Tech-Spring Report 20C
Use of Strain Gauges to Measure Static stresses in Torsion Springs

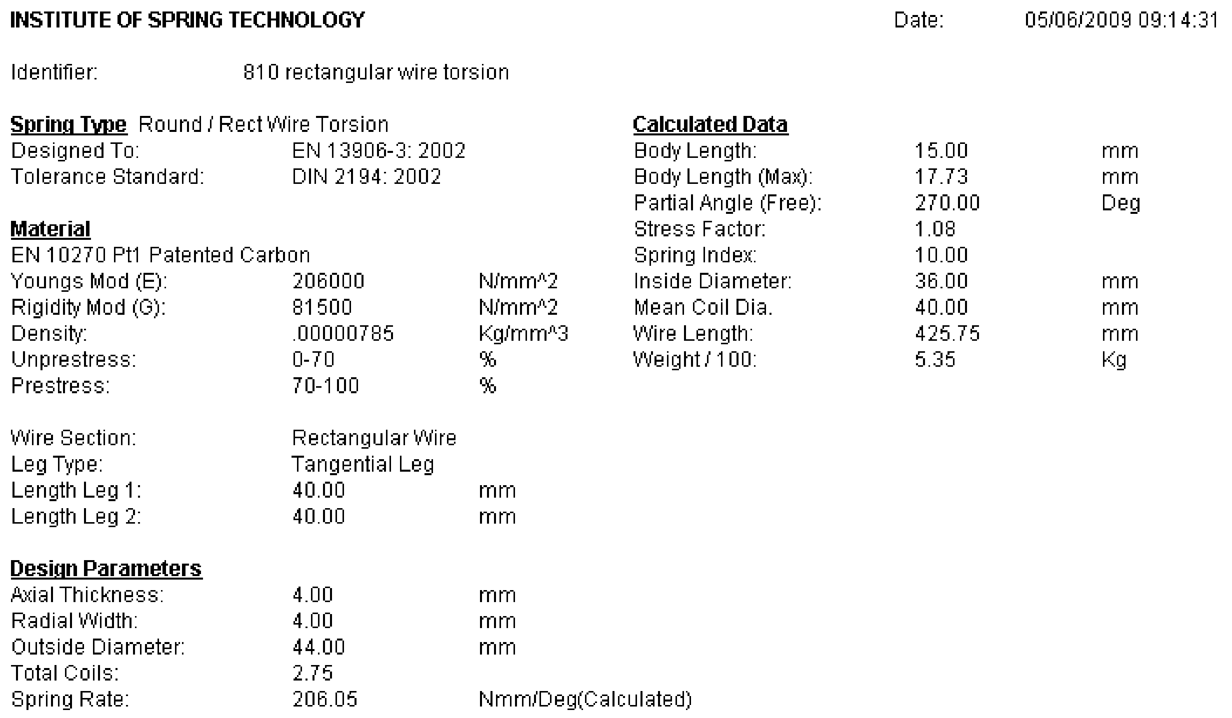
1. Introduction

The stresses in torsion springs can be readily calculated using traditional theory, however this approach ignores the effects of friction, mandrel diameter and loading direction.

This report looks at using strain gauges to investigate these effects.

2. Spring design

The spring used was manufactured from 4mm square wire to the design below:



INSTITUTE OF SPRING TECHNOLOGY Date: 05/06/2009 09:14:31

Identifier: 810 rectangular wire torsion

Spring Type Round / Rect Wire Torsion

Designed To:	EN 13906-3: 2002	
Tolerance Standard:	DIN 2194: 2002	
Material		
EN 10270 Pt1 Patented Carbon		
Youngs Mod (E):	206000	N/mm^2
Rigidity Mod (G):	81500	N/mm^2
Density:	.00000785	Kg/mm^3
Unprestress:	0-70	%
Prestress:	70-100	%
Wire Section:	Rectangular Wire	
Leg Type:	Tangential Leg	
Length Leg 1:	40.00	mm
Length Leg 2:	40.00	mm
Design Parameters		
Axial Thickness:	4.00	mm
Radial Width:	4.00	mm
Outside Diameter:	44.00	mm
Total Coils:	2.75	
Spring Rate:	206.05	Nmm/Deg(Calculated)

Calculated Data		
Body Length:	15.00	mm
Body Length (Max):	17.73	mm
Partial Angle (Free):	270.00	Deg
Stress Factor:	1.08	
Spring Index:	10.00	
Inside Diameter:	36.00	mm
Mean Coil Dia.	40.00	mm
Wire Length:	425.75	mm
Weight / 100:	5.35	Kg

Stress Data

	Lower	Operating Positions	
		% Tensile	
	Tensile	1	2
SL	1270	30 U	68 U
SM	1480	26 U	59 U
DM	1480	26 U	59 U
SH	1680	23 U	52 U
DH	1680	23 U	52 U
Specified			

Operating Data

	Operating Positions	
	1	2
Torque (Nmm)	4121.1	9272.4
Spring Deflection (Deg)	20.00	45.00
System Angle (Deg)	418.07	444.48
Partial Angle (Deg)	290.00	315.00
Stress (N/mm^2)	386	869
Inside Diameter (mm)	35.21	34.26
Body Length (Max) (mm)	17.96	18.25
Load Tol. Grade 1 (Nmm)		
Load Tol. Grade 2 (Nmm)		
Load Tol. Grade 3 (Nmm)		

To ensure the test spring was not subject to plastic deformation the maximum deflection was limited to 45 deg / 7.27Nm, equating to a theoretical uncorrected stress of 869N/mm^2.

3. Equipment used.

A standard gauge amplifier and switch box were used for the test work. Gauges with a grid size of 2 x 1mm were glued to the spring at four locations shown below.

