve. It is surmised that stresses beyond yield contribute to the magnitude of the negative slope. Hysteresis is another component of torque loss that adds to the total loss. From tests with single laminates it was found that interlaminate friction is not a significant cause of the hysteresis. (The single laminate has approximately the same hysteresis as multiple laminates). As of now, it is not known how much of the hysteresis is internal to the material from stress related yielding of the spring stock, and how much is due to bearing friction.

One broad objective is to better understand why the physics (applied mechanics) behind the theoretical formula for torque (Equation 4) does not account for the downward slope of the torque vs. deflection curves. This will assist in design optimization of the relationships between the variables affecting torque and torque loss. It may be that optimum design would be achieved at lower stress levels, by reduced laminate thickness and increased number of laminates. It is desirable to minimize this decreasing torque slope, because the maximum torque is usually needed at the end of deployment to actuate end-of-travel latches or wind-up cable bundles.

Conclusions and Recommendations

This poster paper has discussed how general practice for the design and manufacturing of constant torque springs using empirical design data provides springs that may be stressed beyond the yield strength. Typically, highly stressed springs are subject to stress relaxation and related torque loss under prolonged load and elevated temperature conditions.

It appears that torque loss due to thermal cycling, for the springs cited herein, had achieved a state of stability, in that the torque output of the unwound spring actuated hinges was not continuing to decrease with each successive cycle. Additional cycles are expected to confirm this. It also remains to be determined whether maintaining stresses above the yield strength over periods of prolonged storage in aerospace environments can be done without excessive loss of torque from stress relaxation and without loss of immunity to delayed cracking.

The final outcome from this work will likely be that Type 301 extra hard corrosion resistant steel will continue to be an optimum material for one-time operation in space of constant torque spring actuated hinges. Particularly for the extra hard condition, which is not controlled by ASTM or military specifications, the end user and spring manufacturer should be mutually cognizant of spring material requirements, such as yield and ultimate strength and ductility, and stress relieving and pre-stressing procedures.

Acknowledgments

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