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CHARACTERIZATION OF THE TORQUE LIMITS AND CLAMPING FORCE RELATIONSHIPS FOR SMALL STAINLESS STEEL SCREWS IN TENSILE LOADED JOINTS OF VARIOUS METALS

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ABSTRACT

This study originated during the design of ChemCam, a Laser Induced Breakdown Spectroscopy (LIBS) and imaging instrument being developed for NASA's Mars Science Lab Rover. The mission needs for miniaturization, reduced weight, high reliability, minimal use of thread locking compounds, and the ability to handle harsh environmental conditions dictated the use of small, high strength screws to be threaded into a variety of metal alloys including Be-S200f, Al-6061-T6, Mg-ZK60A-T5, and Ti-6Al-4V. The lack of a credible fastener torque database for small (#0 through #8) high strength stainless steel screws in various parent materials, led to the development of an experimental program to characterize the following:

a. The screw torque value versus angular rotation (which indicates yielding in the screw or parent material) as a function of screw diameter, screw head configuration, depth of thread engagement, type of parent material, type of surface treatment on parent material, presence of thread locking compound, repeatable threaded hole use, and degree of screw pedigree.
b. The relationship between fastener torque and clamping force for a subset of the above mentioned variables.

The database generated from this study will serve as a design reference for utilizing small stainless steel fasteners and provide trending information for other researchers who may be interested in broadening its range of parameters. This paper reviews the related fastener torque and clamping force information from the literature, describes the experimental screw torque and clamping force monitoring equipment, presents the test matrix and experimental procedures, and discusses the empirical results.

INTRODUCTION

The design of spacecraft instrumentation presents unique and difficult challenges to the Mechanical Engineer. Space mission requirements generally demand light, stiff, and high-strength structures that must be able to survive harsh environmental conditions while supporting cutting edge scientific measurements. These mechanical structures often push the technology limits on design, manufacturing, and assembly processes with the inclusion of exotic materials, complex geometries, delicate and small features, etc. One design aspect in particular, the bolted joint, can be extremely challenging for the spacecraft instrument designer. The thermal performance and structural rigidity of the instrument requires accurate knowledge of the clamping forces and associated torque requirements of relatively small screws. Insufficient clamping forces can lead to inefficient heat dissipation, structural degradation or failure, equipment misalignment, etc., all of which will have a negative impact on the operational performance of the instrument.

The recent design of ChemCam, a laser-induced breakdown Spectroscopy instrument for the NASA Mars Science Laboratory mission, provided the motivation for enhancing the understanding of the mechanics of small screws. The optical detection portion of the ChemCam instrument, shown in Figure 1, incorporated two precision optical subsystems and an array of densely populated electronics [1]. ChemCam's architecture relied on an array of metal alloys including Be-S200F, Mg-ZK60A-T5, Al-6061-T6, and Ti-6Al-4V and small fasteners ranging from #6 down to #0 screws. Large random vibration and pyroshock loads and efficient thermal dissipation paths necessitated the accurate understanding of the clamping force of the screwed interface joints within the instrument.

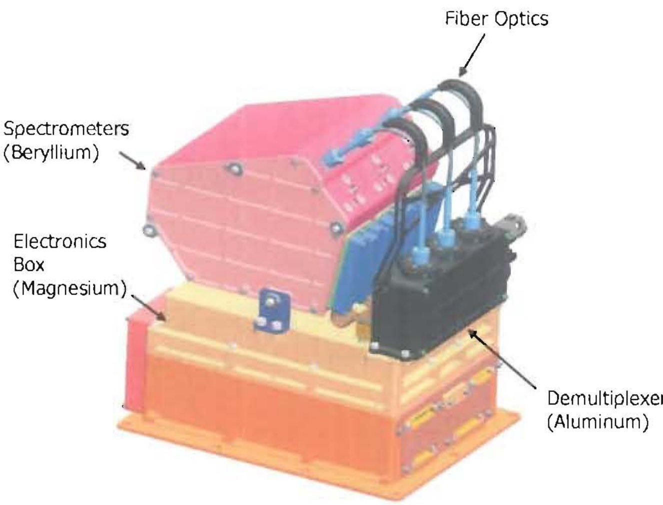

FIGURE 1 CAD MODEL OF THE OPTICAL DETECTOR PORTION OF THE CHEM CAM INTRUMENT.

Fastener Fundamentals

A bolted joint is one of the most common techniques used to assemble mechanical and electrical devices. The bolted joint allows parts to be configured in complex assemblies, disassembled for repair or replacement, adjusted for alignment, etc. The bolted joint is commonly used in a tensile or shear state where it provides a clamping force to hold two or more parts together. Furthermore, the bolted joint is frequently required to prevent relative motion, restrict leakage, or enhance electrical or thermal conductivity between mating parts. In all cases, the bolted joint relies on the clamping force that is provided by the fastener as it is stretched during the tightening process with a corresponding nut (bolt) or threaded hole (screw). The amount of clamping force or preload selected for a particular fastener is generally dictated by the mechanical design and functionality of the bolted joint, as well as the uncertainty in the amount of preload initially established and that maintained throughout the joint's lifetime.