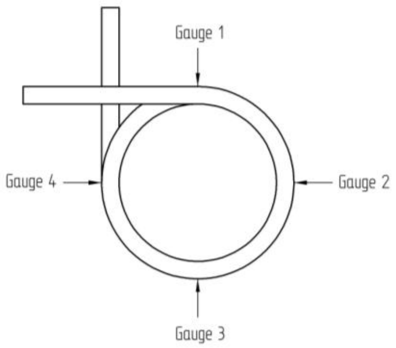


Fig 2 Gauge locations

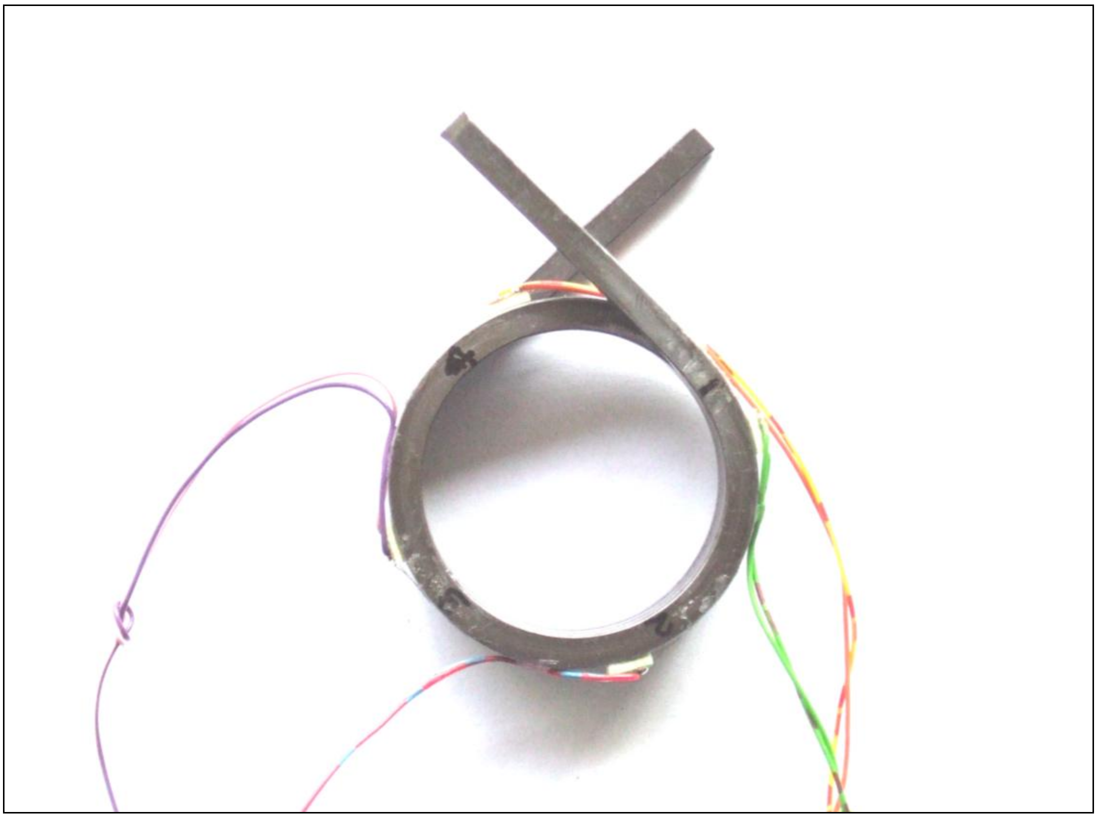


Fig 3 Gauged spring

A cyanoacrylate (superglue) adhesive was used for bonding the gauges. A fifth gauge was bonded to a piece of steel to produce a balance gauge to enable the completion of the bridge in the measuring equipment.

During testing the spring was supported on interchangeable mandrels of Ø32,30,25mm diameter.
The spring legs were loaded using Ø9mm pins at a radius of 40mm from the jig centre.

4. Test Proceedures and results.

Test 1: Effect of mandrel diameter on torque output

The spring was mounted onto mandrels of Ø32,30, and25mm diameter and deflected from free to 45deg in increments of 5deg. The torque levels were recorded at each position.

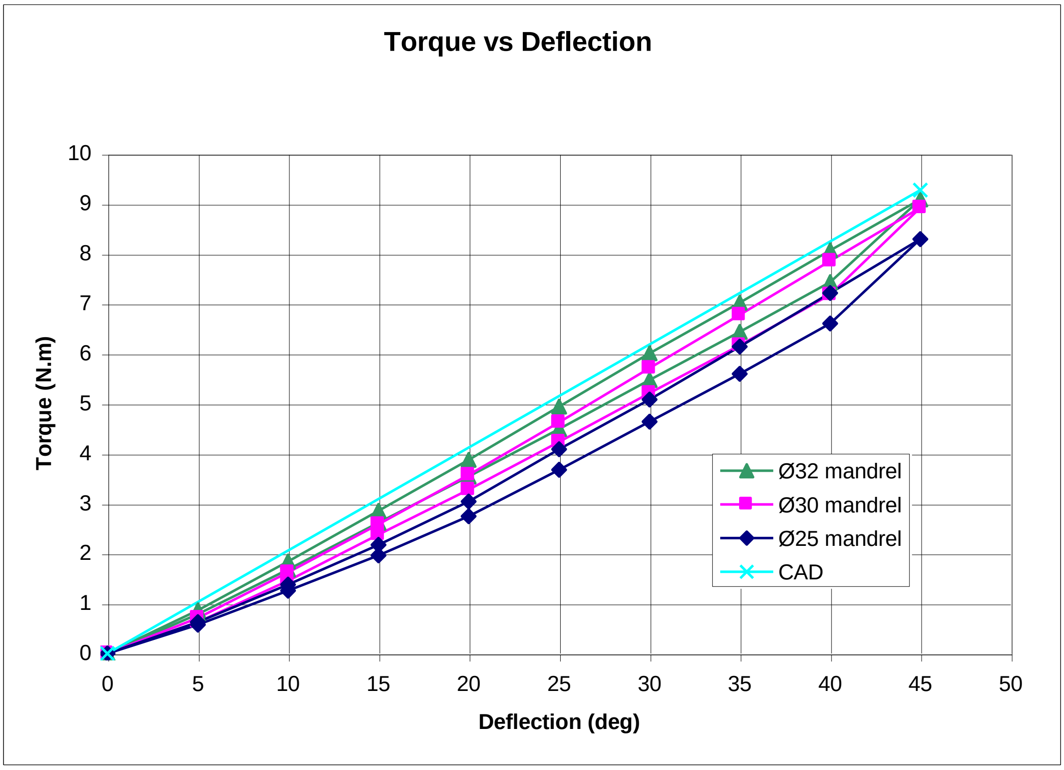


Fig 4 Effect of mandrel size on Torque output

It can be seen that the torque produced by the spring and hence spring rate is directly influenced by the supporting mandrel diameter.

Spring inside Diameter (mm)	Mandrel diameter (mm)	ID to mandrel % clearance	Max torque N.m
36	32	11.1	9.09
36	30	16.7	8.92
36	25	30.6	8.30

Table 1 Mandrel diameter and maximum torque output

Figure 5 clearly shows that the peak torque produced is a function of the spring /mandrel clearance.

Projecting the results back using a trend line indicates the torque with zero clearance is approximately 9.3N.m, which equates closely to the 9.27N.m calculated by the *IST* version 7.5 CAD software (see section 2). When small mandrels are used, the shearing and tilting of the coils becomes significant, producing a reduction in the spring rate.

