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(12) **United States Patent**
Rice

(10) **Patent No.: US 11,873,856 B2**
(45) **Date of Patent: Jan. 16, 2024**

(54) **PRECISION TORQUE CONTROL POSITIVE LOCK NUT**

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(72) Inventor: **Donald Wayne Rice**, Ripley, NY (US)

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(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 336 days.

(21) Appl. No.: **17/395,074**

(22) Filed: **Aug. 5, 2021**

(65) **Prior Publication Data**

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Related U.S. Application Data

(63) Continuation-in-part of application No. 15/906,549, filed on Feb. 27, 2018, now Pat. No. 11,137,015.

(51) **Int. Cl.**
F16B 37/12 (2006.01)
F16B 39/284 (2006.01)

(52) **U.S. Cl.**
CPC ***F16B 39/284*** (2013.01)

(58) **Field of Classification Search**
CPC F16B 37/12; F16B 39/284
USPC 411/178, 438
See application file for complete search history.

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Primary Examiner — Flemming Saether
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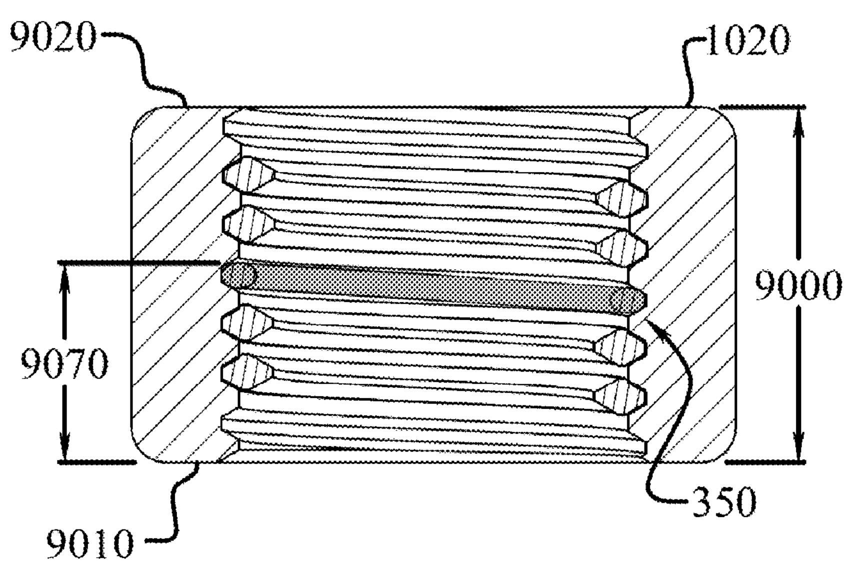
(57) **ABSTRACT**

A torque control fastener system having a fastener nut, a helical wire insert, and a shaft, where the nut and helical wire insert have unique hardness and coefficient of thermal expansion relationships that produce improved performance.

20 Claims, 28 Drawing Sheets



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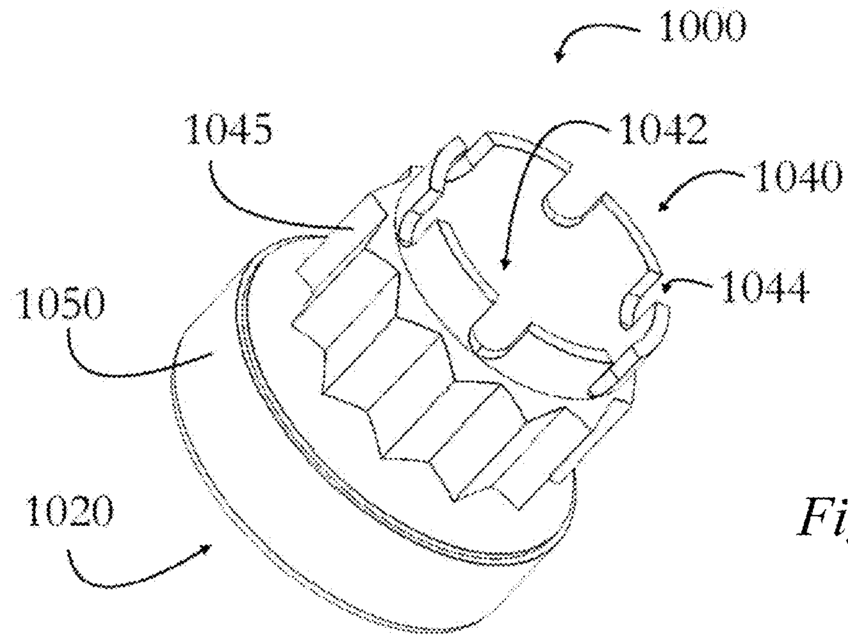

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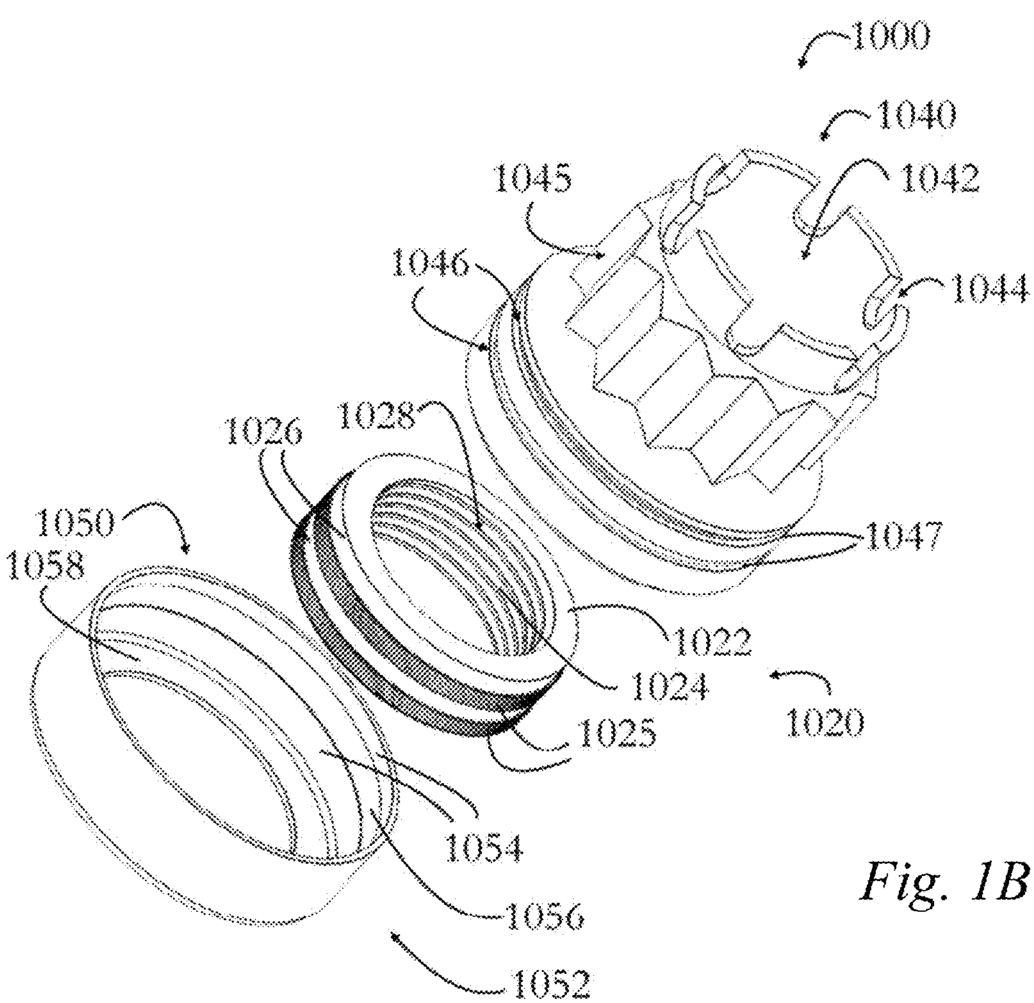