The clamping force in a fastener is generally obtained indirectly by controlling the amount of torque applied to the head of the fastener or the body of the nut [2, 3, 6]. As torque is applied to a fastener, roughly 10% of the input work is stored as potential energy in the stretching of the fastener and joint material [2, 6], the remainder of the work is lost to friction in the threads and contact surfaces, bending or twisting of the bolt, or permanent deformation of the nut or parent material. Furthermore, of the 10% of the work that goes into stretching the bolt and creating the clamping force, some of this can be lost in relaxation as threads and deformations in the joint creep to a lower energy state under the pressure of the clamping force.

When employing fastener torque to establish sufficient clamping force, one must avoid exceeding the yield strengths of the fastener and joined materials, generating undesired distortions of the assembly, and providing insufficient fastener stretch that leads to loosening of the joint during the assembly lifetime. As outlined in [2], there are literally hundreds of variables that influence the behavior of the bolted joint, and many of them are difficult to control or characterize with sufficient accuracy. Some of the most common factors which influence fastener torque and clamping force include the diameter, profile, pitch, surface finish, material properties, and class of fit of the threads, the cleanliness or amount of lubrication on the threads, the size, shape, and finish of the interface between the fastener head and joint material, depth of thread engagement, alignment or contact area of the fastener hole, etc. As a result of these many uncertainties, bolted joints are rarely designed to an efficient and optimized configuration.

To aid the design engineer and assembly technicians, several general correlations and empirical databases have been developed for selecting a fastener torque value to generate the desired clamping force. For many assemblies, the fasteners should be tightened to produce an initial preload or tensile force, Ft, nearly equal to the proof strength of the fastener. This is represented mathematically as:

$$F_t = K\, A_t\, S_p \tag{1}$$

where At is the tensile stress area of the threaded portion of the fastener, Sp is the proof strength of the fastener material, and K is a constant usually specified in the range of 0.75 to 1.0 [3]. Proof strength is defined as the maximum stress level that does not produce a normally measurable permanent set in the material [3]. Different forms of Eqn. (1) can be found in the literature. Reference [6] replaces Sp, with the ultimate static tensile strength, Su, and employs a corresponding range of 0.7 to 0.8 for the constant K. Reference [5] also employs the ultimate tensile strength in Eqn. (1), but recommends a value of 0.5 for the constant K. Finally, reference [2] replaces Sp in Eqn (1) with the yield strength, Sy, and provides tabularized ranges of recommended values of K, depending on the application. While Eqn. (1) applies to the fastener, Reference [5] also proposes load limits for nuts, threaded inserts, and tapped holes. For the latter, the preload limit is associated with the shear strength of the tapped hole material, Ss, and is represented mathematically as

$$F_h = \pi\, S_s \frac{L_e d_{\min}}{P_t}\left[\frac{P_t}{2} + \frac{d_{\min} - d_{\max}}{\sqrt{3}}\right] \tag{2}$$

where Le is the length of thread engagement, Pt is the thread pitch, dmin is the minimum major diameter of the threaded fastener, and dmax is the maximum pitch diameter of the tapped hole. Since this present study did not employ nuts or threaded inserts in the testing, the reader is referred to reference [5] for additional information on these features. Reference [5] recommends employing the minimum preload value predicted with their variant form of Eqn (1) and Eqn. (2) when calculating acceptable fastener torque values.

It is known that within the elastic stress-strain region, the preload in a fastener and the installation torque required to achieve that preload, are linearly related [2, 3, 6]. References [2] and [3] present fundamental mathematical models which relate the fastener tensile force to the torque required to turn the fastener to achieve the tensile force. Complex long-forms of the resulting equations were reduced to the following simple equation to relate the screw tightening torque, T, to the initial tensile force, Ft, nominal major thread diameter, d, and a dimensionless constant N termed the nut factor ($0 < N < 1$), which accounts for the friction effects of the screw head and threads, the torsion or bending of the fastener, plastic deformation of the threads, etc,

$$T = N F_t d \tag{3}$$

The nut factor is a large unknown for most applications, as it is highly influenced by many of the thread parameters and material properties discussed above, as well as the rotational rate used to secure the fastener. Furthermore, its value can vary by ±30% for two seemingly identical fasteners being threaded into the same material [2]. Reference [2] provides an extensive summary of mean, minimum, and maximum nut factor values that were measured for a variety of fasteners, coatings, and parent materials. Although tables of nut factor and torque values exist in the open literature, it is highly recommended [2, 6] that for critical assemblies and applications, that fastener torque and clamping force tests be performed to determine the range and nominal values of N for a given set of conditions. Note that before the correct torque value can be selected, the initial assembly clamping force must be specified for a particular fastener and corresponding joint design. Determining the desired clamping force, the allowable deviation in this force, and the overall joint design, is beyond the scope of this manuscript. The reader is referred to [2] for more information on these topics.

The use of fastener torque monitoring for controlling clamping force has been documented and proven successful for some common fastener sizes. However, quite often the results are very application specific and many of the influential variables such as material composition of the fasteners and joint material, the class of fit and cleanliness of the threads, depth of thread engagement, etc., have not been reported, thus making it difficult to utilize the findings on other applications. Large discrepancies can also be found in the literature for the recommended torque values of certain fasteners. Many of these can be explained by the lack of attention to controlling the influential variables discussed above. Furthermore, very little information exists in the literature on recommended torque values for small fasteners in the size range of #0 through #8.