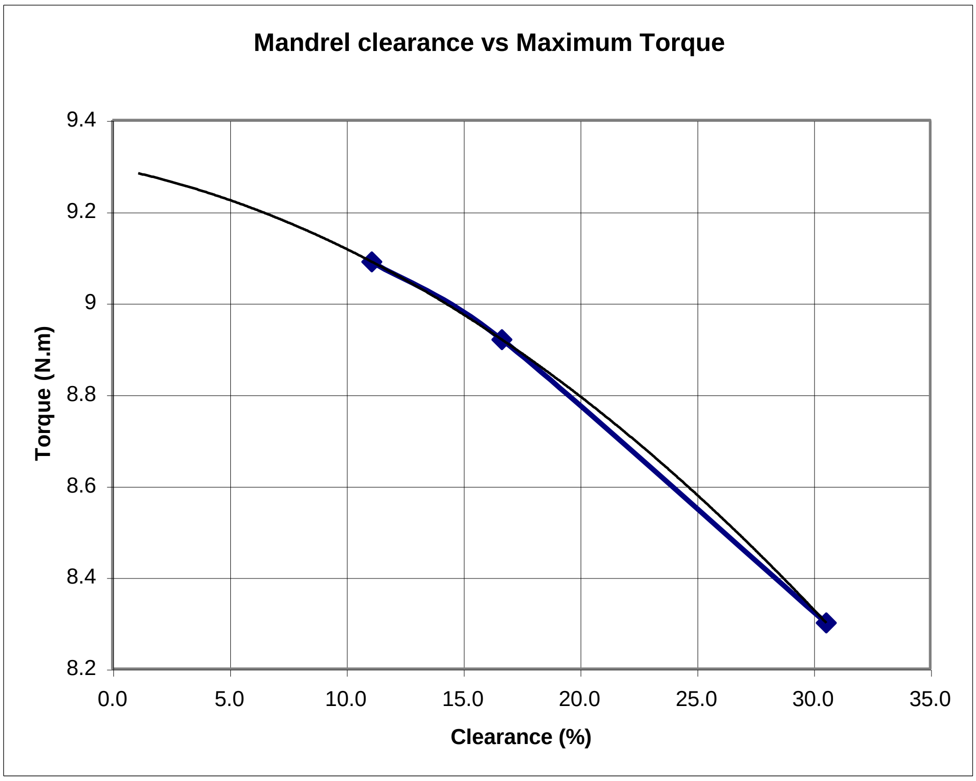


Fig 5 Peak torque vs. Mandrel to spring clearance at 45deg deflection

Test 2: Effect of mandrel diameter on stress levels

The spring was mounted onto mandrels of Ø32, 30, and25mm diameter and deflected from free to 45deg in increments of 5deg. It is known that fatigue failures on torsion springs usually occur at the moving leg relative to the mandrel. To investigate this, measurements were taken with the gauged end of the spring mounted at both the moving and fixed ends.

The stress levels were recorded at 5deg increments from free up to 45deg.

Because different mandrel diameters produced different torques for a given deflection, the results are expressed in terms of stress versus torque. Thus the theoretical stresses are calculated using torque rather that deflection.

