UNCORKED: THE FORCES FOR PULLING A CORK – LOAD RECONSTRUCTION

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SUMMARY

When modeling even the simplest of mechanisms, the need to understand real world loading is imperative for proper simulations. Traditional approaches may use company standards or even MBD simulations. However, these approaches are at best approximations of the actual load conditions. If using an approximation of loading in a simulation, the results are only as accurate as the approximation. Companies often use experimental load transducers to measure real world loading. There are two major disadvantages with this approach. The first is that the load transducers often cannot measure all the desired load inputs for a simulation. The second disadvantage is that the application of the load transducer forces physical modifications of the subject components. The measured loads may not be the actual loads because of the mass and stiffness changes to the subject components. Experimental Load Reconstruction is a non-invasive solution to this problem. The commercial True-Load® Load Reconstruction software will be demonstrated in this application. An unmodified corkscrew is used in this application with only uniaxial strain gauges applied to the corkscrew.

Presented will be an example of using True-Load Load Reconstruction technology to understand complex loading on an off-the-shelf corkscrew. This approach will involve the creation of an FEA model of the corkscrew. The corkscrew will have unit loads applied to the corkscrew. These unit loads will be used to construct a correlation matrix relating strain response to applied loading. The operation of the corkscrew will be used for pulling wine bottle corks. As will be seen, the loads will vary greatly from bottle to bottle.

The advantage in this approach is to get real world structural loads which can be used to drive accurate FEA simulations. Every test event is different; thus, the loading profiles are different. Traditional analytical techniques idealize the conditions and loading environment. The real world is complex, and every event is unique. True-Load loads provide real world loads for real world simulation.

1: Introduction

The following introduction is excerpted from a paper co-authored by this author which is sited in reference [9] with minor modifications.

A structure responds to external loads (or moments) imposed on it with changes in quantities, such as stresses and strains, displacements, kinematic deformations, etc. This paper addresses the problem of measurement of time varying loads acting on a component utilizing direct strain measurement on the structure. A linear relationship between the measured strains and the applied loads is created. The relationship, i.e., the transfer function between the applied loads and the measured quantity, can be established numerically (e.g., using finite elements), mathematically, or experimentally.

Kinematic response measurements using displacement transducers and accelerometers are well established and well documented [1]. An alternative approach involves measurement of strains using strain gauges [2]. The need to measure strains, stresses or other physical quantities is apparent since these are the ultimate concern of a designer interested in ensuring structural safety. Furthermore, since the gauges are relatively inexpensive, the use of strain gauges to measure dynamic forces acting on a structure has become quite popular in structural dynamics testing [2–6]. In these works, both the normal displacement modes and the strain modes are used to describe dynamic characteristics of the structure.

While the concept of modal strain was used in the mid-1980s to describe dynamic behavior of a structure, it was not until 1989 when Bernasconi and Ewins [3] presented a sound theoretical basis of modal stress/strain fields. The relationship between strain frequency response function and displacement frequency response function has also been explored by several authors [4–6]. While both the strain and displacement modes are intrinsic dynamic characteristics of a structure and correspond to each other, it has been noted in [6] that for sensitivity reasons, strain modal analysis is more useful in dynamic design of structures with features such as holes, grooves, and cracks.

To illustrate the use of strain gauges for recovery of dynamic loads, many of the works mentioned above considered a simply supported cantilevered beam on which gauges were located in an ad hoc manner. While the gauge location on a straight cantilevered beam may be intuitive under certain loading conditions, the same cannot be said of a complex structure where a trial-and-error approach to gauge placement can result in poor load estimates. This is because the gauge may be placed at a location where it has a relatively low sensitivity to the load(s) to be estimated. Further, for multi-degree of freedom force gauges, the cross-sensitivity [7] between the gauges may not be small. As a result, the strain data obtained from many of the gauges may be of little use, and the load estimates may not be precisely known.

For static loads, the influence of gauge locations and orientations on the quality of load estimates is discussed in [8]. However, in this work, it was noted that an analysis of all possible combinations of gauge placements would be too time-consuming for most problems. Consequently, only a few ad hoc groups of gauges were selected for analysis. If all possible gauge locations and orientations are not analyzed, the results are not guaranteed to be optimal, which in turn, may not yield the best possible load estimates.

To overcome the shortcomings mentioned above, Dhingra, et al [9] outlines an approach for formulating and solving the gauge placement problem when the imposed loads being estimated induce vibrations in the structure, resulting in time varying dynamic strains. The accuracy of load estimates is dependent on the placement (location and orientation) of the strain gauges, and the number of strain modes retained in the analysis. A sequential exchange algorithm based approach [12,13] is used to select the optimum locations, and angular orientations of the strain gauges. This paper presents the application of this technique to a two corkscrew complete with experimental measurements and comparison of simulated results to measured quantities.

2: Mathematical Foundation

Load reconstruction works on structures that behave linearly during the event of interest. The structure can undergo non-linear behavior prior to or after the event of interest. The term linear in this context means that the strain response is proportional to the applied loading. Portions of the structure may behave non-linearly. For example, local yielding near welds, bolted joints or boundary conditions may undergo non-linear strain response. Load reconstruction will continue to be effective if the nominal portions of the structure undergo linear response to the applied loading. Structures with gross yielding will not be appropriate for load reconstruction. Schematically, the concept of linearity can be illustrated as follows (Figure 1):

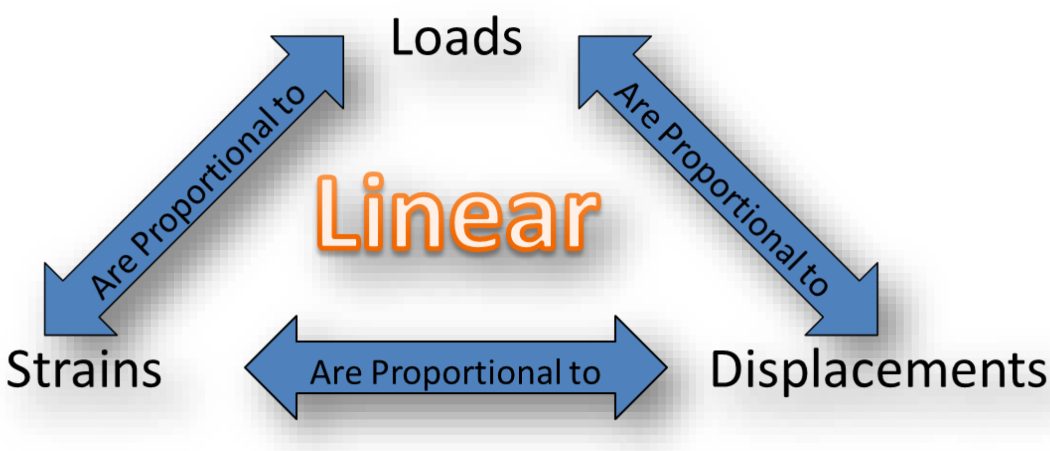


Figure 1: Linear material behaviour schematic

This linear relationship can be represented mathematically as follows:

$$F = Kx$$

Equation 1: Hooke's Law

and

$$\varepsilon C = F$$

Equation 2: Influence Coefficient Equation

onstructing a relationship for the strain equation that would work with fixed strain locations (e.g., gauges) and a series of loads cases will yield:

$$\begin{bmatrix} \varepsilon_{1,1} & \varepsilon_{1,2} & \vdots & \varepsilon_{1,m} \\ \varepsilon_{2,1} & \varepsilon_{2,2} & \vdots & \varepsilon_{2,m} \\ \cdots & \cdots & \ddots & \vdots \\ \varepsilon_{n,1} & \varepsilon_{n,2} & \cdots & \varepsilon_{n,m} \end{bmatrix} [C_{m\,x\,n}] = \begin{bmatrix} F_1 & 0 & 0 & 0 \\ 0 & F_2 & 0 & 0 \\ 0 & 0 & \ddots & 0 \\ 0 & 0 & 0 & F_n \end{bmatrix}$$

