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Failures of bolts in helicopter main rotor drive plate assembly due to improper application of lubricant

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Abstract

The main rotor drive plate of the helicopter is defined as a flight safety part. Failure to follow maintenance instructions might result in an accident involving serious injury or death of crew members and/or serious damage to the helicopter. In this work, the failure of five of the shear bolts in this assembly during installation torque wrenching was analyzed. Visual inspection, fractographic and metallographic examinations, chemical analysis and mechanical testing (static tensile and hardness tests) did not indicate any deviation from the requirements of the aeronautical military standards. A specially designed bolt force sensor, which is based on strain gauge technology, was used to monitor the actual load that develops in the bolt when it is torque-wrenched into a nut. Based on comparison of the behavior of bolts that had been lubricated in different ways with antiseize material, it was concluded that the bolts failed because of improper application of the antiseize material, which led to development of overloads.
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Keywords: Bolt failures; Preload; Assembly torque; Aircraft; Thread lubrication

1. Introduction

The main rotor drive plate of the helicopter is defined as a flight safety part. Failure to follow maintenance instructions might result in an accident involving serious injury or death of crew members and/or serious damage to the helicopter. The main rotor drive plate is fastened to the main rotor head by means of 12 shear bolts–nuts connections (see Fig. 1). The maintenance manual for this part consists of its removal, cleaning, inspection and installation.

Maintenance specifications require that bolts be tightened through the application of predefined wrenching moment so that a specified preload is formed in the bolt. High preloads are often desirable to

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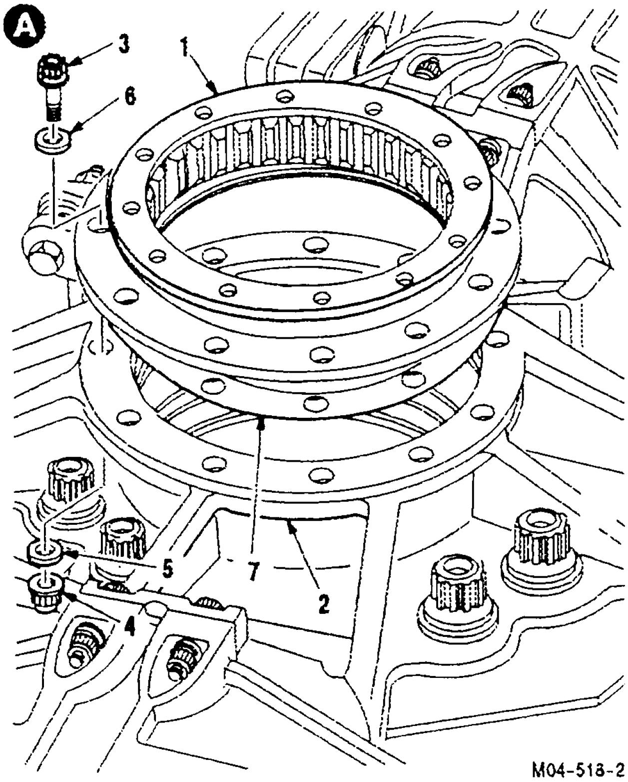


Fig. 1. The main rotor drive plate assembly in the relevant helicopter.

provide good mechanical stability and fatigue resistance to the bolted connection. One can estimate the wrenching torque required to produce a given preload using the following relation [1]:

$$T = \left[\left(\frac{d_m}{2d}\right)\left(\frac{\tan\lambda + \mu\sec\alpha}{1-\mu\tan\lambda\sec\alpha}\right) + 0.625\mu_c\right]F_i d \quad (1)$$