EXPERIMENTAL METHODS

Test Apparatus

A motorized torque and clamping force test stand, universal torque sensor, and torque/force gauges were used to produce all torque and clamping force measurements. The test apparatus, shown in Fig. 2, applies an increasing torque to a fastener and measures either the applied torque or resulting clamping force as a function of the angular rotation of the fastener. For all of these tests, a test fixture (described in the next section) containing fasteners and parent material bodies, is placed in a rotational stage and the corresponding fastener torque tooling is placed above it in a stationary tool holder chuck. The motorized rotation stage provides a constant rotational speed under load. The torque transducer is contained within the chuck housing and remains stationary since the rotation stage provides the applied torque. The torque sensor has a manufacturer quoted accuracy of +/- 0.35% full scale. The torque/force gauge has an accuracy of +/- 0.1% full scale. A computer running Labview software records all of the torque and clamping force data as a function of angular rotation of the stage.

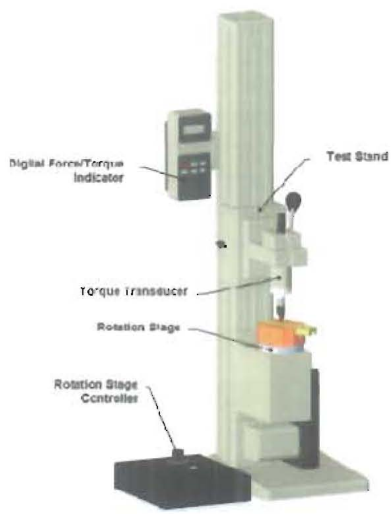

FIGURE 2 FASTENER TORQUE AND CLAMPING FORCE TEST APPARATUS

Test Fixtures

Torque: Figure 3 shows the torque testing mount and sample torque test threaded body which is used in the torque versus angular rotation measurements. The mount is fastened to the motorized rotational stage on the test apparatus and holds the torque test threaded body in place during th torque measurement testing. The torque test threaded body is in the shape of a "C" channel and has five to ten threaded through holes. Different thread sizes, threaded engagement lengths, and parent materials were produced with multiple versions of the torque test threaded body (refer to test matrix).

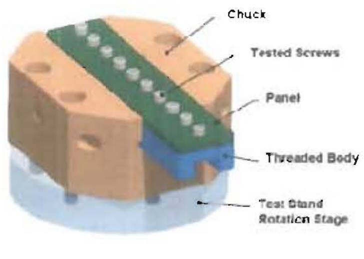

FIGURE 3 TORQUE TESTING MOUNT AND THREADED BODY

Clamping Force: Figure 4 shows a section view of the clamping force test mount which is used to measure the clamping force of a fastener as a function of angular rotation. This mount contains a load cell which has an accuracy of +/- 0.5% of full scale. The clamping force test mount is fastened to the motorized turntable in place of the torque testing mount described previously. A test screw is mounted through the replaceable disc and into the threaded plug. The lever arm rests on the top of the load cell such that there is an offset between the disc and plug. Applying torque the test screw drives the disc and plug closer together generating force on the load cell via the lever arm. The thread engagement length can vary as well as the parent material for a given replaceable disc and threaded plug.

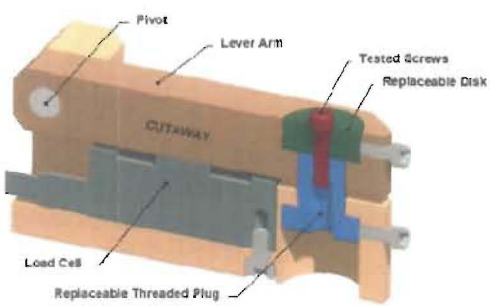

FIGURE 4 SECTION VIEW OF THE CLAMPING FORCE TEST MOUNT

Fasteners and Test Bodies: The fasteners tested in this study are detailed in Table 1. As can be seen in the table the main variable between fasteners of a given size was the head configuration. The various torque test threaded bodies, replaceable discs and threaded plugs are detailed in Tables 2 and 3. The main difference between threaded bodies for the same parent materials is the thread size or engagement length.

TABLE 1 TESED FASTENER LIST

Screw Size	Nominal Bolt Diam (in)	Thread Designation	Screw Head Config	Screw Material
0-80	0.060	UNC-2A	SHCS	SS302
0-80	0.060	UNC-2A	SHCS	SS 302HQ
1-72	0.073	UNF-3A	SHCS	SS 302
2-56	0.086	UNC-2A	SHCS	SS 302HQ
2-56	0.086	UNC-2A	But. Hd. SD	SS 316L
2-56	0.086	UNC-2A	SHCS	SS 302HQ
2-56	0.086	UNC-2A	SHCS	SS 302HQ
2-56	0.086	UNC-2A	Pan Hd. Phillips	UNS S30430
2-56	0.086	UNC-2A	SHCS	SS302HQ
2-56	0.086	UNC-2A	Pan Hd. Phillips	UNS S30430
2-56	0.086	UNC-2A	Pan Hd. Phillips	UNS S30430
2-56	0.086	UNC-2A	Flat Hd., 100 deg. Philips	UNS S30430
4-40	0.112	UNC-2A	Socket Drive	SS 303
4-40	0.112	UNC-2A	SHCS	SS 302HQ
4-40	0.112	UNC-2A	Set Screw Hex Dr.	SS 303
4-40	0.112	UNC-2A	SHCS	SS 302HQ
4-40	0.112	UNC-2A	Hex Head Vented	SS 18-8
6-32	0.138	UNC-2A	SHCS	SS 302HQ
8-32	0.164	UNRC-3A	SHCS	SS 302HQ