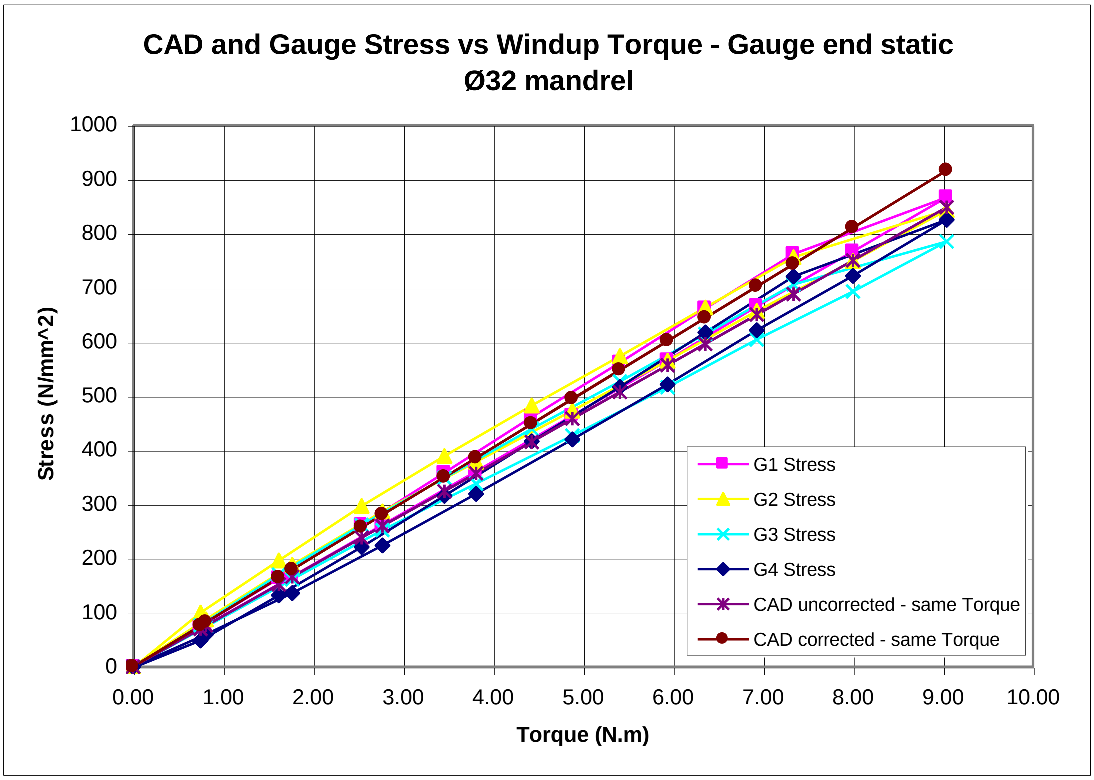


Fig 6 Ø32 mandrel, Static Gauge Stress vs. Torque

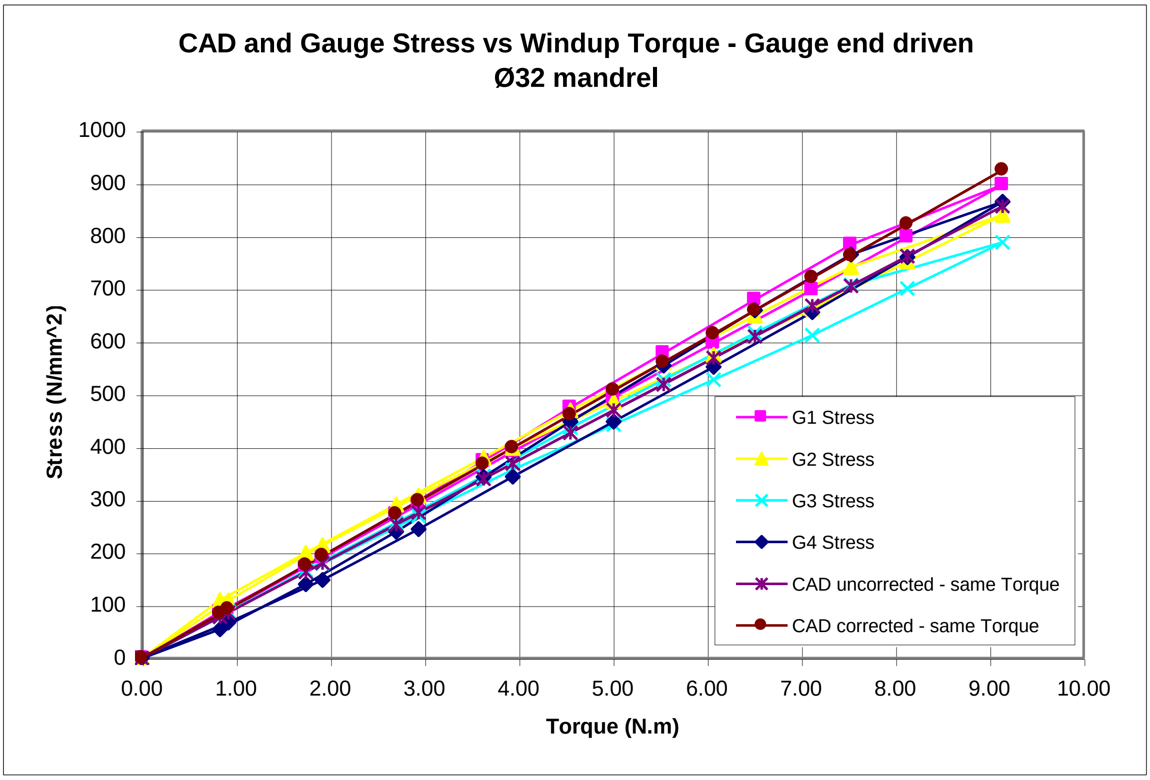


Fig 7 Ø32 mandrel, Driven Gauge Stress vs. Torque

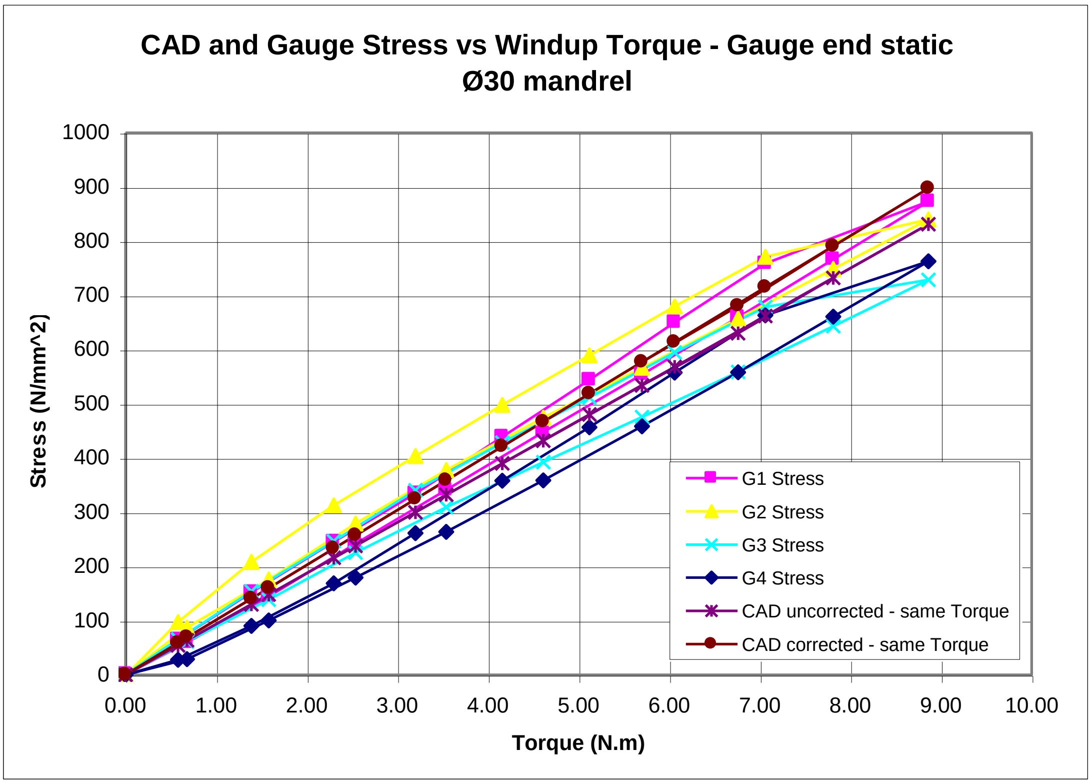


Fig 8 Ø30 mandrel, Static Gauge Stress vs. Torque

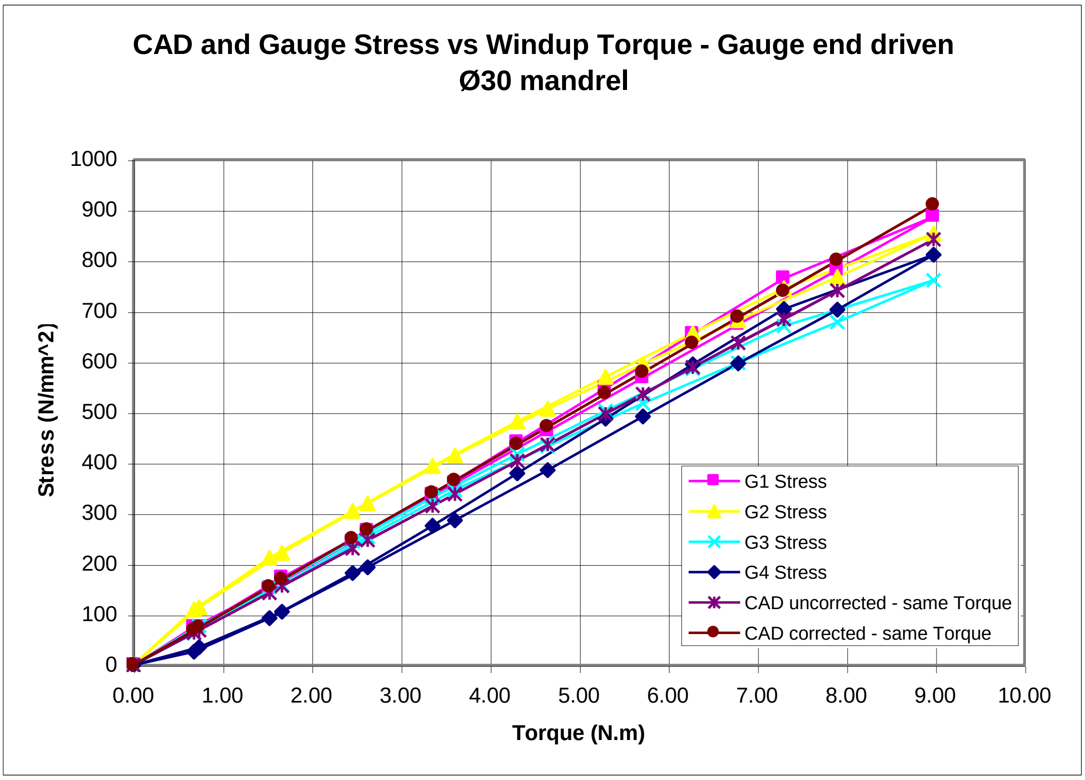


Fig 9 Ø30 mandrel, Driven Gauge Stress vs. Torque

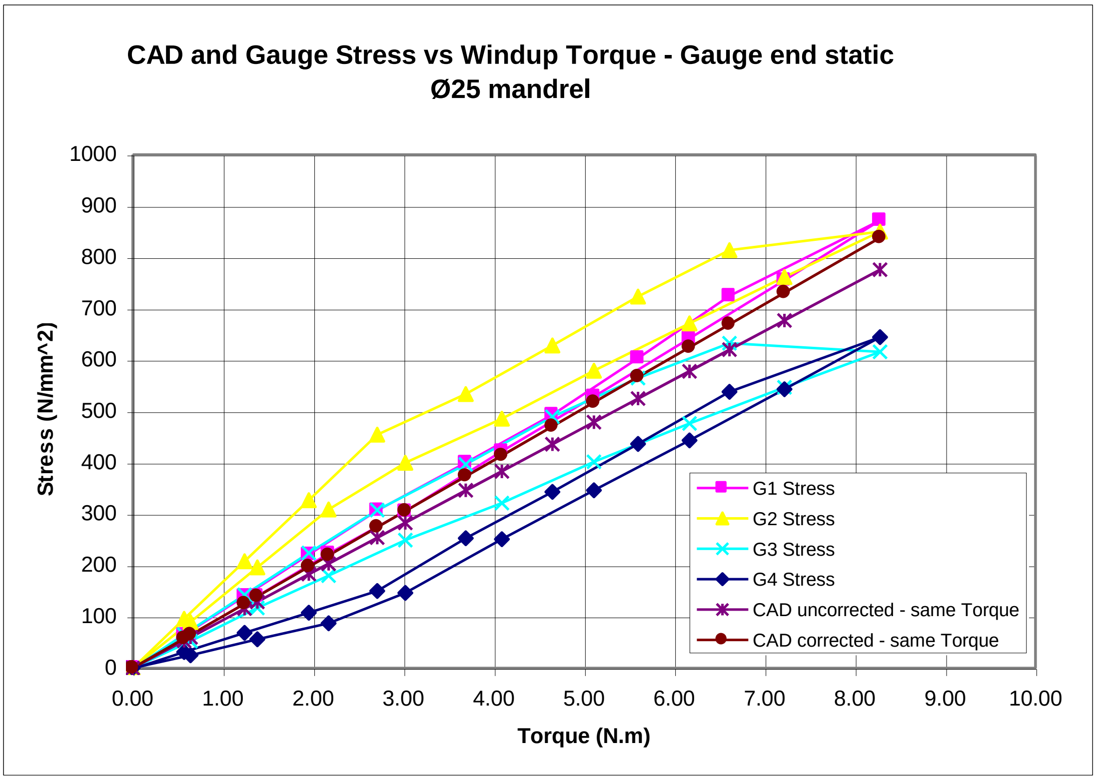


Fig 10 Ø25 mandrel, Static Gauge Stress vs. Torque

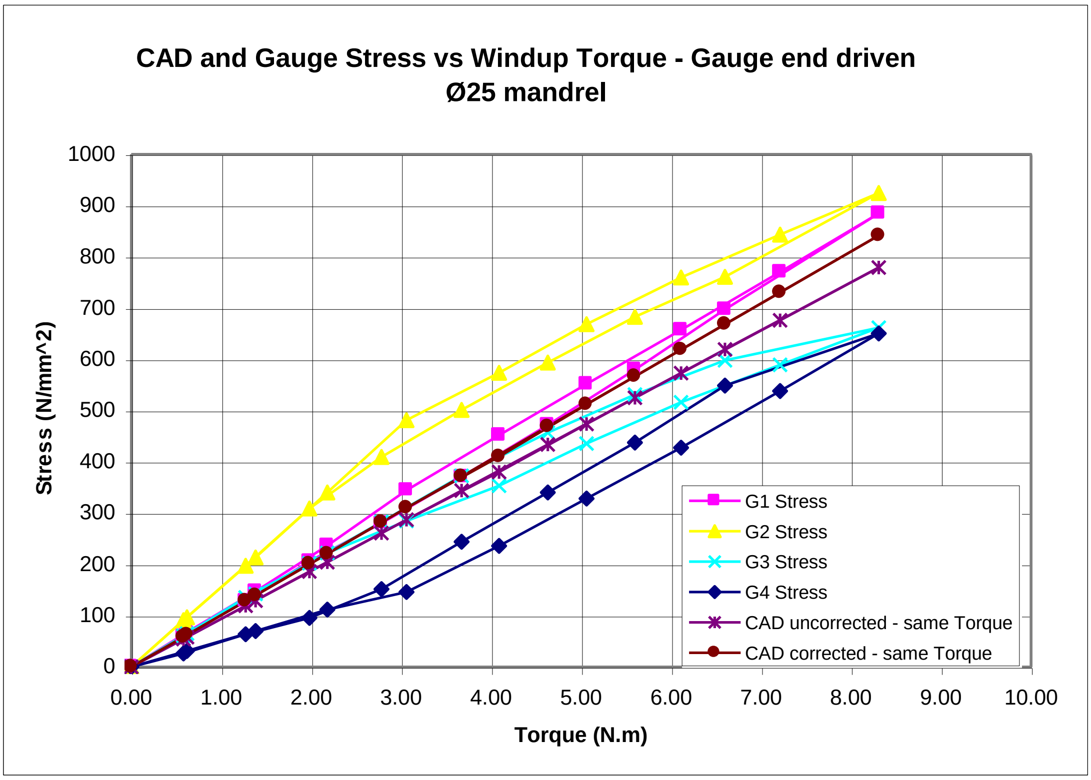


Fig 11. Ø25 mandrel, Driven Gauge Stress vs. Torque

Examination of the graphs, figures 6 to 11 shows that the mandrel diameter has significant effect on the magnitude of the stress at the various gauge positions.

To aid clarity, the stress results were interpolated for a torque of 8Nm, and the average calculated for the loading/unloading stresses measured at the four gauges. The theoretical CAD stress at 8Nm is 749N/mm^2 uncorrected and 809 N/mm^2 corrected.
The measured stress results are documented in Table 2 below.