where d_m is thread mean diameter, d major diameter (i.e., largest diameter of a thread), λ lead angle, μ coefficient of friction, 2α thread angle, μ_c coefficient of collar friction, and F_i preload. The coefficients of friction depend on various parameters such as pitch angle, thread type, surface smoothness and degree of lubrication. On average, both μ and μ_c are about 0.15. Typical values for the term in square brackets (referred to as torque factor, K) are 0.30 for nonplated bolts, 0.18 for lubricated bolts, and 0.16 for Cadmium (Cd)-plated bolts [1]. For a given wrenching torque, one may estimate the force developed in the bolt using Eq. (1). It is evident from this equation that lower values of K (e.g., due to lubrication) will result in higher forces developed in the bolt for a given wrench torque. For example, taking the geometry values typical of the bolts under study ($d_m = 1.374$ cm, $d = 1.426$ cm, $\lambda = 0.008°$, $\alpha = 30°$) and an applied wrenching torque of 19.01 kg m, we get $F_i = 10{,}986$ kg for $\mu = \mu_c = 0.15$ ($K = 0.12$), $F_i = 14{,}797$ kg for $\mu = 0.15$ and $\mu_c = 0.10$ ($K = 0.09$), and $F_i = 16{,}397$ kg for $\mu = \mu_c = 0.10$ ($K = 0.08$). It is worthwhile recalling that bolts are carefully chosen to ensure that when fastened, a specified percent of their maximum load capacity is utilized. Any less and they will undo, any more and they will eventually break. In light of the significance of utilizing an adequate load, one should be aware of the limiting preload determination accuracy involved in monitoring the load with a torque wrench. Consequently, it was decided to design and use in this work a bolt force sensor to monitor the actual load that develops in a bolt when it is torque-wrenched into a nut. The design and use of this sensor will be described in detail in following sections.

In this work, the failure of five of the shear bolts in the main rotor drive plate assembly during installation was analyzed. These bolts fractured during wrenching under torques lower than the specified one (19.01 kg m, or 137.5 ft lb). The following section describes the experimental procedure.

2. Experimental procedure

The experimental procedure consisted of comprehensive characterization of the failed bolts, non-destructive testing of arbitrarily selected nonfailed used bolts, static tensile testing of new bolts, and sensing of forces developed in new bolts during wrenching. The material properties and fracture characteristics of the failed bolts were determined in the lab. Visual examination was carried out by means of unaided eye and stereomicroscopy (Wild model M8). Fractographic examinations were made using a Jeol JSM-840A scanning electron microscope (SEM) equipped with an Oxford energy-dispersive spectrometer (EDS). Metallographic examination was made using a Reichert-Jung MeF3 optical microscope (OM). The chemical composition of the bolts was determined through an EDS analysis, while that of the antiseize material through Fourier transform infrared (FT-IR) spectroscopy combined with EDS. For FT-IR, a Magna-IR 550 machine from Nicolet was used. The hardness of the bulk material was measured by means of the Vickers method under a load of 10 kg (Leco model V-100A). Arbitrarily selected nonfailed used bolts were removed and inspected using the fluorescent penetrant test, in accordance with McDonnell Douglas' specification [2]. New bolts were tested at room temperature for static tensile rupture load in accordance with NASM 1312-8 [3], using a standard 50-ton servohydraulic MTS load frame.

A bolt force sensor, which is based on strain gauge technology, was specially designed to control and monitor the actual axial force that develops in the bolt when it is torque-wrenched into a nut. The principle of operation and construction of this device are described in the Appendix in detail. The helicopter's maintenance manual requires that following surface preparation (i.e., cleaning), a thin coat of Molykote lubricant spray (antiseize material) be applied on the threads of bolts and let dried. Later, the bolts should be held and the nuts torque-wrenched to 17.28 kg m (125 ft lb), using a torque wrench adapter and a torque wrench.

3. Results and discussion

The first stage of this work included collection of background data. It was found that while the manufacturer did not specify any policy for the reuse of bolts and nuts, the maintenance protocols involve replacement of nuts only and reuse of bolts. Secondly, although the helicopter's maintenance manual requires that the nuts be torque-wrenched while holding the bolts, it has become common in the field to wrench the bolts while holding the nuts (for convenience reasons). To compensate for this difference, a torque 10% higher than the specified one is applied (i.e., 137.5 ft lb instead of 125 ft lb). Thirdly, while the helicopter's maintenance manual requires that only the threads of the bolts be sprayed with an antiseize material, the field level people often spray the whole bolt. In spite of these discrepancies, no similar failures were reported in the past in the Israel Air Force.

Dimensions measurements of the failed bolts did not reveal any significant deviation from the requirements in the part drawing. All bolt threads were of type 0.562-18 UNJF-3A, as required. Non-destructive testing of arbitrarily selected nonfailed used bolts by means of the fluorescent penetrants technique did not indicate the existence of any cracks.