TABLE 1 THREADED BODIES USED IN TORQUE TEST MOUNT

Material	Coating	Screw Size	Thread Engagement Lengths (in)
Al 6061-T6	Chromate	0-80	0.06
			0.09
			0.12
		1-72	0.073
		2-56	0.086
			0.129
			0.172
Mg ZK60A-T5	Copper Strike	2-56	0.086
			0.129
		4-40	0.168
			0.224
		6-32	0.15
Be S200F		0-80	0.09
			0.12
		2-56	0.129
			0.172
		8-32	0.25
Ti-6Al-4V	None	0-80	0.09
			0.12
		2-56	0.129
			0.172
		8-32	0.25

TABLE 2 THREADED CODIES USED IN CLAMPING FORCE MOUNT

Material	Coating	Screw Size	Thread Engagement Lengths (in)
Al 6061-T6	Alodyned	0-80	0.09
			0.12
			0.12
		2-56	0.129
			0.172
Ti-6Al-4V	None	0-80	0.09
			0.12
		2-56	0.129
			0.172

Test Procedures

Torque Tests: The procedure for completing the torque tests started with hand tightening the test screws in the torque test threaded body. Each screw was then loosened by approximately ninety degrees so that a measurement of the running torque could be made. The torque test threaded body was then inserted into the torque testing mount. The driver appropriate for the test screws was inserted in the chuck and the chuck was lowered to engage the screw. The computer based data acquisition system was initiated and the motorized turntable was started. The motorized turntable and data acquisition system were stopped at the onset of ultimate failure as indicated by the real time torque vs. angular rotation curve displayed on the data acquisition system.

Clamping Force Tests: The procedure for completing the clamping force tests began by inserting a replaceable disc and threaded plug into the clamping force test mount. The disc and plug were then secured using the set screws on the mount. Next, a test screw was hand tightened through the replaceable disc and into the threaded plug. The screw was then loosened by approximately ninety degrees so that a measurement of the running torque could be made. The computer based data acquisition system was initiated and the motorized turntable was started. The motorized turntable and data acquisition system were stopped at the onset of ultimate failure as indicated by the real time force vs. angular rotation curve displayed on the data acquisition system.

Test Matrix

The test matrix used in this study was driven by the design requirements of the ChemCam instrument. Different fastener head configurations were needed due to physical clearance issues and assembly clearances. Due to the demanding design requirements of the ChemCam mission, several unusual parent materials were used in the instrument design, including Al-6061-T6, Mg-ZK60A-T5, Be-S200f, and Ti-6Al-4V. The Al-6061-T6 and Mg-ZK60A-T5 threads were coated with thin (<5 μm) chromate and copper coatings, respectively, to properly simulate the design conditions used with ChemCam. Tables 4 through 7 show the test matrices for the Al, Mg, Be, and Ti alloy threaded bodies respectively. These were limited quantity tests and were not intended to provide extensive statistical sampling.

TABLE 4 TEST MATRIX FOR Al-6061-T6 THREADED BODIES WITH CHROMATE COATED THREADS

Screw Size (Dia)	Length (in)	Thread Engagement (in)	# of Test Holes in Threaded Body	Panel Material and Thickness
1-72 (0.073)	0.187	0.073	5	Al-6061-T6, Alodyned, 0.060"
2-56 (0.086)	0.250	0.086	5	Al-6061-T6, Alodyned, 0.060"
2-56 (0.086)	0.250	0.129	5	Al-6061-T6, Alodyned, 0.060"
2-56 (0.086)	0.250	0.172	5	Al-6061-T6, Alodyned, 0.060"

TABLE 5 TEST MATRIX FOR Mg-ZK60A-T5 THREADED BODIES WITH COPPER COATED THREADS

Screw Size (Dia)	Length (in)	Thread Engagement (in)	# of Test Holes in Threaded Body	Panel Material and Thickness
2-56 (0.086)	0.188	0.086	5	Al-6061-T6, Ni Coated, 0.060"
2-56 (0.086)	0.188	0.129	5	Al-6061-T6, Ni Coated, 0.060"
4-40 (.112)	0.312	0.168	5	Al-6061-T6, Ni Coated, 0.060"
4-40 (.112)	0.312	0.224	5	Al-6061-T6, Ni Coated, 0.060"
6-32 (0.138")	0.25	0.150	5	Al-6061-T6, Ni Coated, 0.060"

TABLE 6 TEST MATRIX FOR Be-S200-F THREADED BODIES, UNCOATED

Screw Size (Dia)	Length (in)	Thread Engagement (in)	# of Test Holes in Threaded Body	Panel Material and Thickness
0-80 (0.060")	0.312	0.090	5	Al-6061-T6, Ni Coated, 0.060"
0-80 (0.060")	0.312	0.120	5	Al-6061-T6, Ni Coated, 0.060"
2-56 (0.086)	0.25	0.129	5	Al-6061-T6, Ni Coated, 0.060"
2-56 (0.086)	0.25	0.172	5	Al-6061-T6, Ni Coated, 0.060"
8-32 (0.164")	0.375	0.25	5	Al-6061-T6, Alodyned, 0.060"