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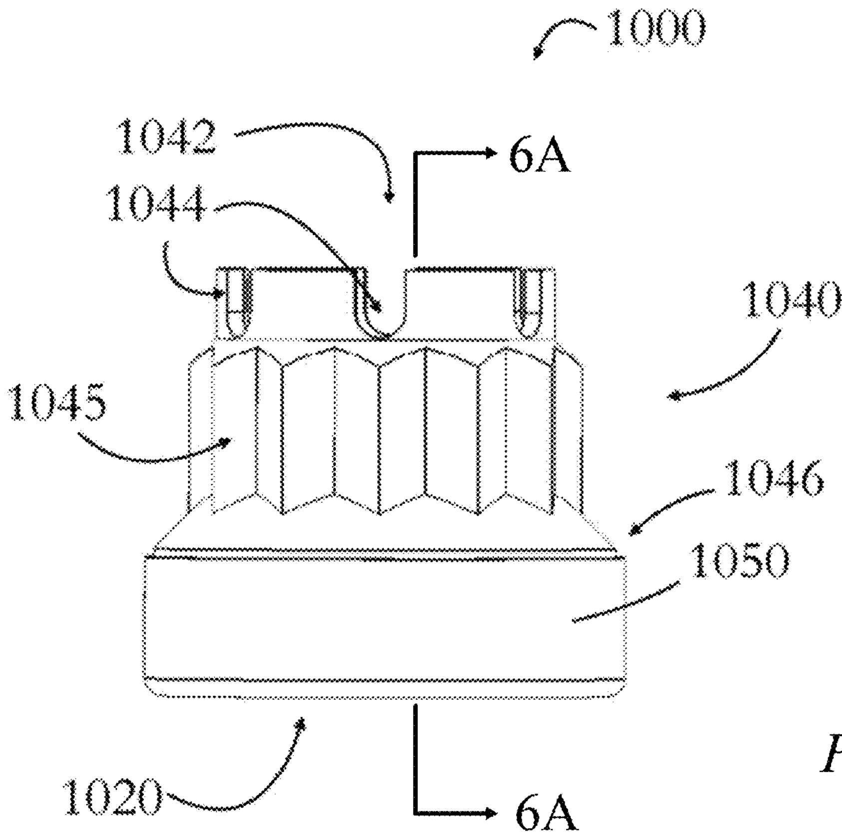
Fig. 1B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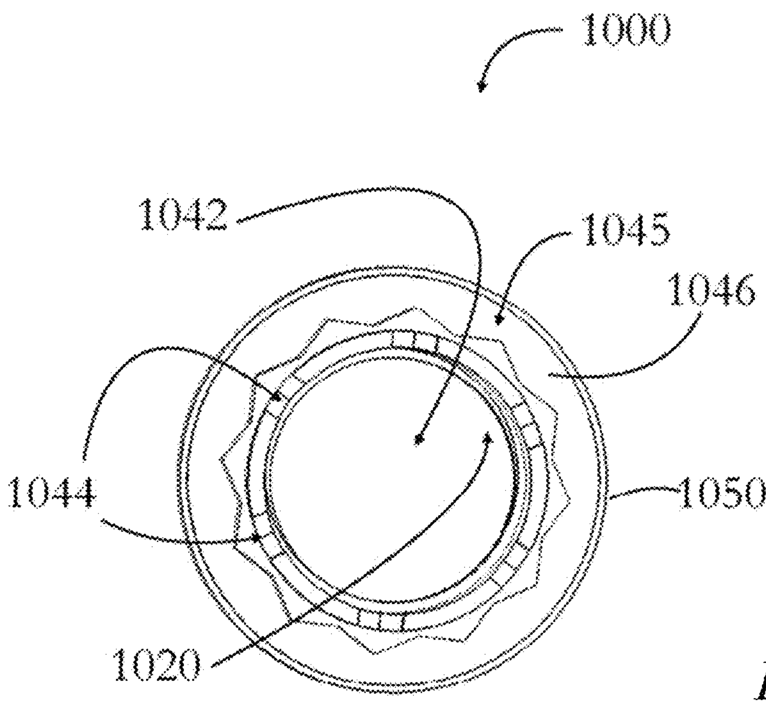
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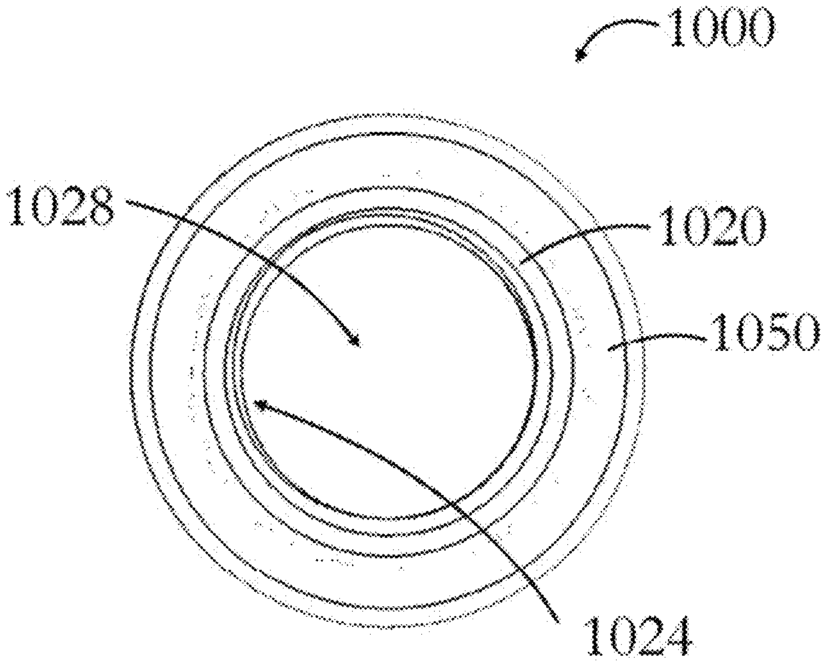