The chemical composition of the failed bolts was found to be similar to Inconel 718. The bolts were coated with Cd, and their hardness was 49–51 HRC (converting from Vickers values). These findings are in accordance with the relevant specification [2], which requires an Inconel 718 alloy with chemistry of AMS 5662, heat treated to 220 ksi minimum tensile strength, and Cd plated per QQ-P-416 Type II Class 2. The chemical composition of the facing nuts was found to be similar to Inconel 718, and their hardness was 46–48 HRC. These findings satisfy the requirements of HS 5489 [4] for an Inconel 718 alloy per AMS 5662 or AMS 5663, heat treated to 44–50 HRC. The chemical composition of the antiseize material was found, through subtraction of the FT-IR spectrum of a new lubricant, to be identical to that of Molykote 3402-C, as required. This is an anti-friction dispersion of solid lubricants, corrosion inhibitors and an organic binder, which has coefficient of friction equal to approximately 0.03, and is suitable for service at temperatures ranging from -198 to $+300$ °C.

Visual inspection, macroscopic and microscopic examination of the fracture surfaces were carried out in the lab. Visual inspection revealed that all bolts fractured transversely, within the threads zone (Fig. 2). Residues of the antiseize material were found along the whole length of the bolts, not just along the threads zone. Inspection of the nuts did not reveal any special findings. Macroscopic examination of the fracture surface revealed a matt-gray granular pattern and flow lines propagating from the periphery of the bolt (Fig. 3). Microscopic examination of the fracture surfaces revealed equiaxed tear dimples, characteristic of ductile overload fracture mechanism (Fig. 4). Metallographic cross-sections, which were prepared perpendicular to the fracture surfaces and examined under the optical microscope, did not reveal any special findings such as unusual or gross metallurgical imperfections.

Static tensile rupture test of new bolts of the same type was carried out in a standard MTS machine. A tensile strength mean value of 50,645 lbf (22,972 kgf) was obtained, higher than the minimum value of 47,900 lbf (21,727 kgf) required in the manufacturer's specification. The effect of surface treatments on the actual axial force that develops in the bolt when it is torque-wrenched into a nut was evaluated using a specially designed force sensor (see Appendix) and a calibrated wrench. The results of these experiments are summarized in Table 1. From this table one can learn the effect on the actual axial force of either MEK cleaning or the way the Molykote lubricant is applied on the bolt. The helicopter's maintenance manual does require that surface cleaning be done before the application of the lubricant. However, we wanted to evaluate what might happen if this requirement is not fulfilled. Hence, several bolts were cleaned with methyl-ethyl-ketone (MEK) and a clean rag prior to lubrication, while other bolts were not. MEK (C_4H_8O) is a colorless liquid with an acetone-like odor. It is used, among other applications, as a solvent for oils, coatings and adhesives. From Table 1 it is evident that MEK cleaning leads to an increase of about 20% in the actual axial force that develops in the bolt during torque wrenching (compare bolt #14 to bolts

Fig. 2. Visual inspection of the failed bolts, showing transverse fracture within the threads zone.

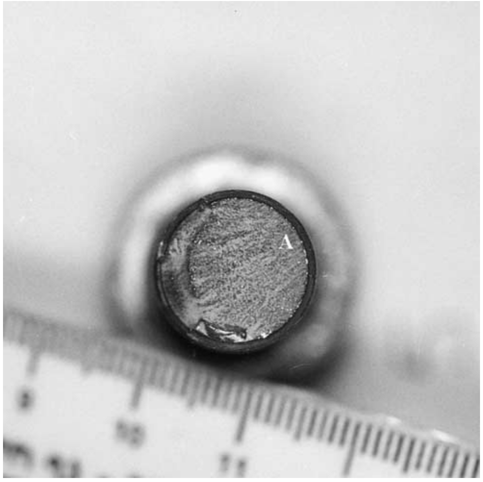


Fig. 3. A stereomicroscope macroscopic view of a typical fracture surface.