TABLE 7 TEST MATRIX FOR Ti-6Al-4V THREADED BODIES UNCOATED

Screw Size (Dia)	Length (in)	Thread Engagement (in)	# of Test Holes in Threaded Body	Panel Material and Thickness
0-80 (0.060")	0.312	0.090	5	Al-6061-T6, Ni Coated, 0.060"
0-80 (0.060")	0.312	0.120	5	Al-6061-T6, Ni Coated, 0.060"
2-56 (0.086)	0.25	0.129	5	Al-6061-T6, Ni Coated, 0.060"
2-56 (0.086)	0.25	0.172	5	Al-6061-T6, Ni Coated, 0.060"
8-32 (0.164")	0.375	0.25	5	Al-6061-T6, Alodyned, 0.060"

RESULTS

Torque Test Results

The torque tests described in Section 3.c.i resulted in data measurements of torque applied to the fastener head versus rotation angle of the fastener. Figure 5 shows a typical data plot of torque vs. rotation angle, as well as the slope of the torque vs. rotation angle curve. The inflection point on the torque vs. rotation angle is accurately identified as the maximum point on the slope curve. This inflection point corresponds to the onset of yielding of the fastener [2]. Beyond this point on the curve some part of the threaded connection is beginning to yield up to the point of ultimate failure, which is represented by the rapid decline in the torque value with increasing rotational angle. The torque value corresponding to the inflection point, noted as the yield torque, is given in the following tables and figures for the various tests.

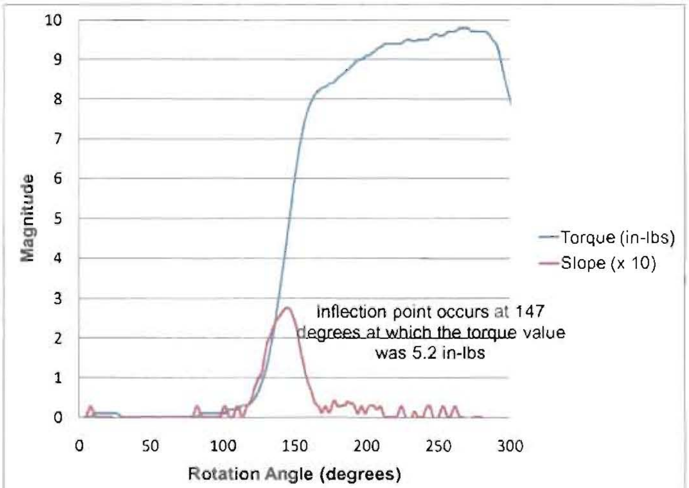

FIGURE 5 SAMPLE TORQUE VS. FASTENER ROTATIONAL ANGLE AND SLOPE CURVE

Due to limited resources the same torque test bodies were used for both the pedigreed and non-pedigreed screws. The pedigreed screws were tested first. The results show that the pedigreed screws have a minimum of 14% higher yield torque than the non pedigreed screws (Figure 6). Also of note is that for the case of 0.172" thread engagement case the non-pedigreed fasteners have more than twice the yield torque variation as the pedigreed fasteners. These results made it clear that pedigreed fasteners should be used wherever possible.

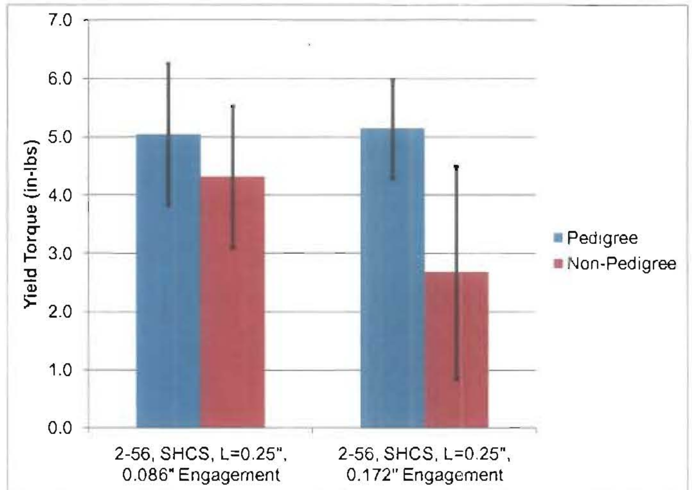

FIGURE 6 FASTENER YIELD TORQUE FOR PEDIGREED VS. NON-PEDIGREED FASTENERS

The effect of inserting and removing multiple screws in the same threaded hole was tested and the results are shown in Figure 7. The results indicated that the second set of inserted screws have a 26% higher average yield torque than the first set. The third set of inserted screws have a 12% higher average yield torque than the first set. This comparison made it clear that the yield torque of the fastener is not reduced by reusing the same threaded connection.