Equation 3: Influence Coefficient Equation Matrix Form

In the above equation the strain matrix [ε] has dimensions of n loads by m gauges. The load matrix [F] on the right hand side has dimensions of n loads by n loads. The matrix of proportionality [C] then must have dimensions of m gauges by n loads.

Each row in the strain matrix represents the strain values at a set of specific locations and orientations in the FEA model. The values in each row represent the strain response due to the corresponding load case. The columns of the strain matrix represent individual uniaxial gauge strain response. In the construct presented above, the loading matrix has been diagonalized. In general, this is not necessary, but for the developments presented here, it is convenient. Furthermore, the diagonal entries in the force matrix represent scalar multiples of the corresponding load cases. For our purposes we will set the scalar multiples to unity. This will then yield:

$$\begin{bmatrix} \varepsilon_{1,1} & \varepsilon_{1,2} & \vdots & \varepsilon_{1,m} \\ \varepsilon_{2,1} & \varepsilon_{2,2} & \vdots & \varepsilon_{2,m} \\ \dots & \dots & \ddots & \vdots \\ \varepsilon_{n,1} & \varepsilon_{n,2} & \dots & \varepsilon_{n,m} \end{bmatrix} [C_{m\,x\,n}] = [I]$$

Equation 4: Influence Coefficient Equation set to Unity.

Then to solve for C, a simple pseudo inverse needs to be constructed.

$$[C] = [\varepsilon^T \varepsilon]^{-1} \varepsilon^T$$

Equation 5: Correlation Matrix

The matrix C exists for a very large possible choices for strain gauge locations. The C matrix is optimal and most stable when the determinant of the self-projected strain matrix is maximum. A sequential exchange search algorithm is deployed that looks for the gauge locations that maximize this determinant.

Once the C matrix is calculated, loading profiles can be back calculated. Given vectors of strains collected from the test structure, the loads can simply be calculated via:

$$\begin{bmatrix} \varepsilon_{t_1,1} & \varepsilon_{t_1,2} & \vdots & \varepsilon_{t_1,m} \\ \varepsilon_{t_2,1} & \varepsilon_{t_2,2} & \vdots & \varepsilon_{t_2,m} \\ \dots & \dots & \ddots & \vdots \\ \varepsilon_{t_{end},1} & \varepsilon_{t_{end},2} & \dots & \varepsilon_{t_{end},m} \end{bmatrix} [C_{m\,x\,n}] = \begin{bmatrix} F_{1_{t_1}} & F_{2_{t_1}} & \dots & F_{n_{t_1}} \\ F_{1_{t_2}} & F_{2_{t_2}} & \dots & F_{n_{t_2}} \\ \vdots & \vdots & \vdots & \vdots \\ F_{1_{t_{end}}} & F_{2_{t_{end}}} & \dots & F_{n_{t_{end}}} \end{bmatrix}$$

Equation 6: Time domain expansion of Forces

The strain matrix on the left hand side of the above equation represents strain gauge values (columns) at each point of time of data collection (rows). This is the strain data that has been collected from a test event. The right hand side of the equations represents a set of vectors for scaling each load case. If the individual load cases are scaled by each vector and the results are linearly superimposed, then the resulting strains at the gauge locations at the corresponding row in the test strain matrix are guaranteed to match. Furthermore, any other response in the structure that is behaving linearly will be available through this superposition.

3: Solution Procedure

Summarized next are the steps involved in the recovery of dynamic loads acting on a component which has a finite number of strain gauges located on the component to measure time varying strains.

1. Create a series of unit load cases on the FEA model that represent locations and directions of loads applied to the structure. These loads are unit loads (e.g., 1KN) and should be designed such that if they were linearly superimposed on the structure, they could approximate the operating loads. Solve the FEA model for the unit loads constructed in this step.
2. Search the structure for optimal strain gauge placement using the technique referred to in the introduction. Store the correlation matrix to disc. For the purposes of this paper, this was accomplished using Wolf Star Technologies' True-Load/Pre-Test software.
3. Place the strain gauges on the physical part and measure time histories of strain in operation.
4. Calculate the time varying loads using Equation 6.

This process can be summarized with the following diagram:

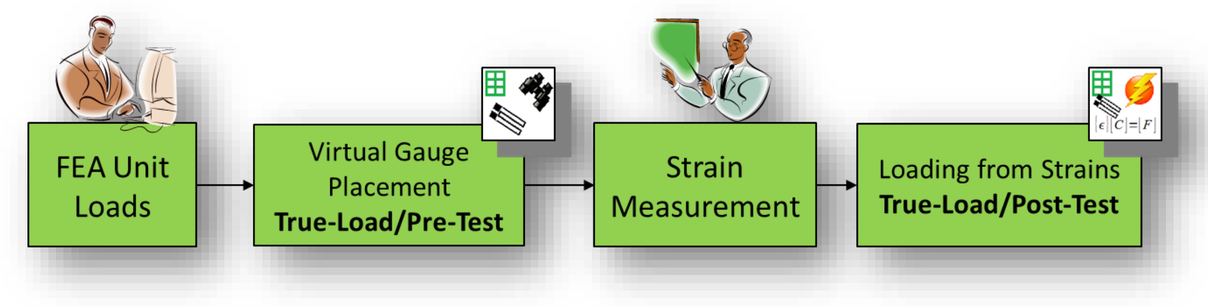


Figure 2: Load Reconstruction Process Schematic

4: The Corkscrew: Problem description

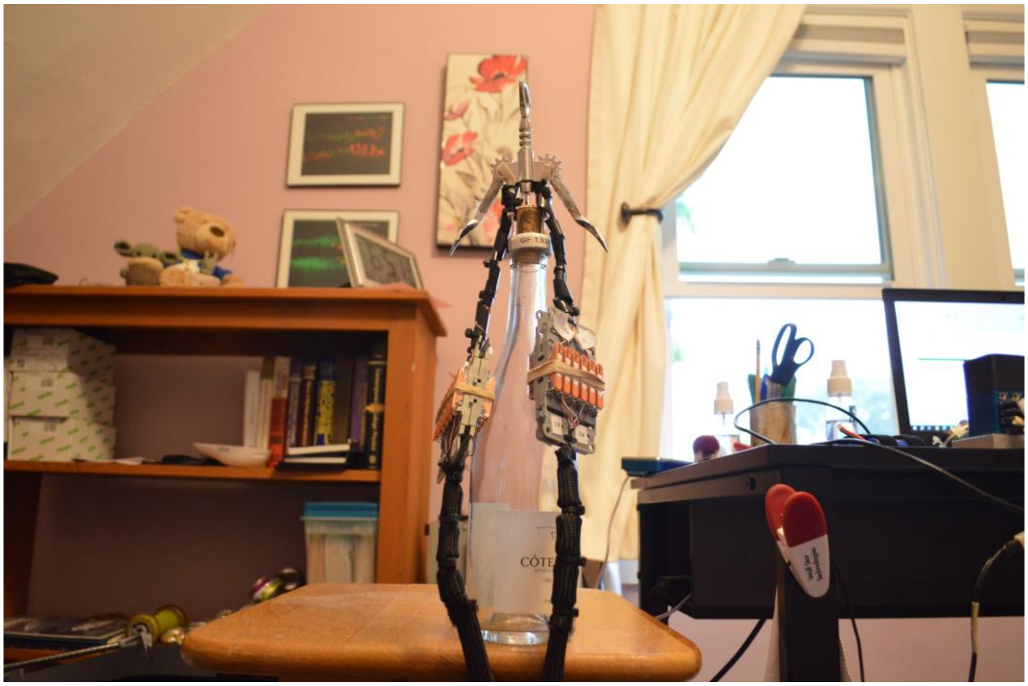


Figure 3: Corkscrew being tested.

This exercise will recover the loading on a corkscrew (Figure 3). The loading scenario will be a normal procedure for opening a bottle of wine. Note that the test articles have been bottles that have been re-corked. The 3D model of the corkscrew was created from 3D scan data provided courtesy of Milwaukee School of Engineering.

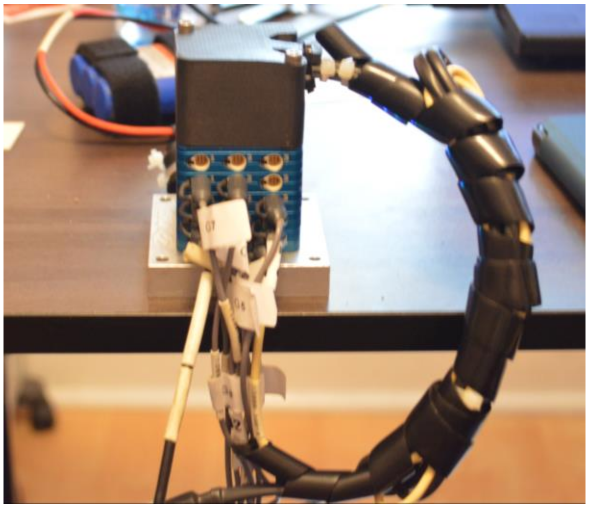


Figure 4: DTS Slice Micro DAQ

The DAQ system being used is a DTS Slice Micro DAQ with 8 channels of ¼ bridge strain sensing (Figure 4). The unit is powered by a small battery. Data is downloaded via USB cable. The strain gauges used were Micro Measurements C5K-060S5145-350-33F strain gauges. These gauges are 0.76 x 1.76 MM gauges (matrix: 3.1 x 2.6 MM) with pre-soldered lead wires. Three inch lengths of the lead wires are unshielded. The short lead wires were attached to the shielded cabling of the DAQ system to minimize external electronic noise. In order to protect the gauges, the gauges are coated with RTV (Figure 5). RTV is a room temperature vulcanizing silicone sealant used to seal electrical wiring in automobiles and other applications. The wires for the gauges are routed through the hinge point of the corkscrew arm to minimize wire movement.

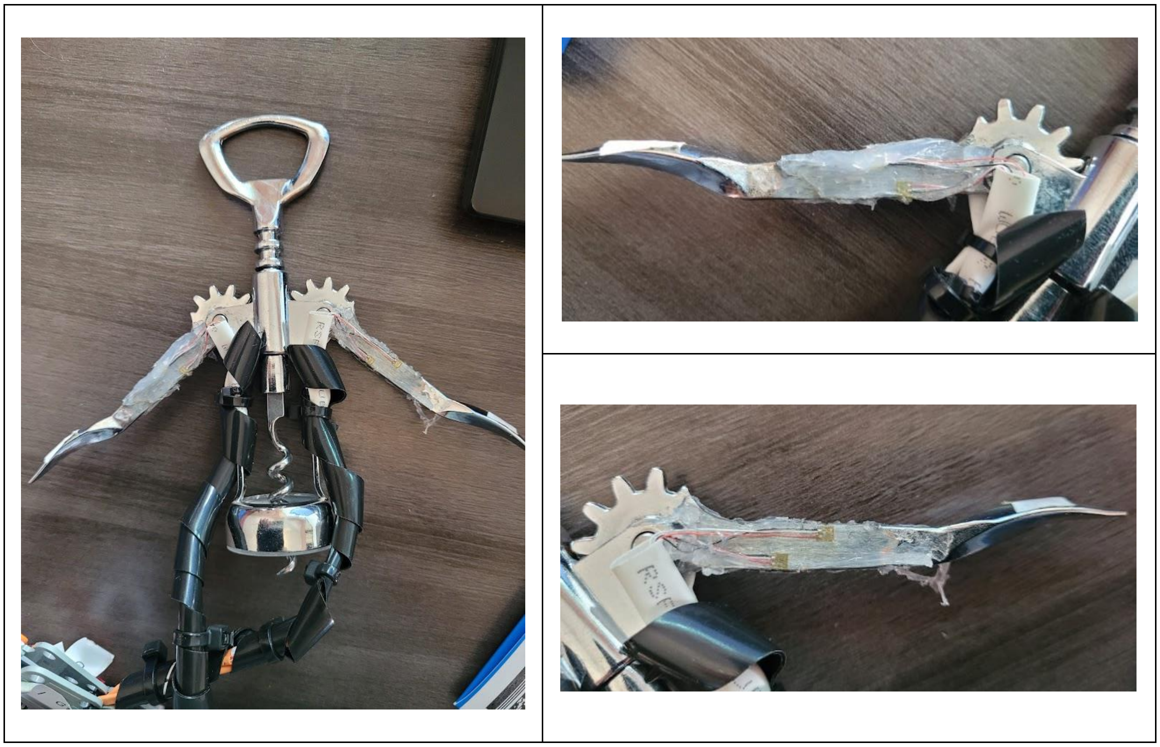


Figure 5: Strain Gauge Placement

5: The Corkscrew: Unit Loads

The unit loads for the Corkscrew are created in a Siemens SimCenter model. The FEA solver was Siemens NX Nastran. Figure 6 shows the unit loads are applied at center of the face of the arm handle (FY, FZ).