Table 1
Summary of torque wrenching measurements using a specially designed bolt force sensor, following various surface treatment procedures

Bolt S/N	Surface treatment	Wrenching torque (ft lb)	Axial force (lbf)
11	No MEK cleaning, antiseize material on whole length	137.5	36,080
12[a]	No MEK cleaning, antiseize material on whole length	137.5	36,000
13[b]	No MEK cleaning, antiseize material on whole length	137.5	36,080
14[c]	MEK cleaning, antiseize material on whole length	137.5	43,262
17	MEK cleaning, antiseize material on threads only	137.5	24,440
19	No MEK cleaning, no antiseize material	135	25,706

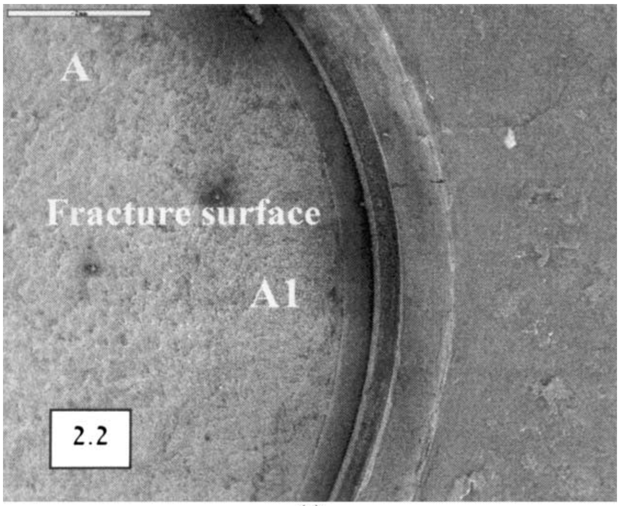
[a] Fractured under a wrenching torque of 200 ft lb, when the axial force was 45,620 lbf.

[b] The measured axial force after releasing the bolt and retightening it at a wrenching torque of 137.5 ft lb was 29,451, 27,916, 21,472 and 19,835 lbf on the second, third, fourth and fifth measurement, respectively. Following nut removal and reinstallation it was 26,800 lbf.

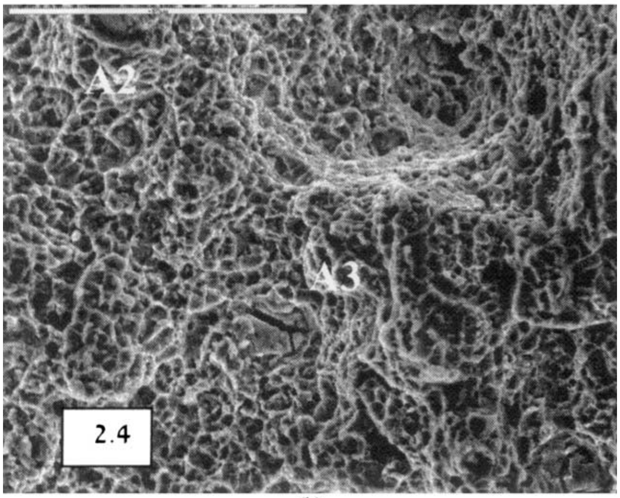
[c] Fracture within the threads zone occurred immediately upon increasing the moment above 137.5 ft lb.

#11, 12 and 13). Therefore, no harmful effect on the mechanical behavior of the bolt could be anticipated if the requirement for cleaning is not followed. From Table 1 it is also evident that all three bolts (#11, 12 and 13) that were not cleaned with MEK but were lubricated along their whole length behaved in a similar way under wrenching. This finding supports the precision of measurements using the specially designed bolt sensor device. The Appendix proves that this device is also accurate. The influence of lubrication on the actual axial force can be determined through comparison of results for bolt #14 vs. bolt #17, as well as for bolt #19 vs. bolts #11, 12 and 13 (note that the wrenching torque for bolt #19 is slightly lower). It is clear that lubrication of the threads only leads to the development of the lowest axial forces in the bolt. Application of the antiseize material on the whole length of the bolt, on the other hand, results in axial forces higher than those which develop in the absence of an antiseize material!

Thus, it may be concluded that the way in which the antiseize lubricant is applied has significant effect on the magnitude of the axial force that develops in the bolt during torque wrenching. As aforementioned, lower torque factor values due to lubrication will lead to the development of higher axial forces for a given wrenching torque [Eq. (1)]. In order to estimate the magnitude of the stresses which developed in the bolts,



(a)



(b)

Fig. 4. SEM secondary electron images of a typical fracture surface: (a) high magnification of region A from Fig. 3; (b) region A1 from (a), showing tear dimples characteristic of overload fracture.

the values of the actual axial force were normalized by the cross-section area at the bolt's shaft zone. Performing this calculation, stress values of 98, 145 and 174 ksi were obtained for bolts #17, 11–13 and 14, respectively. The value of 174 ksi obtained for the bolt that had been cleaned with MEK and lubricated along its whole length is higher than the yield point ($\sim$170 ksi), but lower than the fracture stress (220 ksi) of the bolt material [2]. One should note that if the effective area at the threads zone was used for the calculation, the values of actual stresses would be even higher. Indeed, from notes *a* and *c* of Table 1 it is evident that the actual stress that developed in bolt #14 when it was torque-wrenched at 137.5 ft lb was very close to the fracture stress of the material.