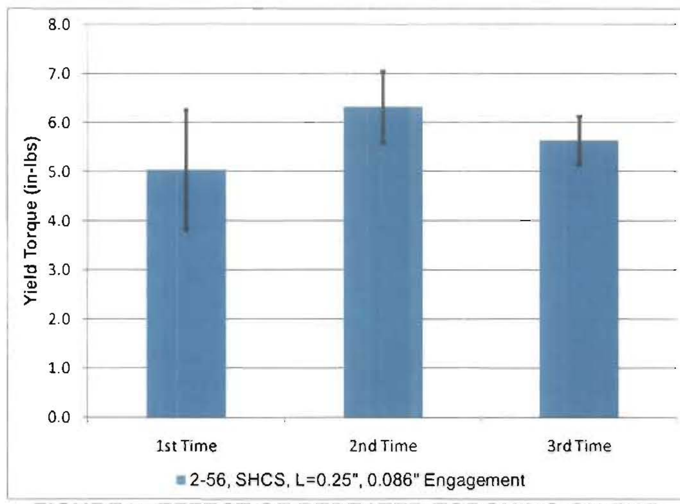

FIGURE 7 EFFECT OF REPEATED TORQUING ON THE SAME THREADED HOLE

Comparing the average yield torque of identical screws in Mg ZK60A-T5, Be S200F and Ti-6Al-4V to those in Chromate Coated Al 6061-T6 shows a decrease in the average yield torque of 24%, 22% and 19% respectively (Figure 8). A linear comparison of the yield strength, shear strength, ultimate strength, modulus of elasticity, shear modulus, hardness, and Poisson's Ratio of these four materials does not result in a matching trend.

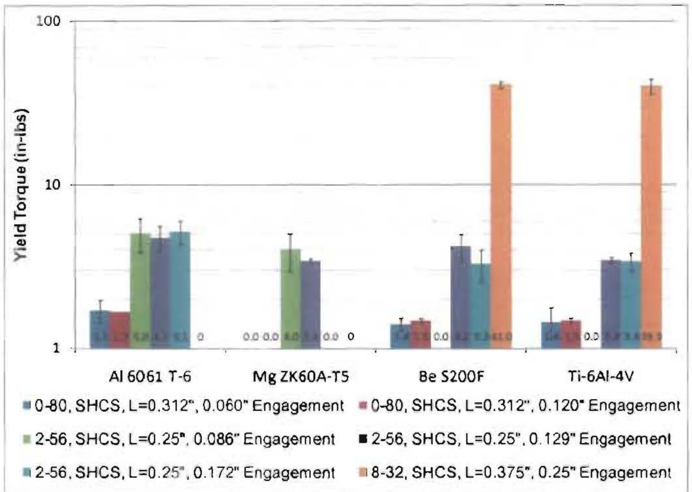

FIGURE 8 EFFECT OF PARENT MATERIAL ON FASTENER YIELD TORQUE

At least two different thread engagement lengths were tested in Al, Mg, Be, and Ti bodies for 2-56 SHCS. No definitive trend between thread engagement and screw yield torque was observed (Figure 9).

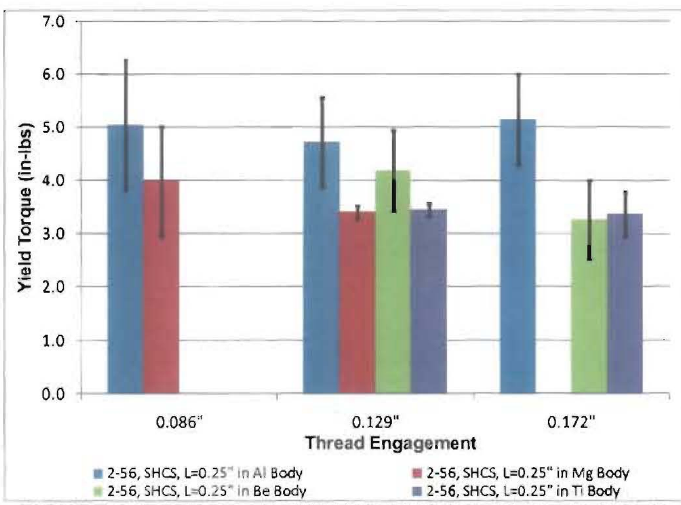

FIGURE 9 EFFECT OF THREAD ENGAGEMENT LENGTH ON FASTENER YIELD TORQUE

All tests for screw head configuration were for 2-56 screws. The data, in Figure 10, shows that the Button Head Socket Drive screws have a 74% lower average yield torque than the SHCS, Pan Head, Phillips driven screws had a 15% lower average yield torque than the SHCS and Flat Head 100°, Phillips driven screws had a 16% lower average yield torque than SHCS. These test results made it clear that wherever possible SHCS should be used in the instrument where a significant clamping force is needed.

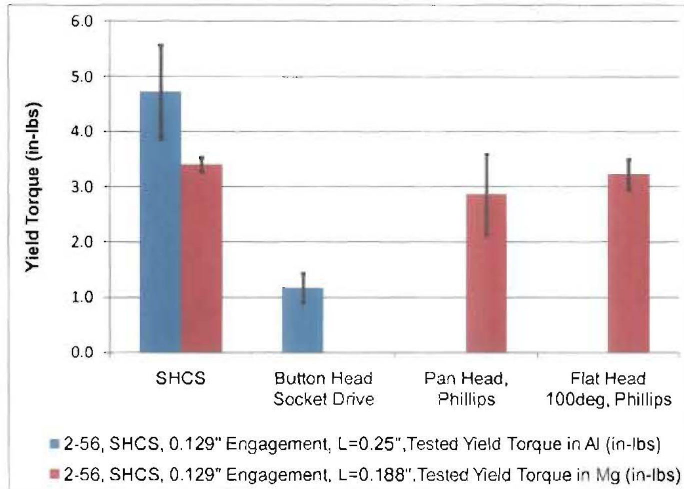

FIGURE 10 EFFECT OF FASTENER HEAD CONFIGURATION ON FASTENER YIELD TORQUE

The staking compound Vibratite was used on both 0-80 and 2-56 screws and the screw yield torque values were compared to the values for dry threads (Figure 11). Test results show that the Vibratite increased the screw yield torque by an average of 23%. Note that the data spread bands overlap for the same size fasteners.