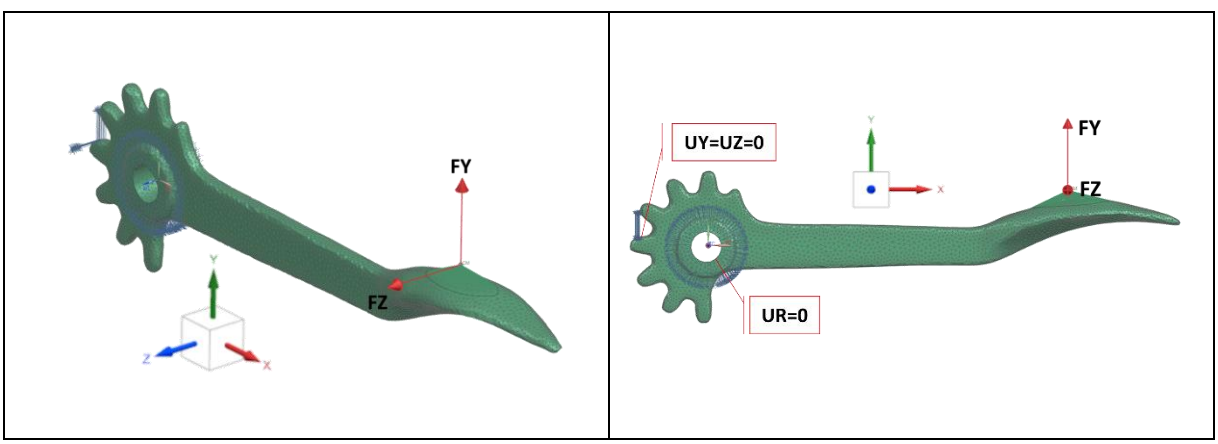


Figure 6: FEA Unit Loads

The center of the corkscrew is restrained in a radial direction (UR=0). One gear tooth of the handle was restrained (UY=UZ=0). The gear tooth restraint is an approximation; however, this should have minimal to no influence on the resulting load calculations.

6: The Corkscrew: Pre-Test

The True-Load/Pre-Test software was used to load in the two unit load cases and the corresponding strain results from the FEA model. The GUI from the True-Load software is shown below with the table of the unit load cases loaded (Figure 7).

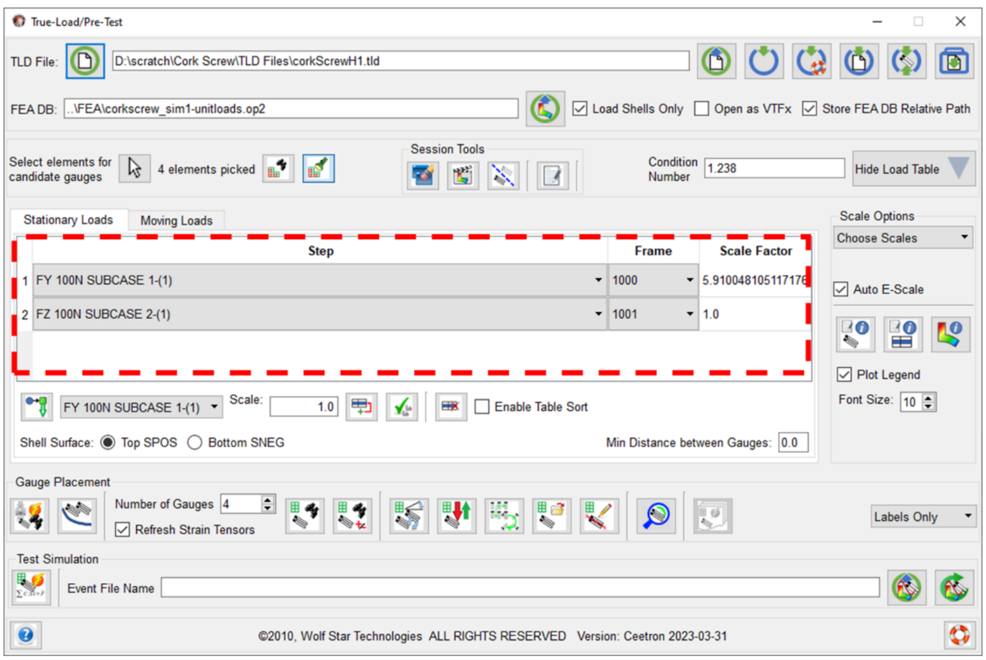


Figure 7: Pre-Test GUI with Load Table

The final strain gauge placement is shown below. The gauges on the fore / aft sides of the arm are mirrored about the central plane (Figure 8). Since there are two arms, there are two True-Load (TLD) files created, one for each arm. Each arm has four strain gauges to be sensitive to two loads. A total of eight strain gauges will be used in testing, four on each arm.

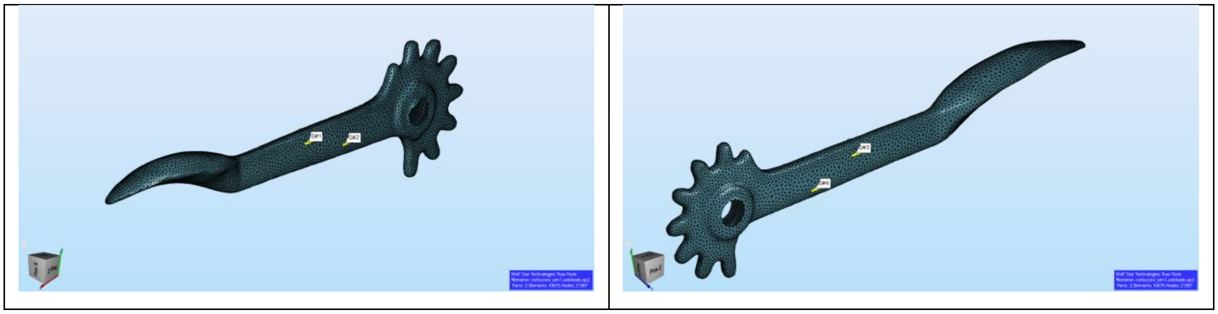


Figure 8: Virtual Strain Gauge Placement

An important phenomena to understand is the stability of the correlation matrix. The True-Load software provides a utility that calcuates the ideal strain for each unit load case and then applies a 5% random signal noise to the idealized strain. These strain signals are then multiplied by the correlation matrix to determine the correpsonding load response. Ideally, each load should be turned on one by one and the other loads would be turned off. Figure 9 below shows the load sensitivty to strain noise for this configuration of gauges. Also shown in this plot is the Load Assurance Criteria (LAC Matrix). Ones on the diagonal and near zeros on the off diagonal indicate tolerance to modest amount of signal noise and that the gauges chosen comprise a stable system of equations.