In conclusion, the bolts in the helicopter's main rotor drive plate assembly failed in overload mechanism. The cause of failure was improper application of the antiseize material on the whole length of the bolt, which resulted in the development of axial tension forces approximately 1.7 higher than normal. Following this investigation, the maintenance team was instructed to always apply the antiseize material on the threads zone only, without necessity of surface precleaning with MEK. Although no limitation was found with respect to reuse of bolts (see note b of Table 1), it was recommended to clarify this issue with the manufacturer.

4. Conclusions

Based on these results, the following conclusions were drawn:

1. The bolts in the helicopter main rotor drive-plate assembly failed in overload mechanism. The cause of failure was improper application of the antiseize material on the whole length of the bolt, which resulted in the development of axial tension forces approximately 1.7 higher than expected.
2. The antiseize material should be applied on the threads zone only. Surface precleaning with MEK is not desirable with respect to overload failure prevention.
3. The bolt force sensor, which was designed and used in this work, provides accurate and precise measurement of the axial forces that develop in the bolt when it is torque-wrenched into a nut. Therefore, it may be used as a powerful tool in failure analysis of similar bolt–nut connections.

Appendix

This Appendix describes the design, construction and validation of a bolt force sensor device. In such a device, tensile loading of the fastener induces compression loading of the sensor. Force applied to the sensor is translated into an electrical signal by means of changes in electrical resistance of strain gauges bonded to the exterior structure of the device. The strain gauges are connected in a four-arm (Wheatstone bridge) configuration network. This arrangement allows reduction of temperature change related errors as well as largely canceling signals caused by extraneous loading. Related strain gauging techniques for transducers are described elsewhere [5].

The structure of the device had to satisfy several requirements such as: (1) height small enough to allow complete fitting of the bolt into the nut; (2) elastic behavior under compression forces induced by bolt tightening; and (3) good bonding surfaces for strain gauges. The structure was fabricated from a 17-4PH martensitic stainless steel, precipitation hardened at 496°C (H925 thermal state) [6]. Strain gauges (type N2A-13-T035R-35B from Vishay) were bonded on two opposing faces of the structure. These are open-faced constantan foil gauges with a thin, laminated, polyimide film backing. They are characterized by low and repeatable creep performance and are primarily recommended for use in precision transducers as well as for stress analysis applications employing large gage patterns.

Installation of strain gauges was carried out as follows. First, the surface of the structure was polished and cleaned with a water-based acidic surface cleaner (M-Prep Conditioner A from Vishay). Second, a general-purpose cyanoacrylate adhesive (M-Bond 200 from Vishay) that cures fast at room temperature was applied. Third, circuitry was soldered using a solder material that has excellent electrical conductivity and very good corrosion resistance (361A-20R-25 solder type from Vishay). Subsequently, an air-drying polyurethane coating (M-Coat A from Vishay) was applied on the wiring and solder joints to protect them from the degrading effects of moisture, chemical attack and mechanical damage. Finally, the strain gauge wire leads were connected to Vishay's model P-3500 portable strain indicator. This instrument has unique features for use in stress analysis testing or with strain gauge-based transducers. The bolt force sensor device is shown in Fig. A1.