	Mandrel Diameter					
	Ø25		Ø30		Ø32	
	Static	Driven	Static	Driven	Static	Driven
	Angle (Deg) at 8Nm					
	43.5	43.3	40.6	40.8	40.0	39.6
Position	Stress (N/mm^2)					
G1	856	850	822	814	798	809
G2	915	928	820	796	787	763
G3	685	689	714	711	731	733
G4	626	621	714	741	753	788

Table 2. Stress at the Static and Driven ends subject to a constant 8Nm torque

	Mandrel Diameter					
	Ø25		Ø30		Ø32	
	Static	Driven	Static	Driven	Static	Driven
	Torque(Nm) at 45deg					
	8.23	8.31	8.86	8.98	9.04	9.19
Position	Stress (N/mm^2)					
G1	871	884	872	886	866	897
G2	850	825	840	853	843	840
G3	616	662	729	760	785	788
G4	644	650	762	811	825	865

Table 3. Stress at the Static and Driven ends subject to constant 45deg deflection

Comparing the results for constant torque and the constant angle deflection modes, it can be seen that for the constant torque mode the stress levels in the driven and static ends of the spring are very similar.

However the results in table 3 for the constant 45deg deflection clearly show that the driven spring end is subject to a higher stress. It should be noted that the measured torques are lower with the smaller mandrels and hence the induced stresses will also be lower. If the stress levels were to be increased in proportion to the torque reduction then the smaller mandrel would induce the highest stresses at G1 & G2 positions.

For a constant torque results, with the larger Ø32 and Ø30mm mandrels it can be seen that gauge 1 is the highest stressed position, which is to be expected. However when the mandrel diameter becomes excessively small then the maximum stress position moves to gauge 2, so that with the Ø25 mandrel, gauges 3 & 4 have a reduction in stresses indicating that the coil shearing is producing an increase stress at positions 1&2 and a reduced stress at position 3&4.

Table 2, (constant 8Nm) also shows that as the mandrel diameter reduces the maximum stress measured increases.

Table 3, (constant 45deg) shows that G1 and G2 stresses are not subject to a significant mandrel diameter effect, but G3 and G4 are affected with the stresses reducing with mandrel diameter. Again it must be remembered that the torque at 45deg is less for the smaller mandrels.

The hysteresis of the spring/mandrel reduces the torque output of the spring when being released. This has the effect that the measured stress for a given measured torque is higher in the releasing direction.

Test 3: Effect of Winding direction on Torque

Using the same fixturing the spring was deflected in the unwind direction in steps of 5 degrees up to 45 degrees measuring the torque at each position.

Figure 12 shows the torque / deflection curve for the wind up and the unwind direction. It can be seen that the spring exhibits a much lower spring rate in the unwind direction and deviates considerable from the CAD predictions – see Table 4.

Method/direction	Rate (N.m/Deg)
CAD	0.206
Wind up	0.201
Unwind	0.162

Table 4. Calculated and measured spring rates

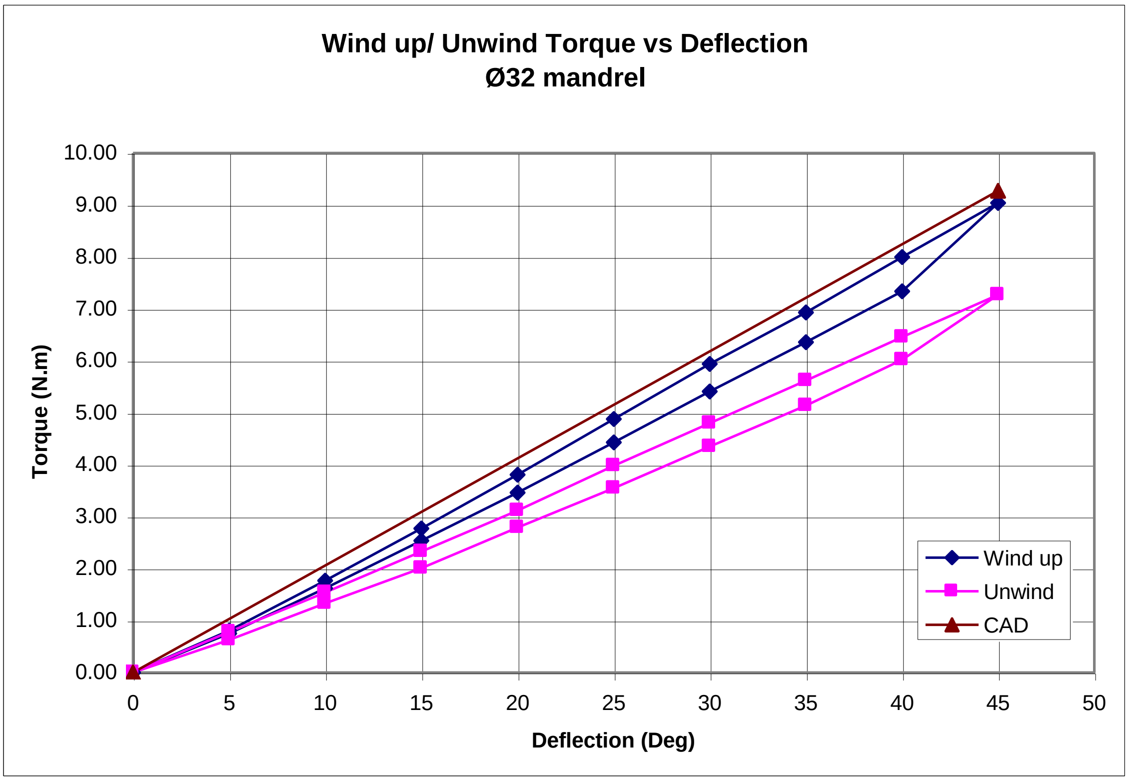


Fig 12. Ø32 mandrel, Windup / Unwind Torque vs. Deflection

Test 4: Effect of deflection direction on Stress

Using the same fixturing and a Ø32 mandrel the spring was deflected in the unwind direction in steps of 5 degrees up to 45 degrees measuring the torque and stress at each position.

In the unwind direction the tensile stress is on the spring inside surface, but due the mandrel it was not considered practical to fix gauges on this surface. Hence the gauges are measuring the compressive stress on the external surface.

By inspection of the graphs in figures 13 to 16 it can be clearly seen that the stresses measured are much higher than the CAD predictions shown in table 5.

	At applied Torque = 7.27 Nm			
	Gauge stress	CAD uncorrected	CAD Stress Factor	Measured Stress Factor
G1	866	682	1.08	1.27
G2	843	682	1.08	1.24
G3	785	682	1.08	1.15
G4	825	682	1.08	1.21

Table 5. Calculated and measured stresses and Stress Correction Factors for unwind.

It should also be noted that the stresses measured are compressive and still measured on the outside surface of the spring. The stress on the inside surfaces will be higher due to the reduction in diameter.

The graphs are all plotted as Stress vs Torque and it should be remembered that in the unwind directions the springs exhibit a lower rate and hence deflect further for a given applied torque.