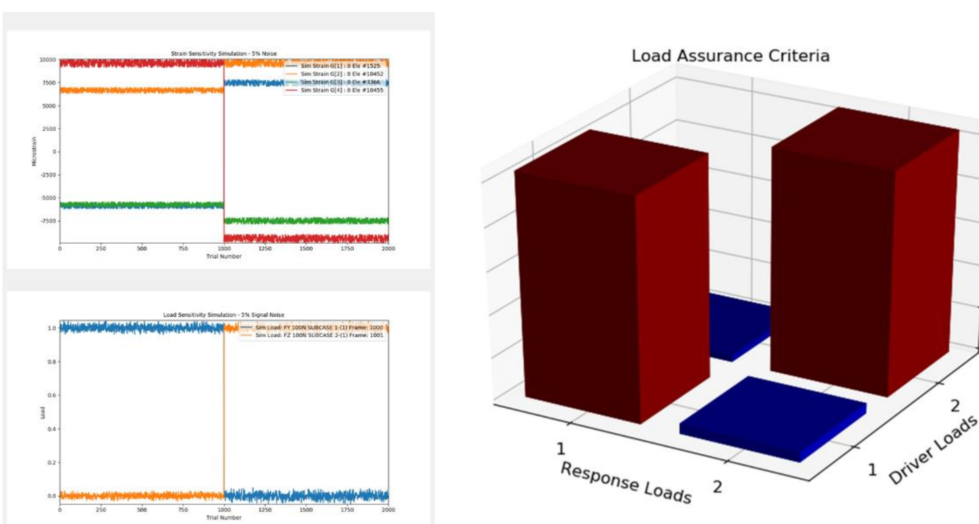


Figure 9: Load Sensitivity to Strain Signal Noise

This plot shows that the system of gauges chosen produces a very stable system of load reconstruction which can tolerate noise in the strain signals.

7: The Corkscrew: Strain Gauge Application

A series of drawings were created which located the strain gauges on the physical structure (Figure 10). These drawings were then used to place the gauges on the physical part using calipers and other measurement techniques.

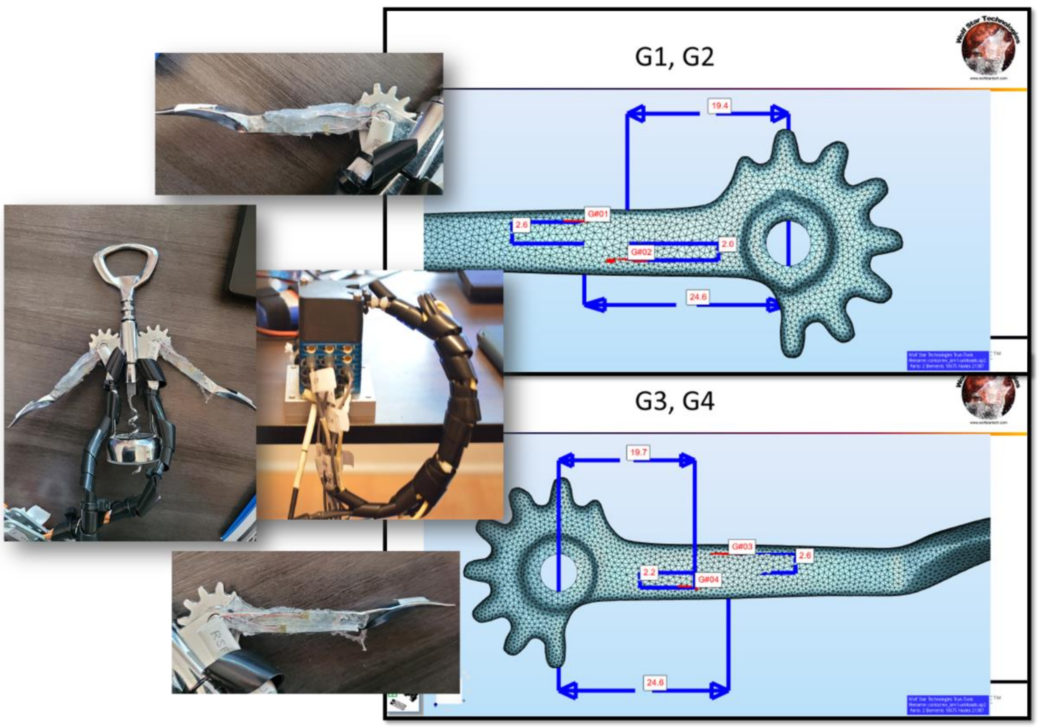


Figure 10: Strain Gauge Installation

8: The Corkscrew: Calibration

Since the strain gauges on this corkscrew are very small -- 3.1 x 2.6 MM (0.122 x 0.102 in), and the geometry on the arm has no well-defined fixed datums, it was decided to go through a calibration procedure to check and update the virtual strain gauge placement due to positioning errors in test. The calibration procedure involved a 5 Lbf bar bell dead weight. The 5 Lbf dead weight has an actual weight of 5.122 Lbf.

The corkscrew device was mounted in a bench vice with the arms of the corkscrew in a horizontal position (Figure 11). The 5.122 Lbf bar bell was hung from the horizontal corkscrew using nylon strapping.

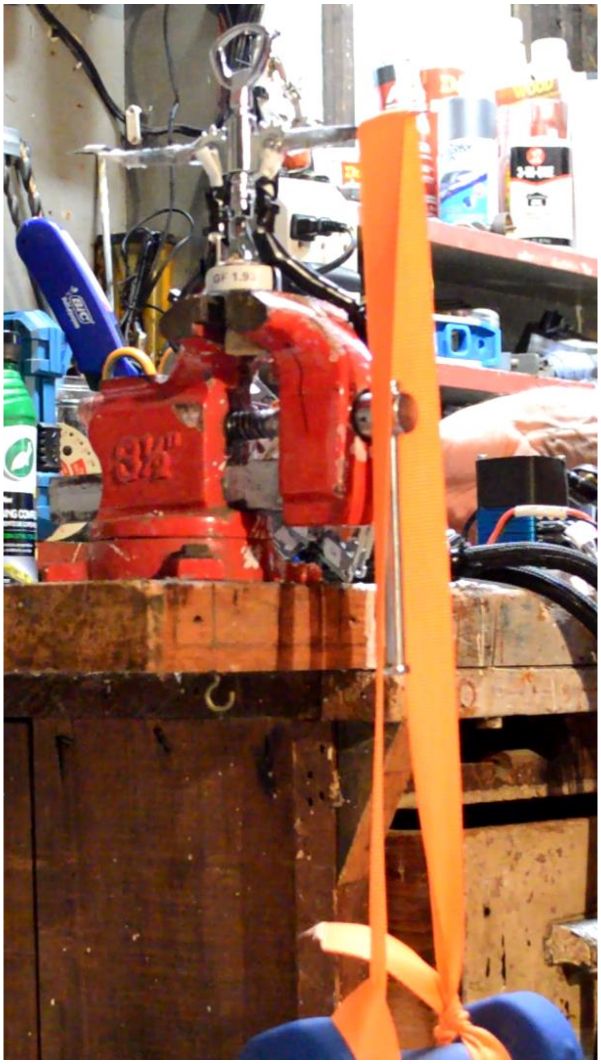
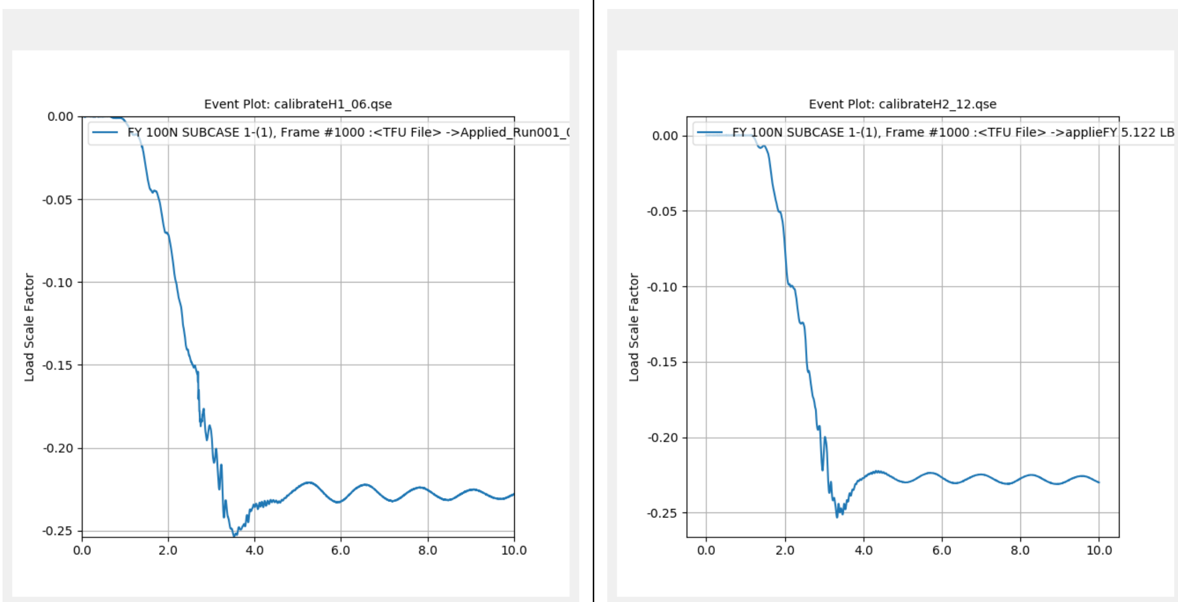