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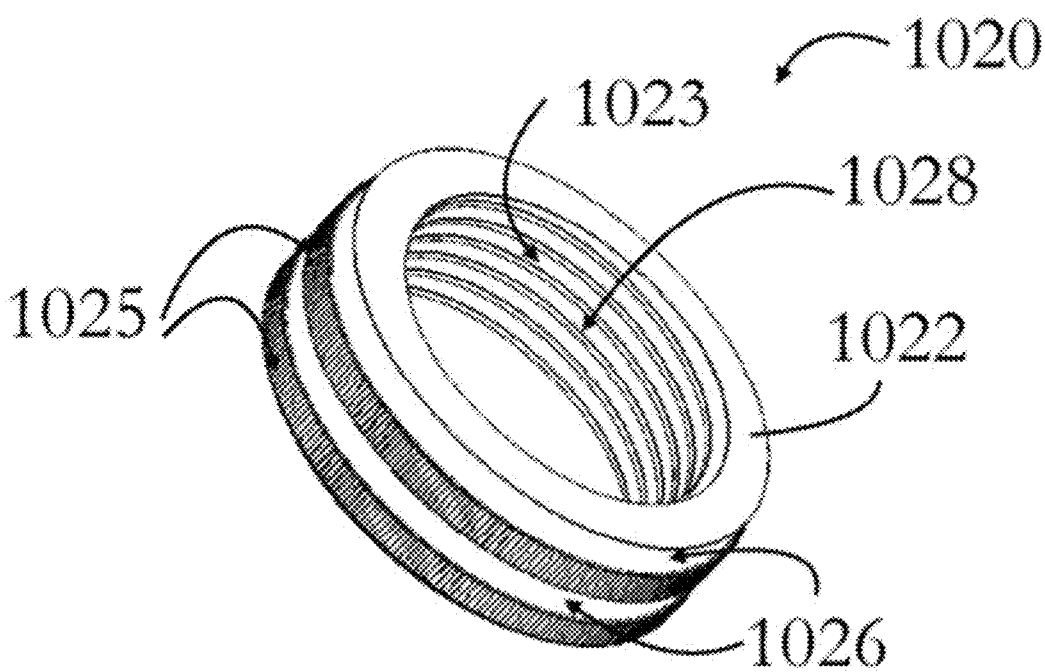

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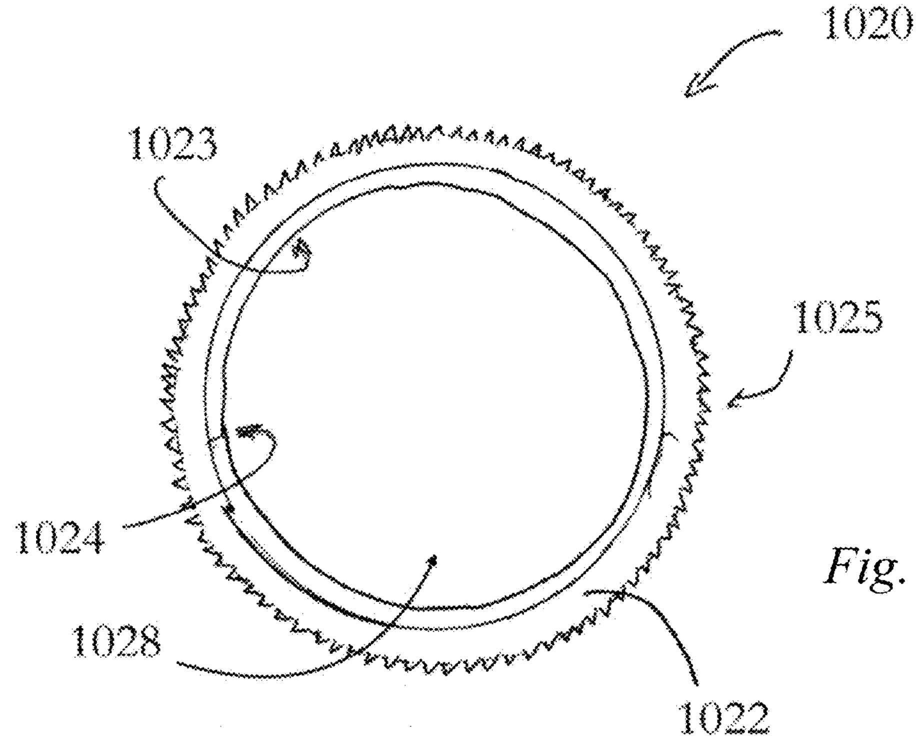
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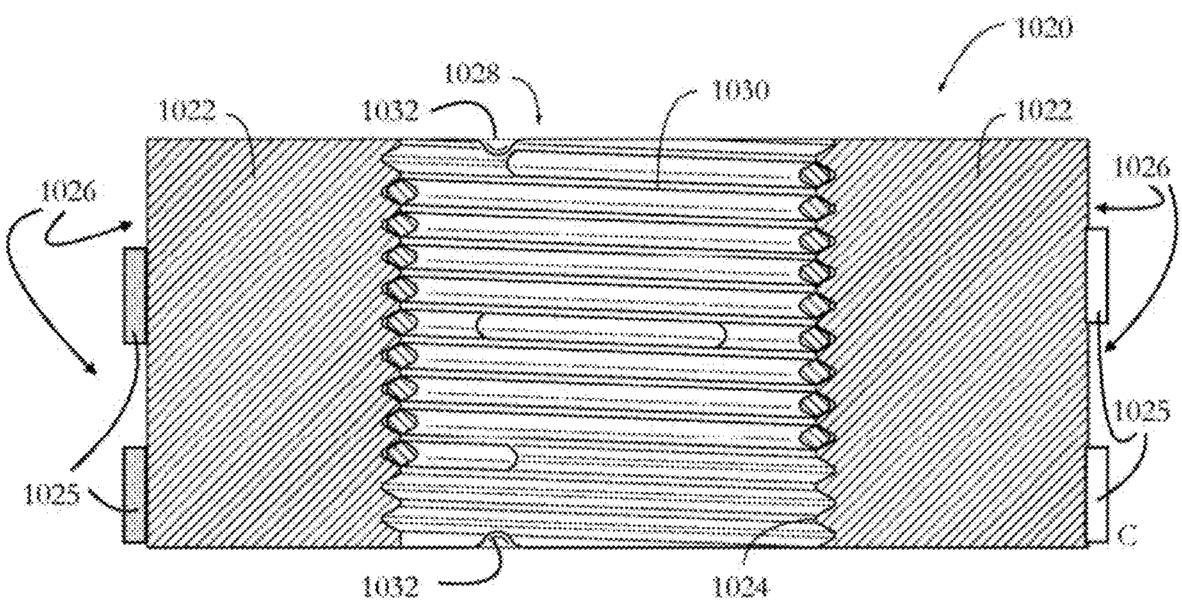
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Fig. 3C