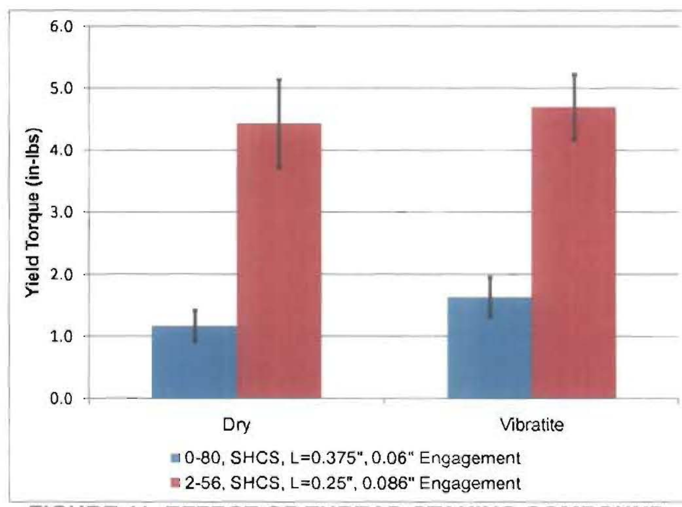

FIGURE 11 EFFECT OF THREAD STAKING COMPOUND USE ON FASTENER YIELD TORQUE

0-80 and 2-56 screws were torqued in standard Al 6061-T6 test bodies and Al 6061-T6 test bodies which had been chromate coated (Figure 12). It was found that the screws in the bare Al test bodies had a 30% lower yield torque than the screws in the chromate coated test bodies.

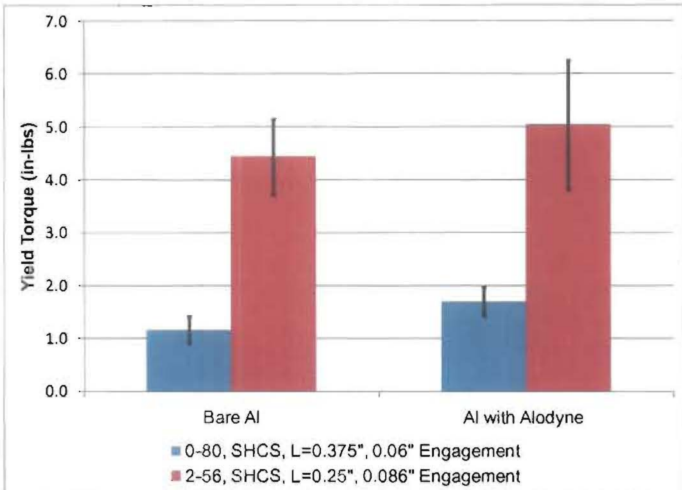

FIGURE 12 EFFECT OF PARNET MATERIAL SURFACE TREATMENTS ON FASTENER YIELD TORQUE

In Table 8 only the results for the fasteners that were used in the ChemCam instrument are shown. These fasteners were chosen because they had the highest yield torques, the most consistent yield torque results and pedigree. In the actual assembly of the ChemCam instrument 80% of the experimentally measured yield torque was applied to the fasteners and the instrument assembly was subjected to the prescribed vibration launch tests and passed all of them.

TABLE 8 FASTENER YIELD TORQUE RESULTS

Parent Material	Sample Size (# of Screws)	Fastener Designation	Thread Engagement (in)	Fastener Head Configuration	Fastener Material	Experimentally Measured Yield Torque (in-lbs)	Max/Min Experimental Yield Torque (in-lbs)
Al 6061-T6	5	0-80	0.060	SHCS	Stainless	NA	NA
	2	0-80	0.090	SHCS	Stainless	1.65	2.0/1.3
	3	0-80	0.120	SHCS	Stainless	1.70	2.0/1.5
	5	1-72	0.073	SHCS	Stainless	2.98	3.7/1.8
	5	2-56	0.086	SHCS	Stainless	5.04	6.1/3.4
	5	2-56	0.129	SHCS	Stainless	4.72	5.8/3.6
	5	2-56	0.172	SHCS	Stainless	5.14	6.0/4.2
Mg ZK60A-T5	5	2-56	0.086	SHCS	Stainless	3.98	2.8/5.5
	5	2-56	0.129	SHCS	Stainless	3.40	3.3/3.6
	5	6-32	0.150	SHCS	Stainless	12.90	15/11.3
Be S200F	5	0-80	0.090	SHCS	Stainless	1.40	1.5/1.2
	4	0-80	0.120	SHCS	Stainless	1.48	1.5/1.4
	5	2-56	0.129	SHCS	Stainless	4.18	5.0/3.3
	5	2-56	0.172	SHCS	Stainless	3.26	4.1/2.1
	5	8-32	0.250	SHCS	Stainless	41.00	43.0/37.9
Ti-6Al-4V	5	0-80	0.090	SHCS	Stainless	1.44	1.9/1.1
	4	0-80	0.120	SHCS	Stainless	1.48	1.5/1.4
	5	2-56	0.129	SHCS	Stainless	3.44	3.8/3.2
	5	2-56	0.172	SHCS	Stainless	3.36	4.0/2.9
	5	8-32	0.250	SHCS	Stainless	39.90	47.2/36.9

Clamping Force Test Results

The clamping force tests described in Section 3.c.ii resulted in measurements of torque applied to the fastener head versus the fastener clamping force. These experiments produced the clamping force values for the corresponding yield torques shown in Table 9. Furthermore, the nut factor variable from Eqn. (3) was determined from these clamping force and torque measurements.