Figure 11: Corkscrew dead weight calibration test.

Using the utilities provided by True-Load's True-QSE software, a Quasi Static Event (QSE) was created to apply the 5.122 Lbf load the way it was in test. This involved manipulation of True-Load/Post-Test calculation results which will not be detailed in this paper. Figure 12 shows the resulting load profiles that were created for each arm.



Arm 1 | Arm 2

Figure 12: Calibration Load Profiles

The True-QSE software provides a utility to compare virtual gauge response to test data. Figure 13 shows the study performed on Arm 1. The red curves in the plot indicate the "best" virtual gauge, the blue lines in plot represent the as design location for the virtual gauge. The green curves represent the measured strain. Note that there still remains error due to the out of plane loading created by the oscillation of the dead weight. It should also be noted, that Gauge 1 on arm 1 was damaged and is non-functioning and thus excluded from the plots.

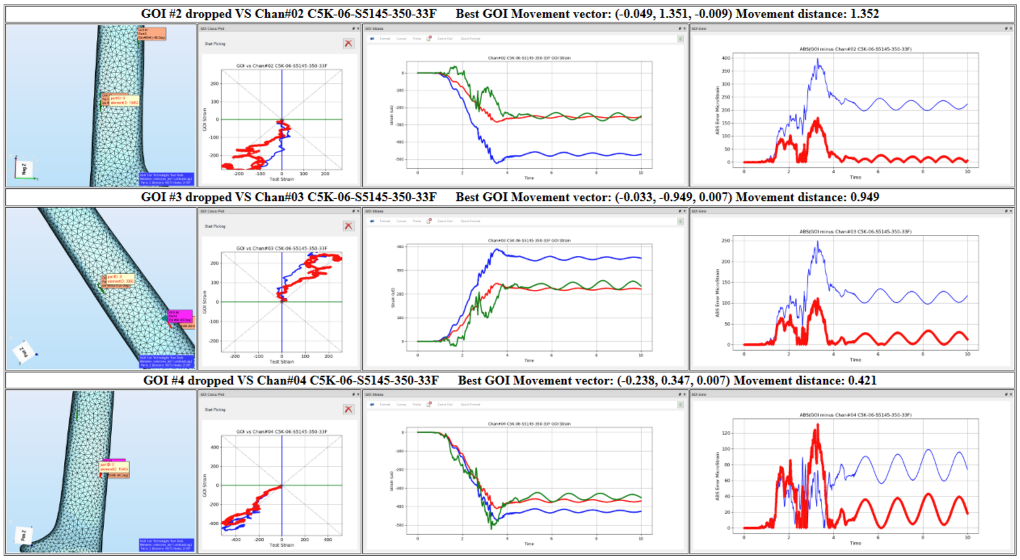


Figure 13: Best Gauge search to match Test Data

The True-Load (TLD) files were updated with these new locations and a confirmation calibration run was performed to ensure the proper dead weight was back calculated. At this point the two arms are ready for the in use test on the wine bottles.

9: The Corkscrew: Test Data Collection

Once the corkscrew was fully instrumented and calibrated, the strain gauges were connected to a DTS Slice Micro DAQ system (Figure 14). The strain data was sampled at 1000 samples per second.

Figure 14: DAQ used for Strain Data Collection

The data collection was performed while opening several bottles of wine. A typical trace of strain data is shown below.

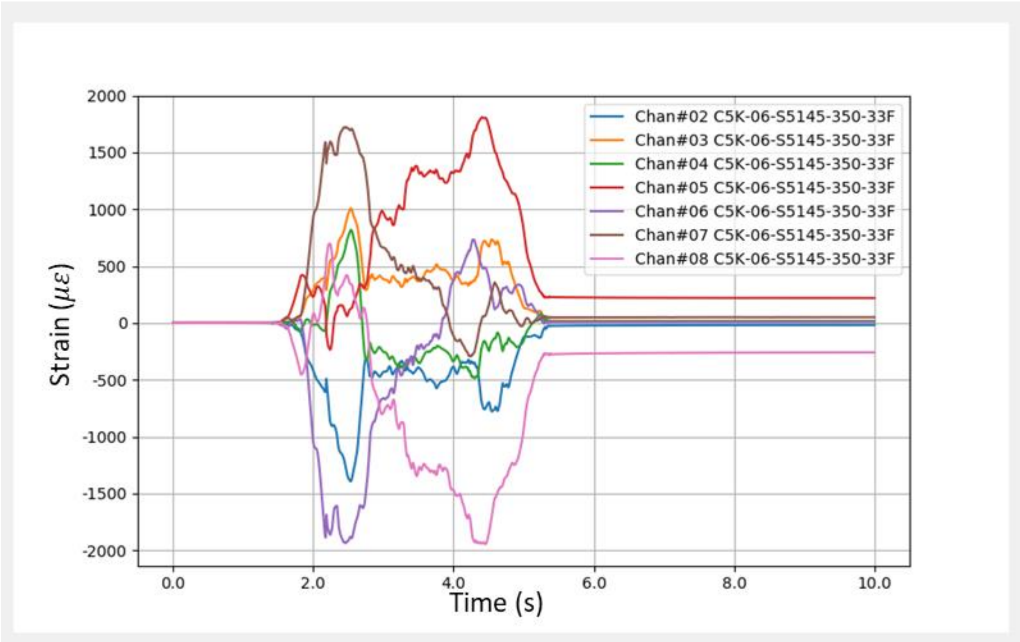
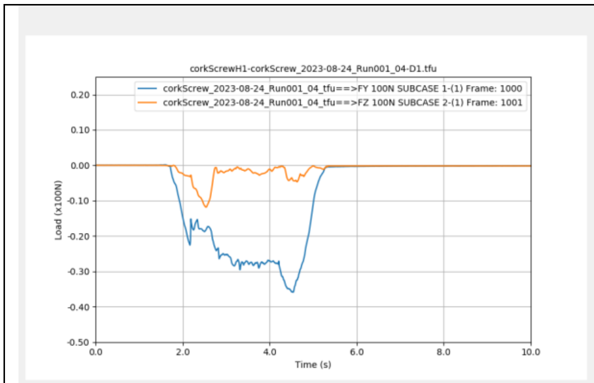