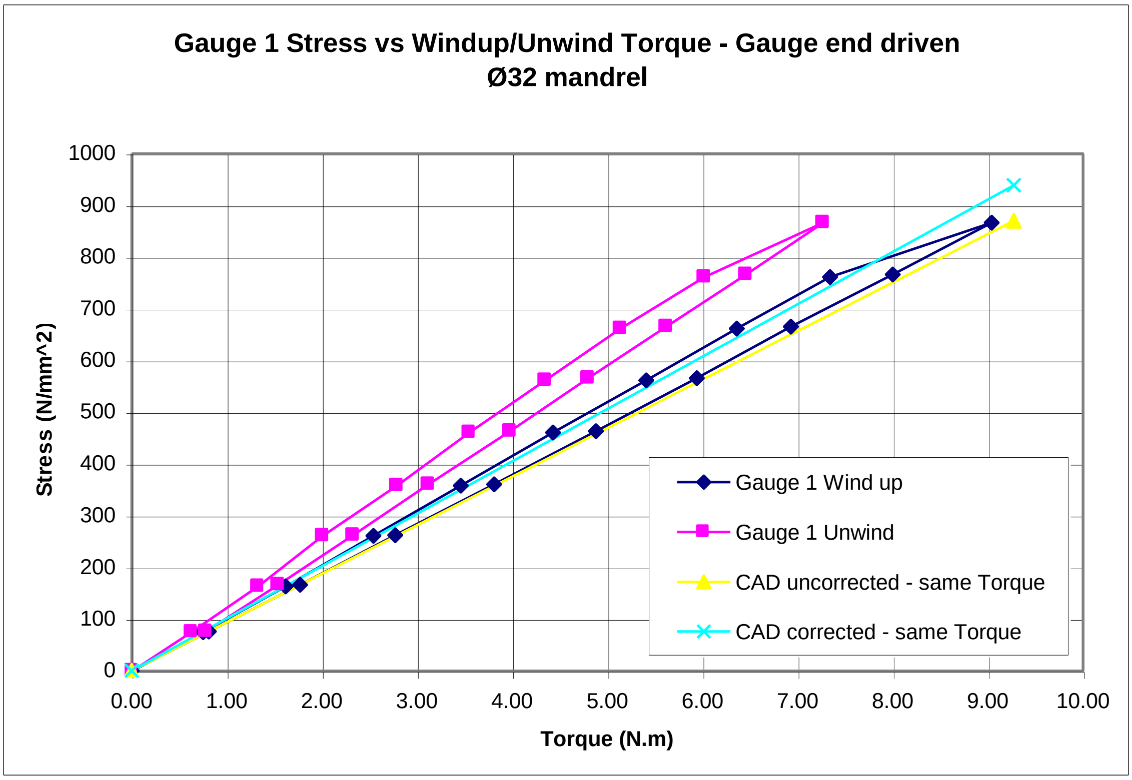


Fig 13. Ø32 mandrel, Driven Gauge 1 Stress vs. Windup / Unwind Torque

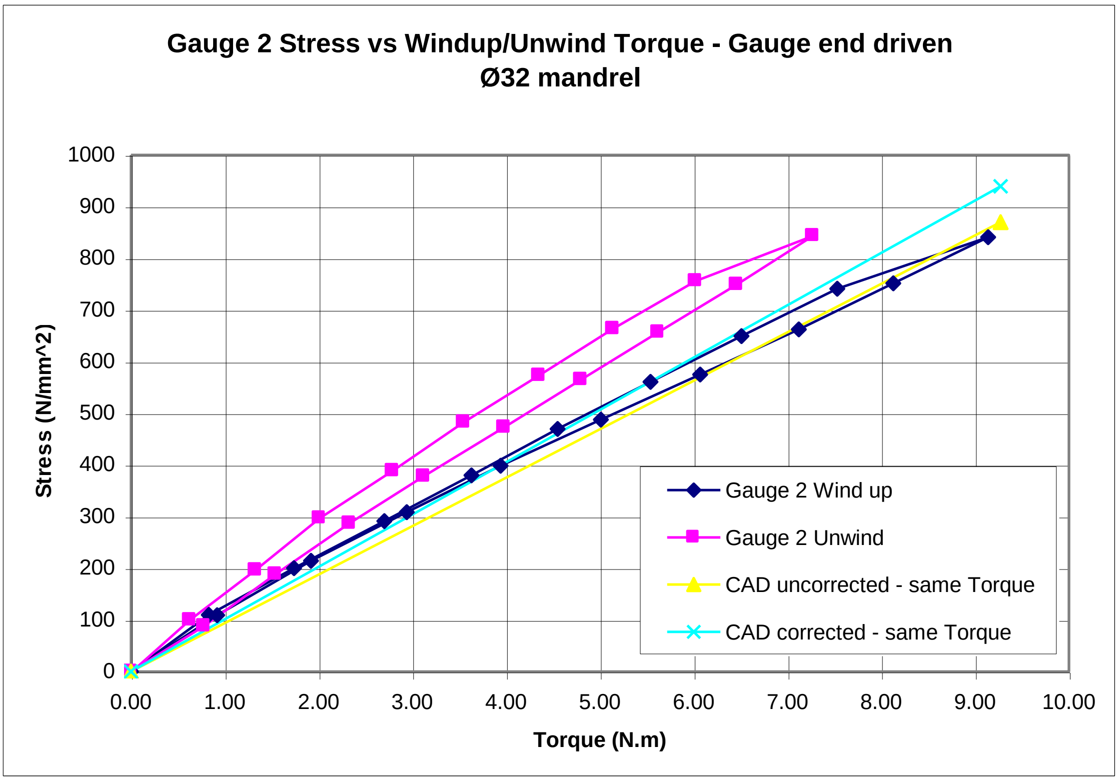


Fig 14. Ø32 mandrel, Driven Gauge 2 Stress vs. Windup / Unwind Torque

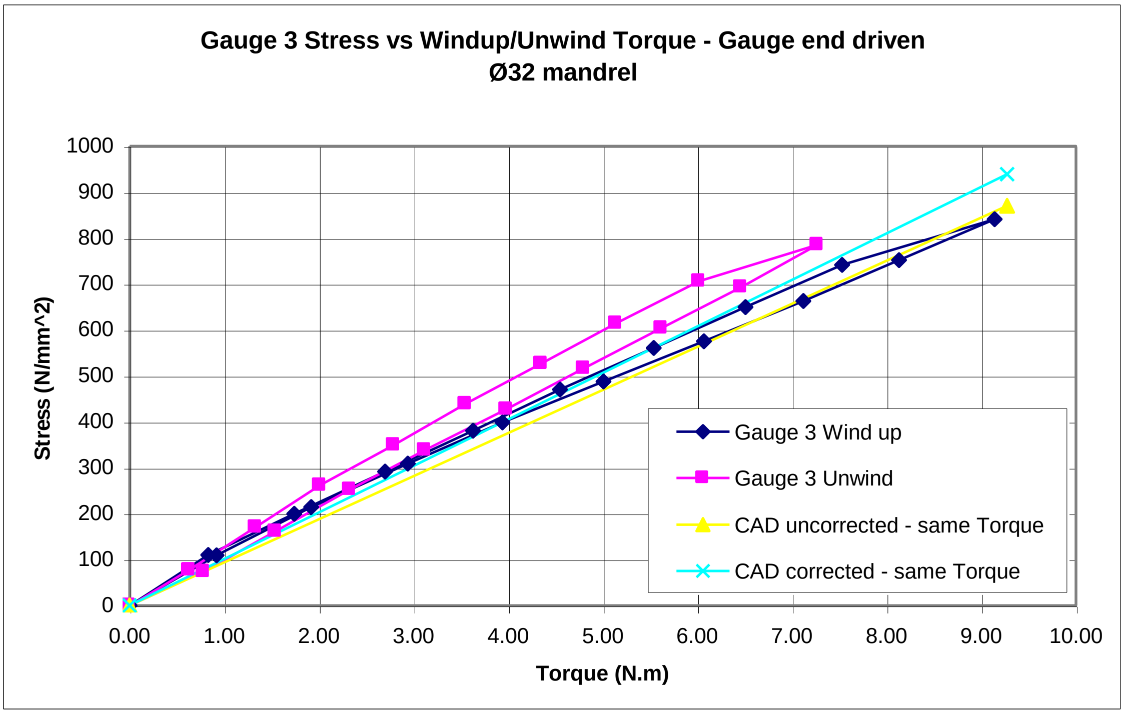


Fig 15. Ø32 mandrel, Driven Gauge 3 Stress vs. Windup / Unwind Torque

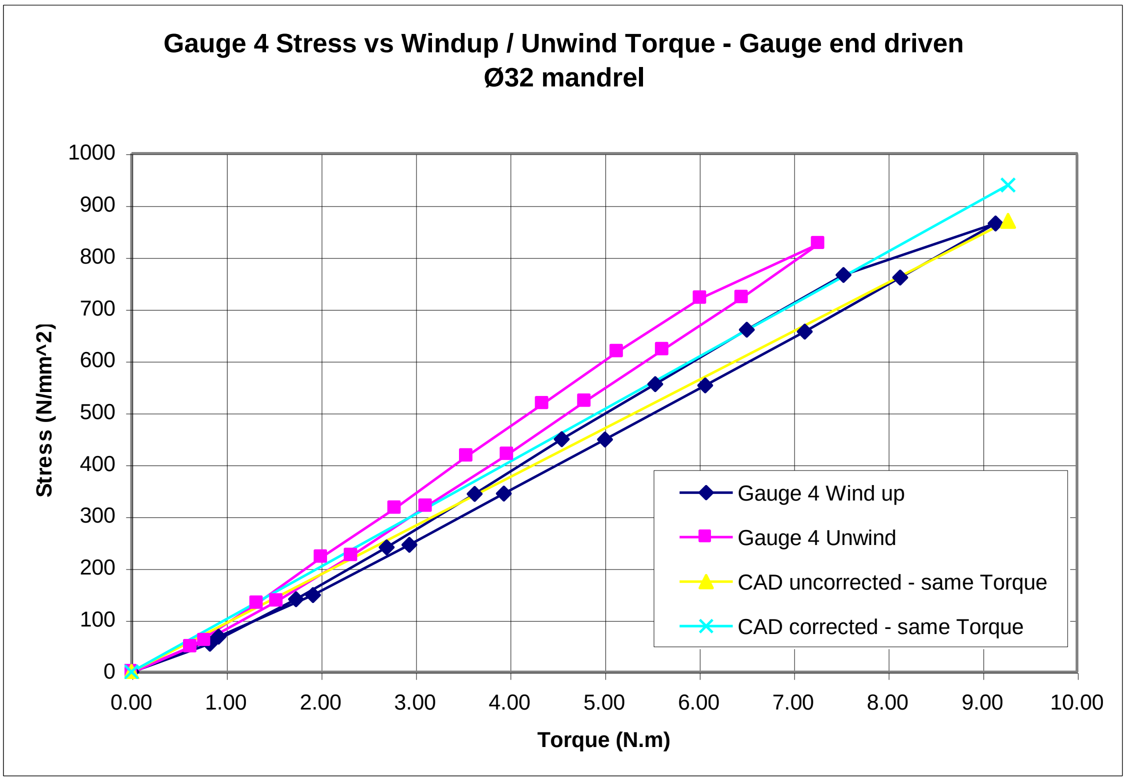


Fig 16. Ø32 mandrel, Driven Gauge 4 Stress vs. Windup / Unwind Torque

5. Discussion

The first test clearly showed the importance of using a sensibly sized mandrel so that the spring is supported correctly and the coils move in a torsional mode with shearing minimised. The results indicated that with a mandrel equal to the inside diameter the CAD torque/deflection prediction would be accurate. This is obviously impractical and for optimum performance using the formula in EN 13906-3, the mandrel diameter D_d should be:

$$D_d = 0.95\left[\left(Di - \left| A_D \right| \right) \times \frac{n}{n + \frac{\alpha_{\max}}{360}} \right]$$

Where:
Di = inside diameter of the spring (mm)
A_D = Diameter tolerance (mm)
n = Number of turns
α_{max} = Maximum deflection (degrees)

A mandrel of this diameter will give a spring rate error of approximately 1%.

The Stress vs Torque graph for the four gauges and different mandrel diameters highlighted various trends. The driven end of the spring exhibits a higher stress when deflected over a constant angle and thus corroborates the fact that torsion springs tend to fail at the driven end.

As the mandrel diameter reduces gauge 2 sees a dramatic rise in stress and becomes the most highly stressed position. The spread of stress results from gauge to gauge also widens considerably as the mandrel reduces, due to the increase in coil shearing. Coil shearing increases the stress to gauges 1 & 2 and reduces the stresses on gauges 3 & 4.