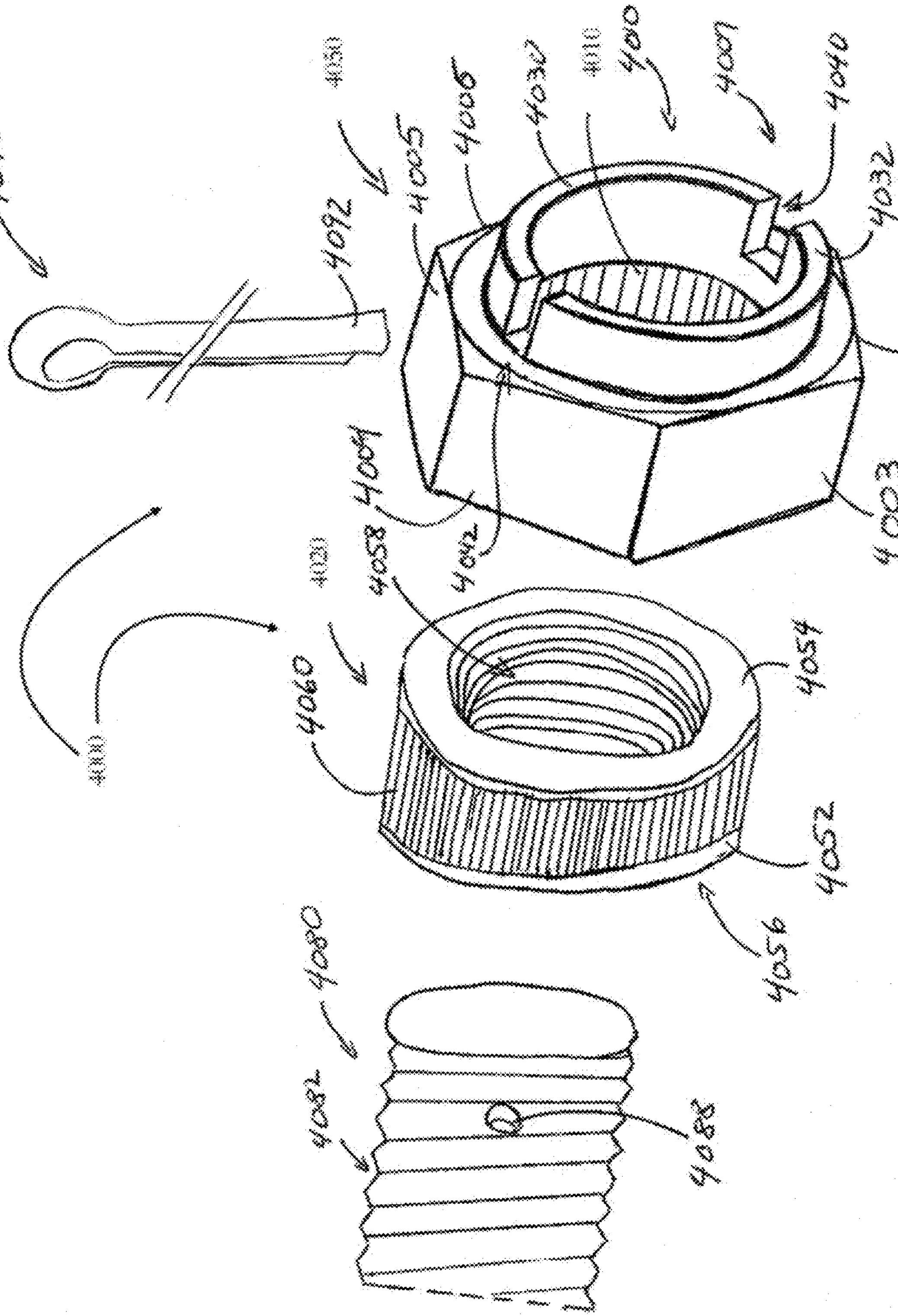
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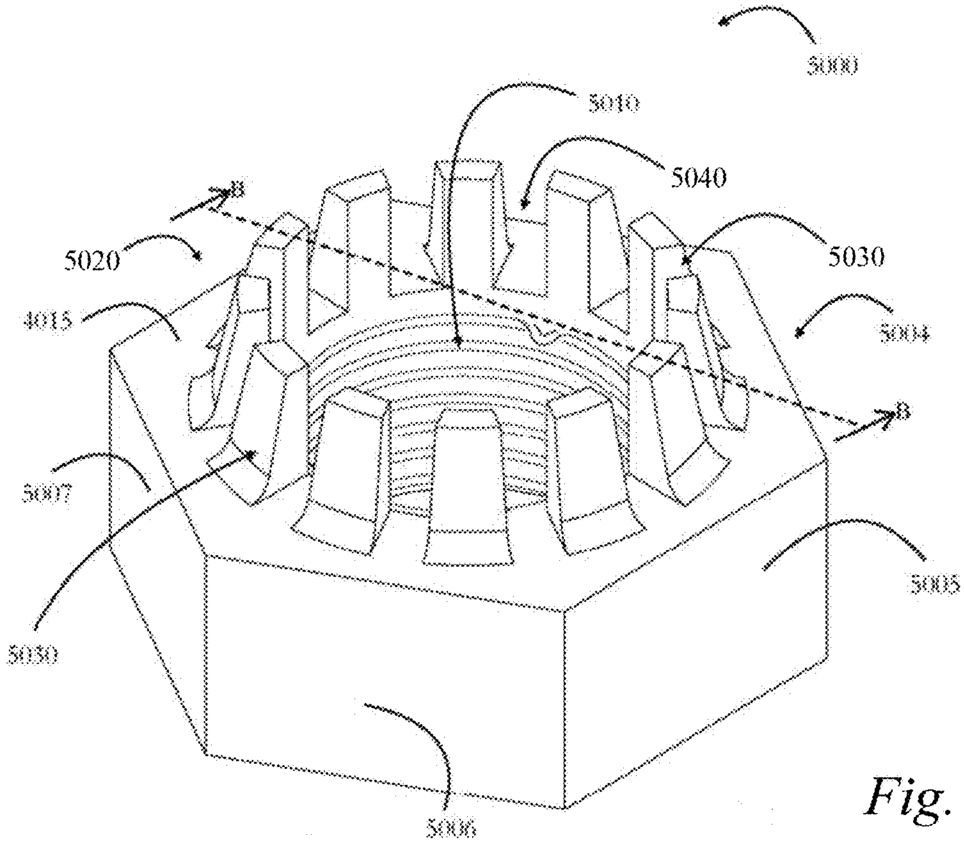
Fig. 4