Figure 15: Typical Strain Traces from Test

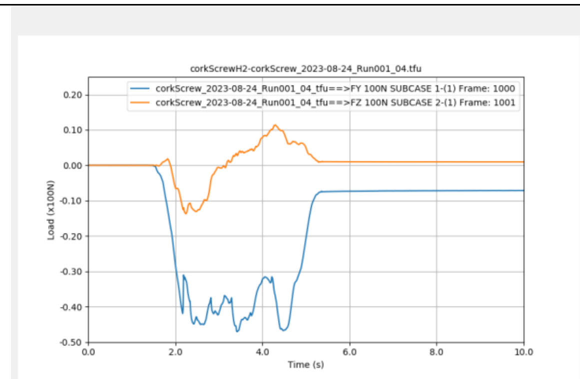
10: The Corkscrew: Post-Test

Once the strain data has been collected, it is processed to reconstruct the applied loading to the system. This is done by multiplying the measured strain data times the correlation matrix extracted from the FEA model. The result will be a time history of loading scale factors for each of the applied loads to the corkscrew.

As can be seen from the load plots (Figure 16) arm 2 is experiencing about 48 N (10.8 Lbf) and arm 1 is experiencing about 35 N (7.9 Lbf). Also, it is apparent from these traces that arm 1 is being in the negative Z direction whereas arm 2 is being pushed and pulled in the Z direction.



Arm 1 Loads



Arm 2 Loads

Figure 16: Reconstructed Loads

Three different corks on three different bottles were tested. As would be suspected, the loads for each bottle are different. Some of the corks were very tight in the bottle and others were easier to pull. Figure 17 shows how these loads differ from side to side and run by run.

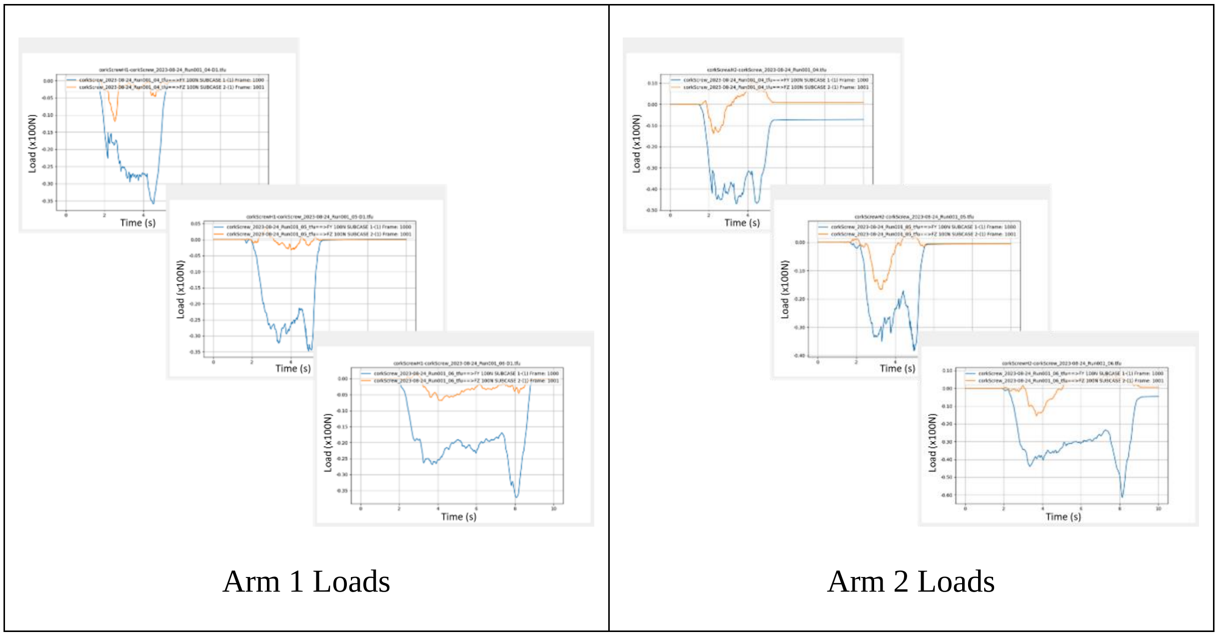
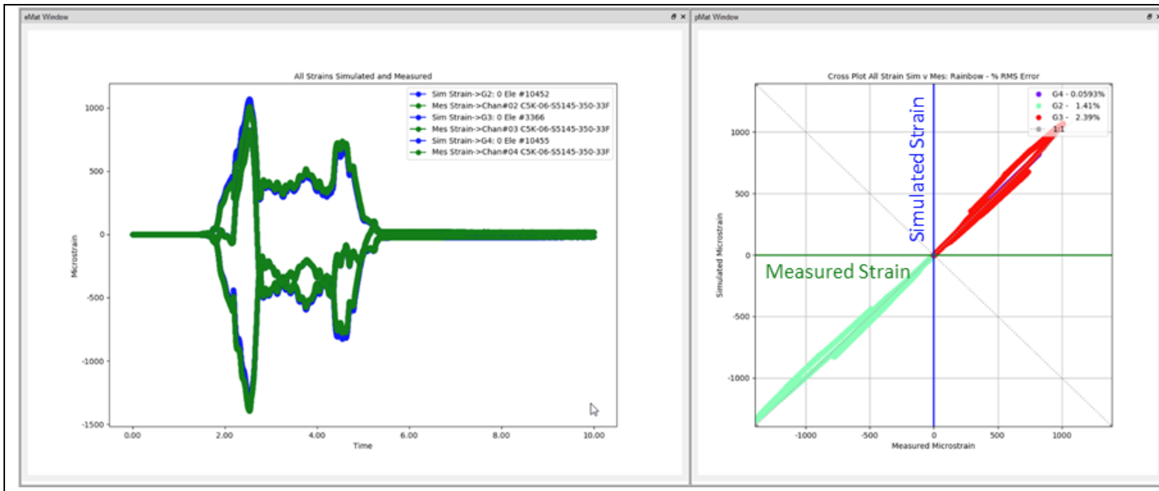
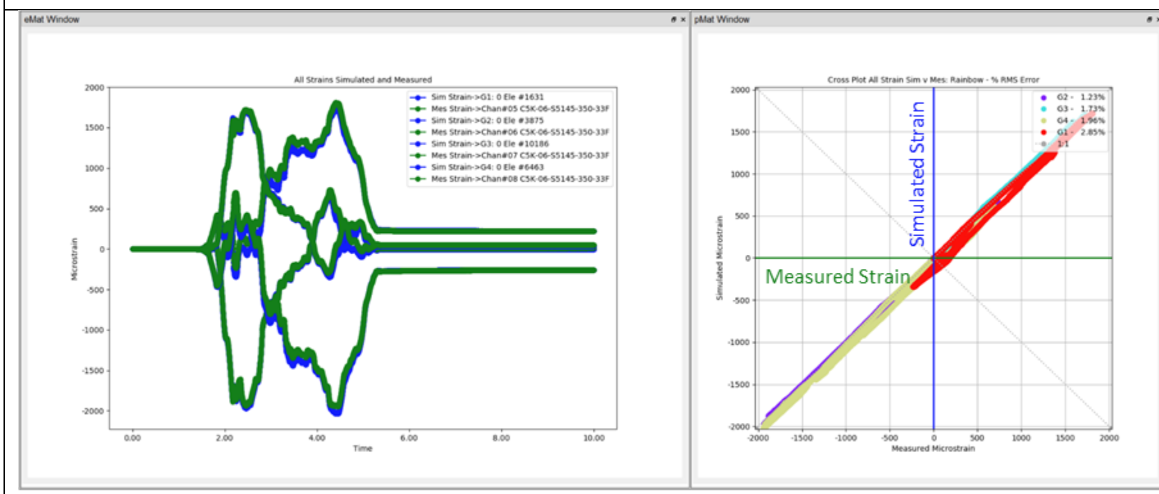