The shearing of the coils associated with the smaller mandrels reduces the torsional spring rate of the spring. It is surmised that this effect would be reduced in a spring with more active coils. However the end stress effects as described above would still be present in the first active turn at each end of the spring regardless of the number of turns.

Using the spring in the Unwind direction dramatically reduced the spring rate, due to reduction in coil support and increased shearing. The unwind rate measured was 19% lower than the wind up direction.

The effect on stress was considerable indicating the stress at gauges 1 & 2 were 24 to 27% higher than the CAD uncorrected stress. The CAD correction factor calculates as 8% suggesting that the standard correction factor is much too low for Unwind springs. It should also be noted that if the gauges had been mounted on the inside of the spring then the measured stress would be even higher due to the smaller inside diameter producing a higher strain.

6. Conclusion

1. The mandrel size directly affects the torsional spring rate.
2. A small mandrel reduces spring rate.
3. The mandrel size directly effects the stress distribution around the spring.
4. A small mandrel increases the maximum stress.
5. The optimum mandrel diameter may be calculated using the formula in EN 13906-3.
6. The driven end of the spring has slightly higher stresses induced.
7. Using springs in the unwind direction reduces the spring rate due to coil shearing.
8. Using springs in the unwind direction induces higher stresses due to coil shearing.
9. The Stress correction factor is too low for the unwind direction.