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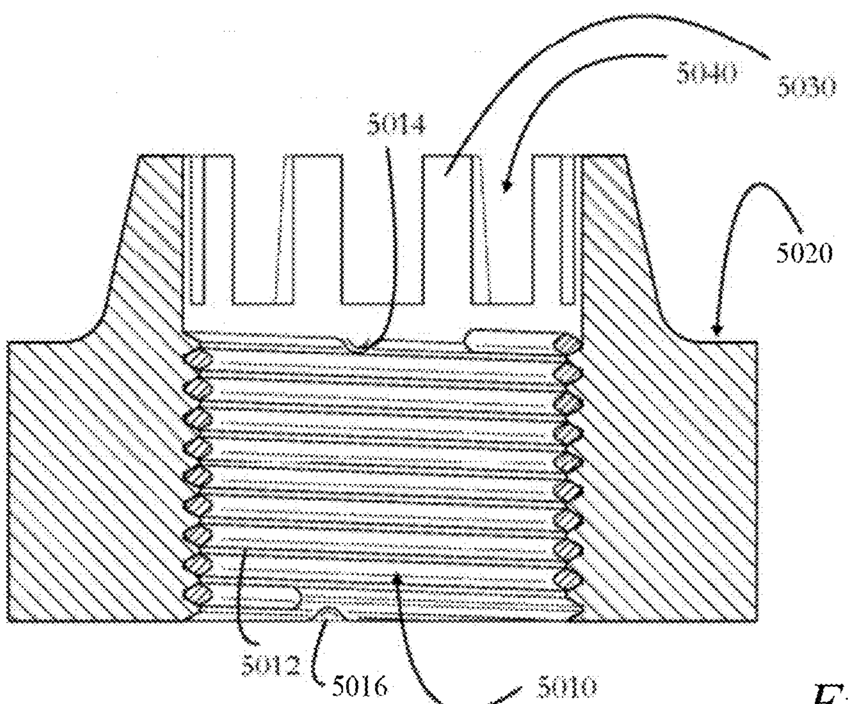
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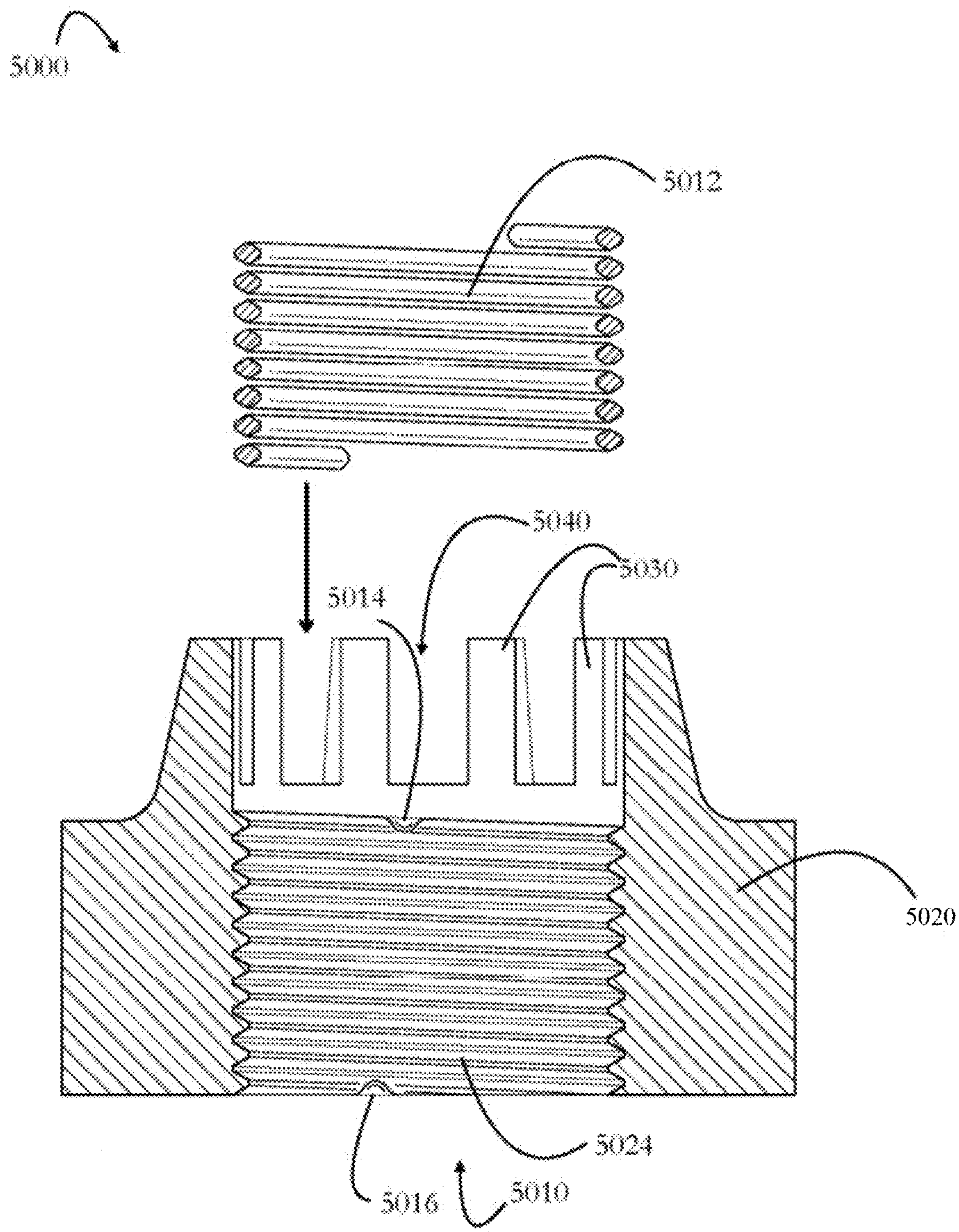