Figure 17: Varying Loads over Multiple Tests

For this application, the True-Load/Post-Test software was used to perform this load reconstruction. In addition to the load reconstruction, several automatic post processing tasks are performed. This will produce an HTML report which contains plots of the reconstructed loads and a set of plots showing the measured strain and simulated strain from the reconstructed loads at the strain gauge locations in the FEA model (Figure 18). These measured / simulated strain plots are summarized in an overall plot of the simulated strain (blue) and the measured strain (green). In addition, there will be a cross plot of simulated vs measured strain. Ideally this would be a perfectly straight line on a 45 degree angle. Note that in the cross plots of strain the maximum error in strain correlation is reported as 2.85% RMS error.



Arm 1 Strain Correlation



Arm 2 Strain Correlation

Figure 18: Overall Strain Correlation Plot

11: The Corkscrew: Post-Test

Once there is confidence in the reconstructed loads, detailed post processing of the FEA model may be performed. Having a complete time history of loads it is possible to construct operating deflection shapes (ODS) of the corkscrew utilizing the time history of loading and the FEA model. Below is a typical plot of an arm from an operating deflection shape on the corkscrew.

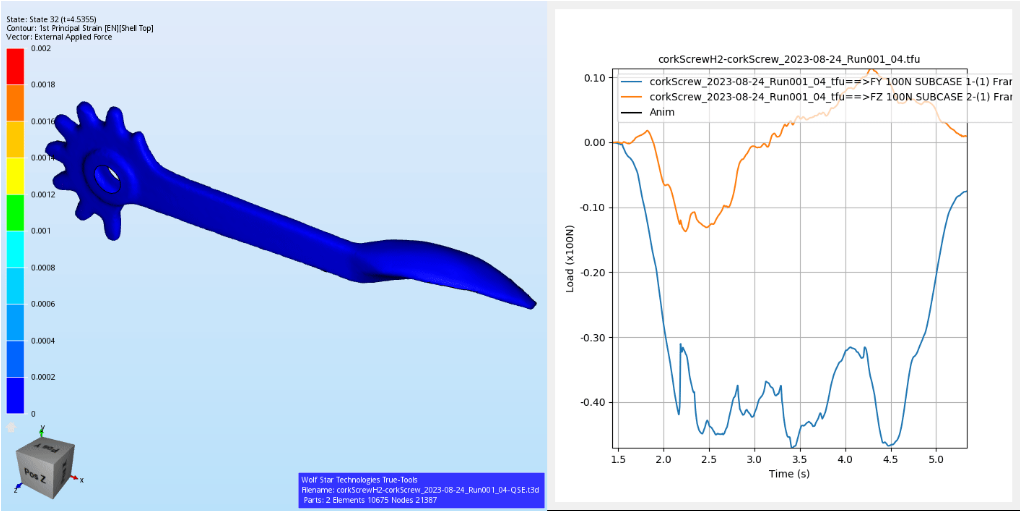


Figure 19: Typical Reconstructed Operating Deflection Shape

12: Conclusion

It has been shown in this paper that complex / nonlinear loading on a structure can be recovered at very high accuracy. The loading DOF were moderately complex (FY, FZ) on the two handles to make this a non-trivial problem. If traditional load measurement techniques were to be deployed, the corkscrew would have been rendered inoperable. A 2 DOF load transducer could perhaps be reasonably applied at the at the handles. However, a traditional load transducer would render the corkscrew unusable in a normal fashion.

With moderate skill and test plan processes, efficient placement of strain gauge can be placed on the structure to back calculate virtually any load conceived of by the FEA analyst. The cost for calculating these loads is two uniaxial strain gauges per loading DOF which is approximately $20. This is a highly cost effective and efficient process for determining complex loading on structures.

REFERENCES

[1] Ewins, D. J., 2000, Modal Testing: Theory, Practice, and Applications, Research Studies Press Ltd., Baldock, England.

[2] Hillary, B., and Ewins, D. J., 1984, “The Use of Strain Gages in Force Determination and Frequency Response Function,” Proceedings of the 2nd IMAC, Or- 391 lando, FL, pp. 627–634.

[3] Bernasconi, O., and Ewins, D. J., 1989, “Modal Strain Stress Fields,” J. Modal Anal., 4(2), pp. 68–76.

[4] Li, D. B., Zhuge, H., and Wang, B., 1989, “The Principles and Techniques of Experimental Strain Modal Analysis,” Proceedings of the 7th IMAC, Las Vegas, NV, pp. 1285–1289

[5] Tsang, W. F., 1990, “Use of Dynamic Strain Measurements for the Modeling of Structures,” Proceedings of the 8th IMAC, Kissimmee, FL, pp. 1246–1251.

[6] Yam, L. H., Leung, T. P., Li, D. B., and Xue, K. Z., 1996, “Theoretical and Experimental Study of Modal Strain Analysis,” J. Sound Vib., 191(2), pp. 397-398

[7] Sommerfeld, J. L., and Meyer, R. A., 1999, “Correlation and Accuracy of a Wheel Force Transducer as Developed and Tested on a Flat-Trac Tire Test Sys- 399 tem,” SAE Paper No. 1999-01-0938, Detroit, MI.

[8] Masroor, S. A., and Zachary, L. W., 1990, “Designing an All-Purpose Force Transducer,” Exp. Mech., 31(1), pp. 33–35.

[9] Dhighra, A. K, Hunter, T. G., and Gupta, D. K, 2013, “Load Recovery in Components Based on Dynamic Strain Measurements”, ASME Journal of Vibration and Acoustics, 135(5), VIB-12-1056; doi: 10.1115/1.4024384

[10] Szwedowicz, J., Senn, S. M., and Abhari, R. S., 2002, “Optimum Strain Gage Application to Bladed Assemblies,” ASME J. Turbomach., 124(4), pp. 606–613.