TABLE 9 EMPIRACALLY-DERIVED NUT FACTORS FOR 0-80 AND 2-56 FASTENERES IN Al-6061-T6 AND Ti-6Al-4V

Sample Size (# of Screws)	Fastener	Nominal Major Diameter (in)	Thread Engage. (in)	Fastener Head Config.	Exp. Measured Avg. Yield Torque (in-lbs)	Max/Min Exp. Yield Torque (in-lbs)	Exp. Measured Avg. Clamping Force at Yield Torque (in-lbs)	Max/Min Exp. Clamping Force at Yield Torque (in-lbs)	Average Nut Factor (N = T/F/d)
2	0-80	0.060	0.090	SHCS	1.65	2.0/1.3	43.6	55/32	0.63
2	0-80	0.060	0.120	SHCS	1.70	2.0/1.5	53.6	62/45	0.53
4	2-56	0.086	0.129	SHCS	4.72	5.8/3.6	84.1	88/78	0.65
2	2-56	0.086	0.172	SHCS	5.14	6.0/4.2	86.5	92/81	0.69
4	0-80	0.060	0.090	SHCS	1.44	1.9/1.1	51	56/44	0.47
4	2-56	0.086	0.129	SHCS	3.44	3.8/3.2	82.1	92/77	0.49
2	2-56	0.086	0.172	SHCS	3.36	4.0/2.9	73.8	77/71	0.53

Clamping force tests were performed using both Al 6061-T6 and Ti-6Al-4V parent materials for the threaded bodies. In every comparable test the fasteners in titanium generated a higher clamping force from a lower yield torque (Figure 13). The screws generated a minimum of 6% and maximum of 13% higher average clamping force in Ti-6Al-4V compared to Al-6061-T6 for otherwise identical tests.

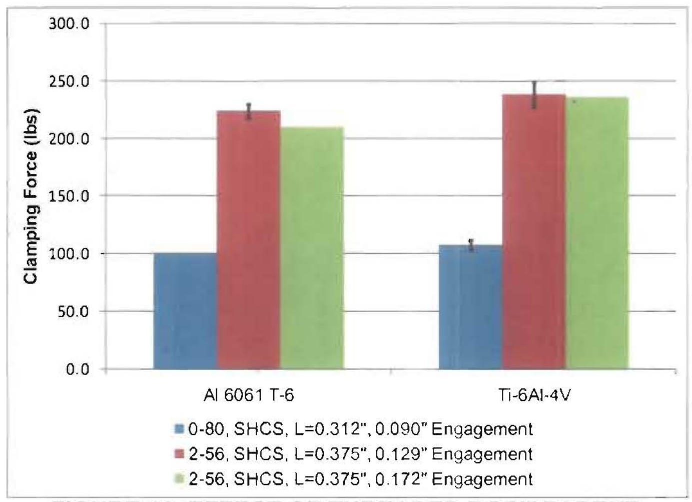

FIGURE 13 EFFECT OF THREADED BODY PARENT MATERIAL ON CLAMPING FORCE AT YIELD TORQUE

CONCLUSIONS

The unique design requirements of the ChemCam instrument including miniaturization, reduced weight, high reliability, minimal use of thread locking compounds, and the ability to handle harsh environmental conditions, dictated the use of small, high strength screws that were threaded into a variety of metal alloys including Be-S200f, Al-6061-T6, Mg-ZK60A-T5, and Ti-6Al-4V. A cursory search through available literature on fastener technology revealed diverse formulations and tabular data for determining fastener torque requirements as well as clamping forces.This was particularly true for the small, lightweight stainless steel fasteners (#0 - #8) used in this instrument. The ChemCam design team decided to experimentally determine the proper fastener torque values using a limited number of fasteners, a torque test stand, torque test fixture and clamping force test fixture. From the experimental data of this study, the following key conclusions were drawn:

1. The socket head cap was the superior head configuration providing the highest yield torque for the #0 through #8 stainless steel fasteners considered in this study.
2. Pedigreed, lot tested and certified screws had higher and more consistent yield torque values than non-pedigreed screws of identical material and geometries.
3. Repeated use of threaded holes in threaded bodies had no effect on the yield torques of new fasteners.
4. Thread engagement length had little effect on yield torque over the range of 1 to 3 diameters of thread engagement.
5. The threaded parent material influenced the yield torque and clamping force of the stainless steel screws to a small degree.
6. The Alodyne surface treatment of the Al-6061-T6 influenced the yield torque to a small degree.
7. The use of vibratite thread locking compound had negligible influence on the yield torque.
8. Experimentally determined yield torque values were determined for the #0 to #8 fasteners in Al, Mg, Be and Ti parent bodies.
9. The clamping force at the yield torque was determined for #0 and #2 fasteners and corresponding nut factors were calculated.

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