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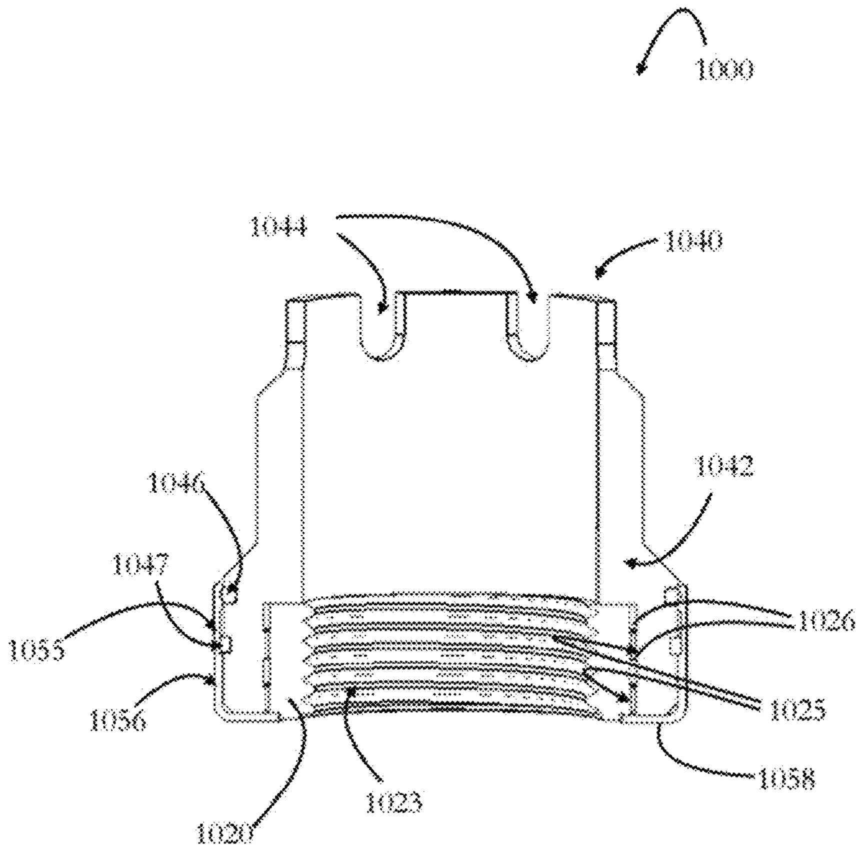
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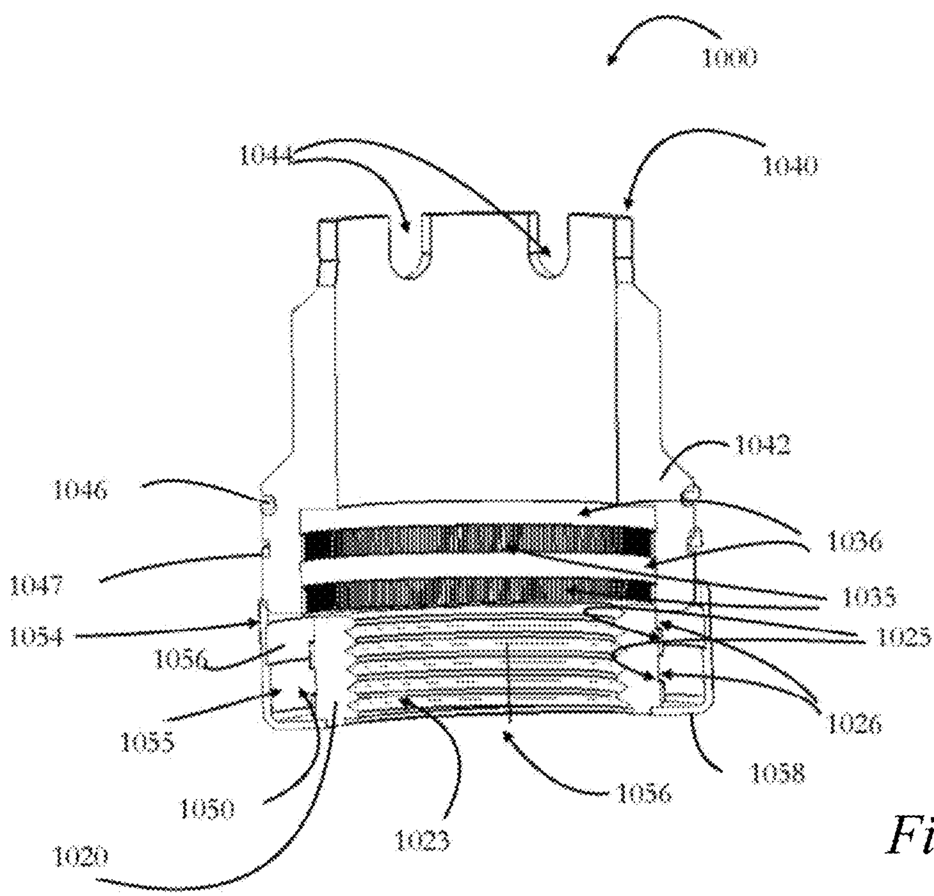
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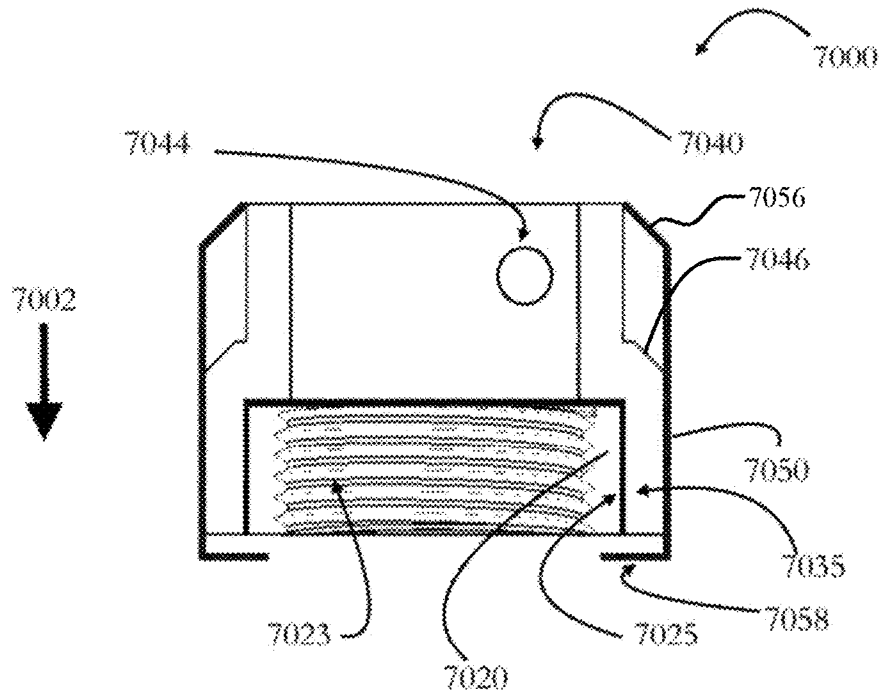
Fig. 6B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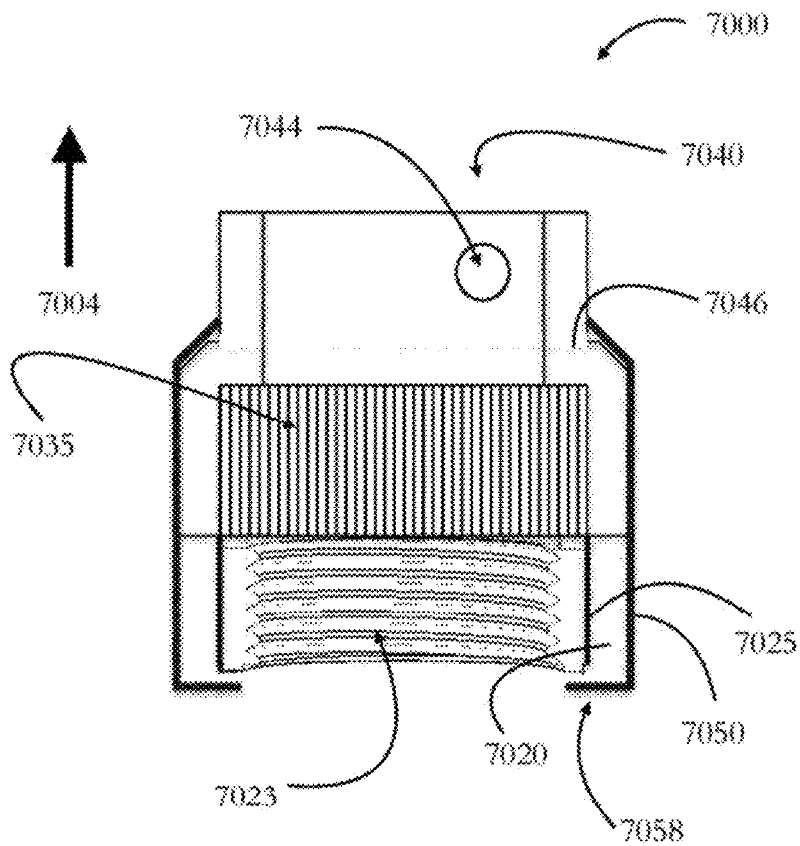
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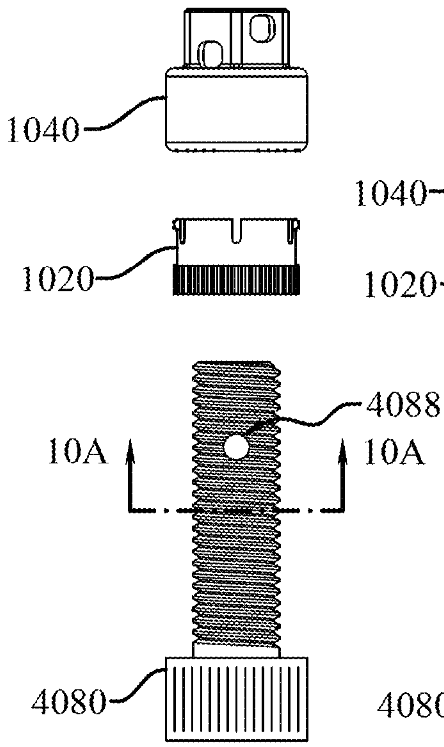