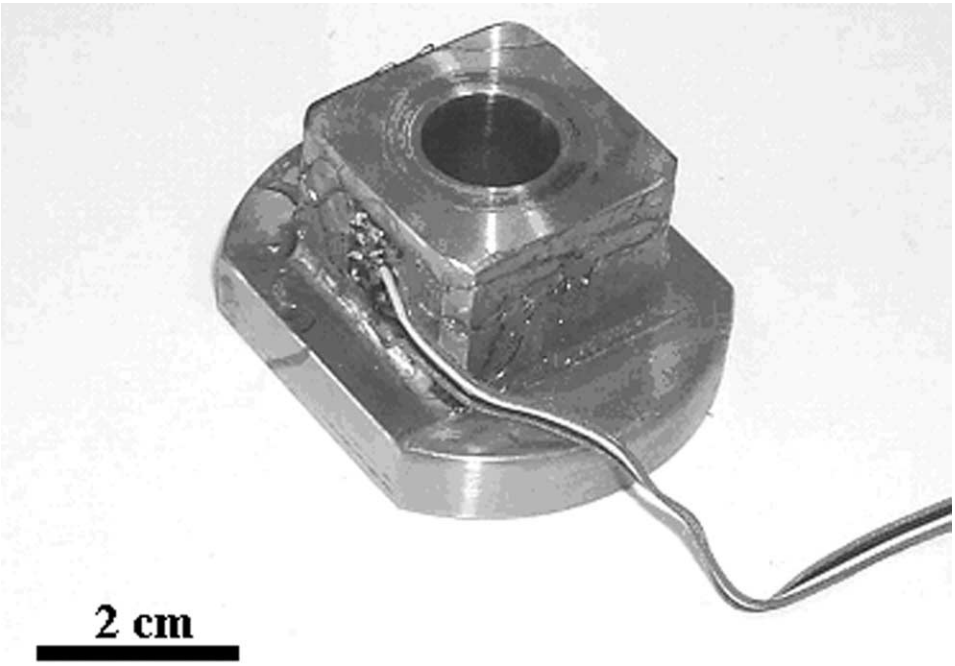


Fig. A1. General view of the bolt force sensor device that was designed, manufactured and used in this work.

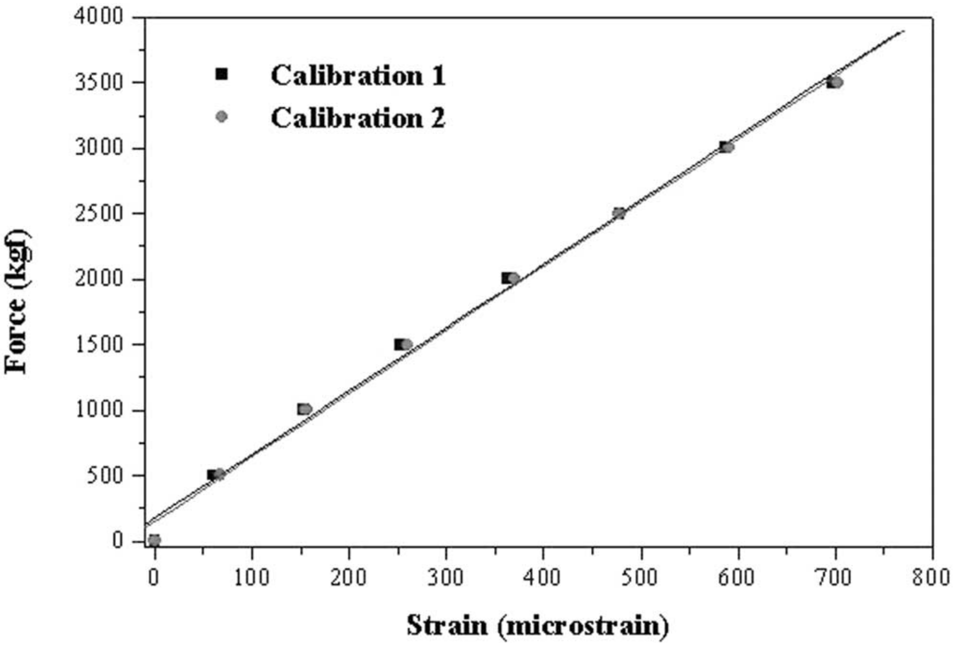


Fig. A2. Calibration plots for the bolt force sensor device indicating good reproducibility and linear force–strain relation.

Before testing the bolts, the force sensor device was calibrated to allow accurate and reliable translation of strain readings to forces. A 15-ton Instron screw mechanical testing machine was used to apply and monitor compression force on the device. In order to simulate compression of the device due to bolt tightening, the calibration was carried out while the upper part of a bolt was placed on the upper surface of the device and a nut placed at the bottom end of the device. Strain values were obtained using the P-3500 instrument. Fig. A2 shows typical calibrated plots, from which a linear relation between applied compression force and resulting strain is evident. Good reproducibility of results is also evident.

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