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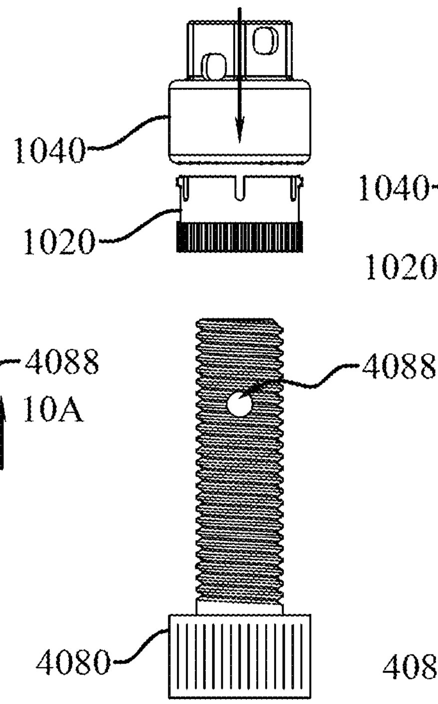

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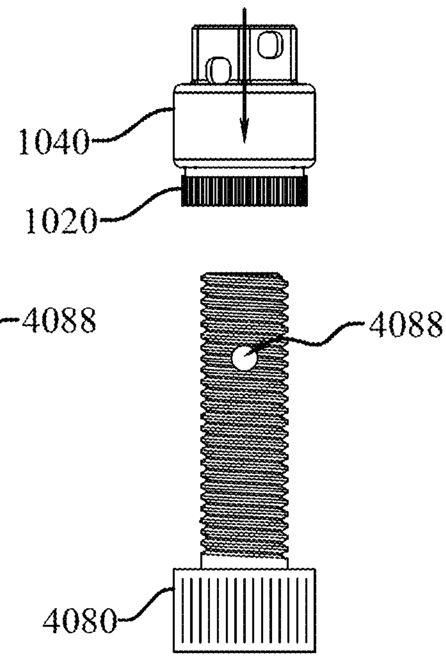

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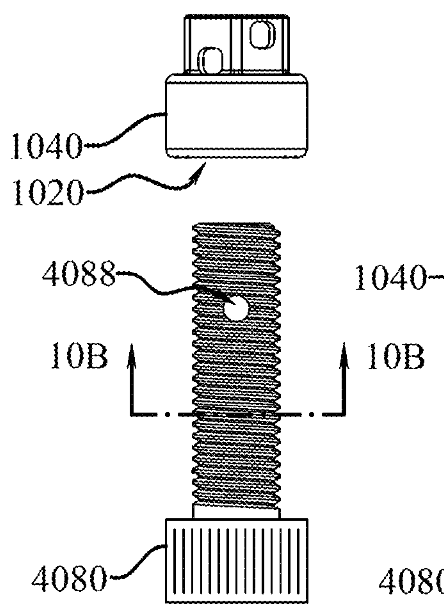

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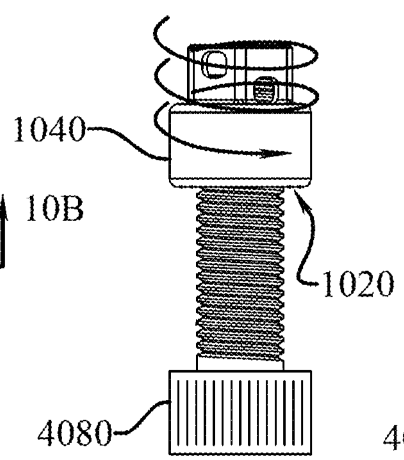

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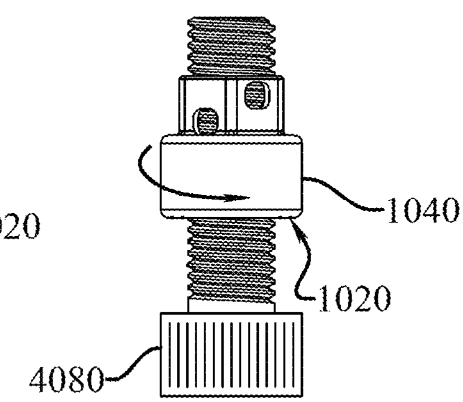

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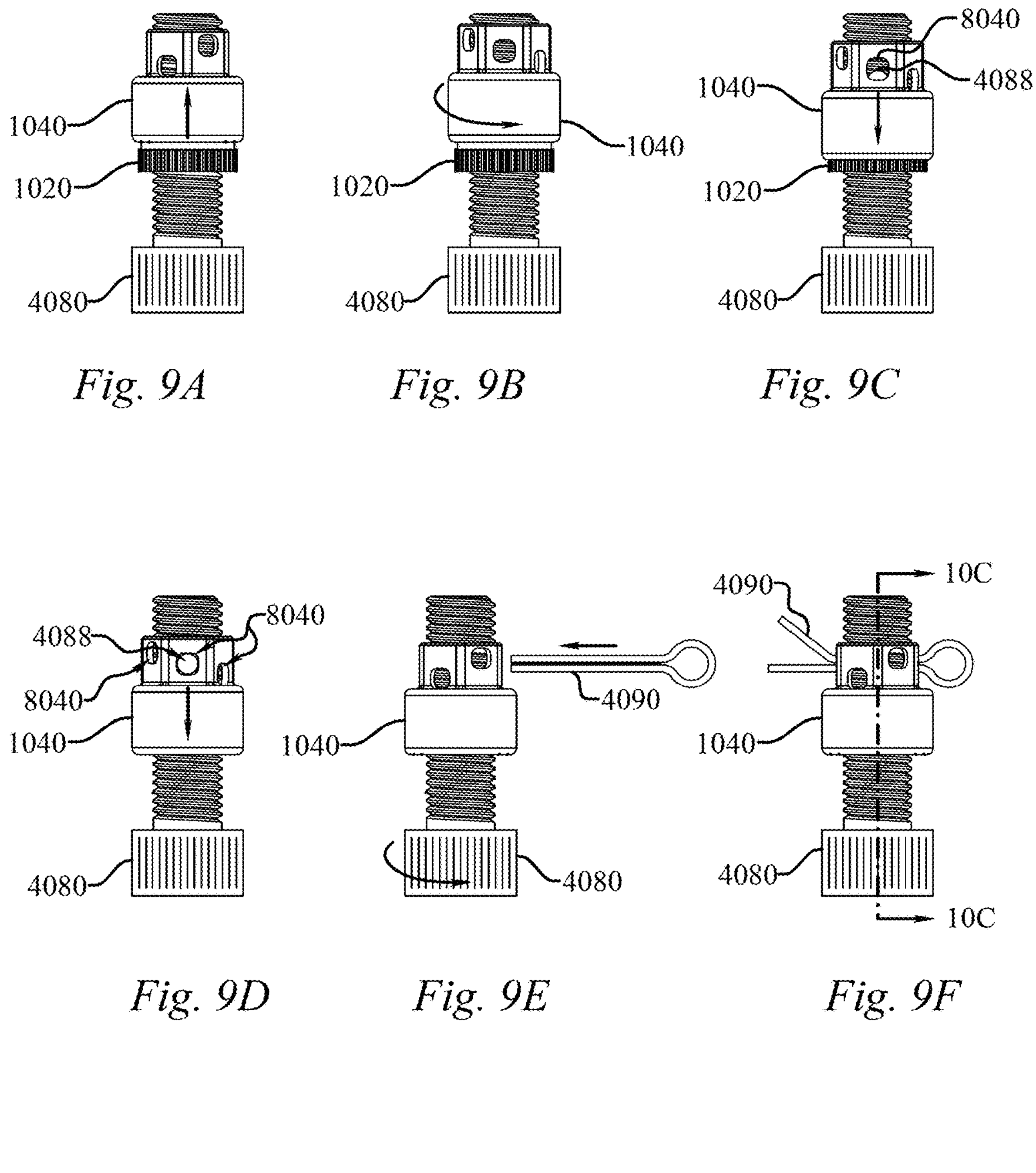


Fig. 9A Fig. 9B Fig. 9C

Fig. 9D Fig. 9E Fig. 9F

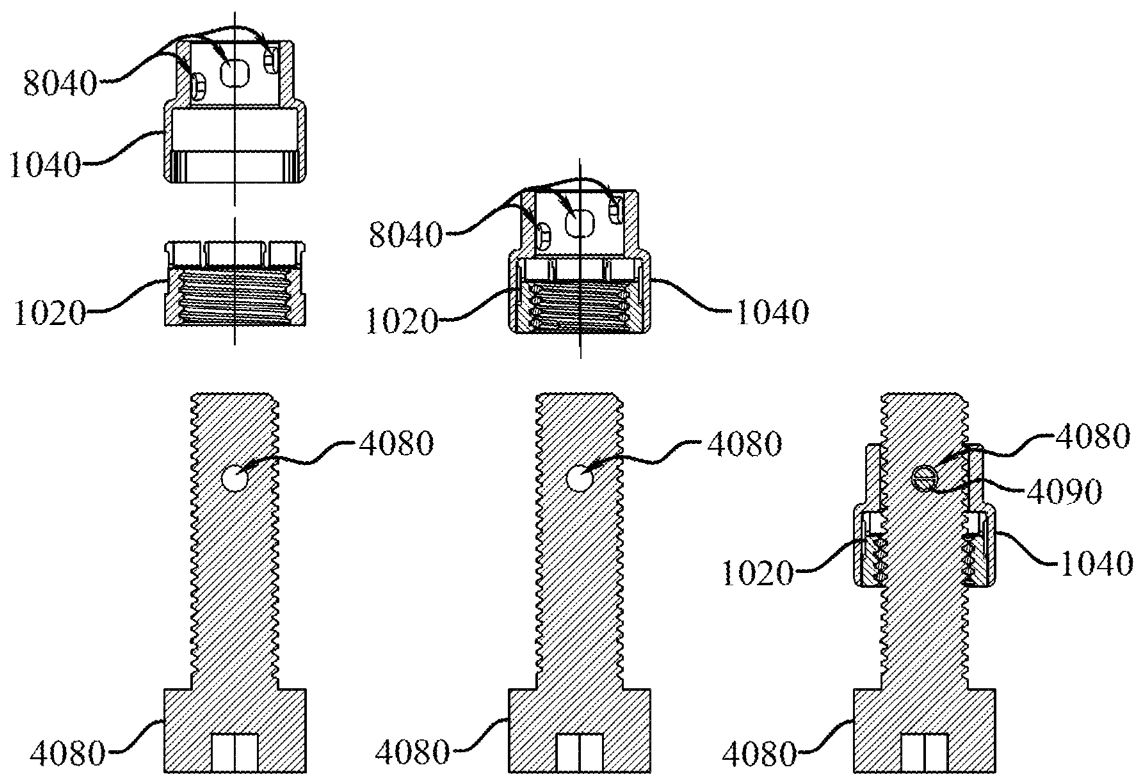
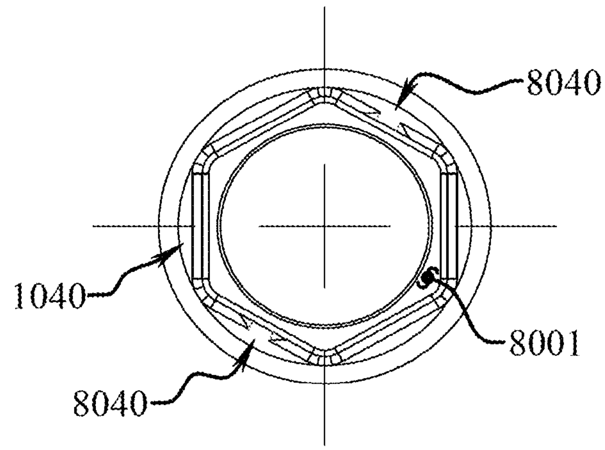


Fig. 10A Fig. 10B Fig. 10C



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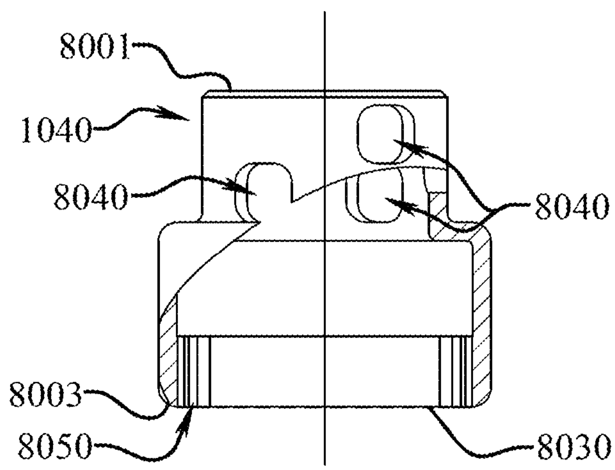

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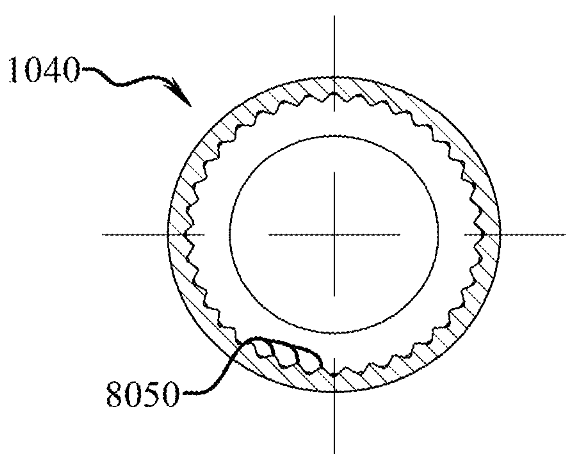

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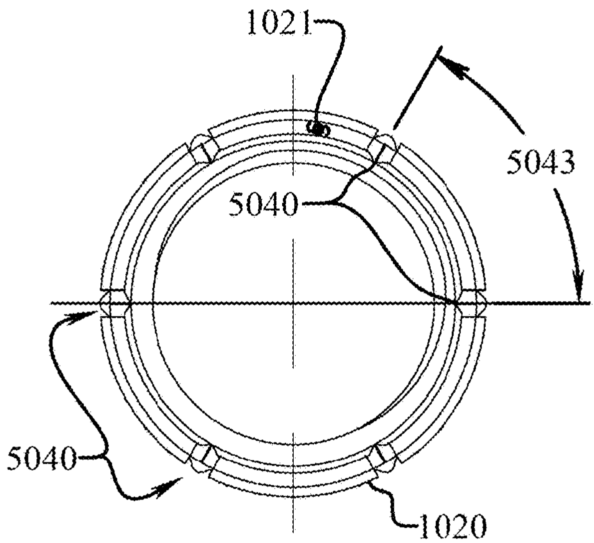

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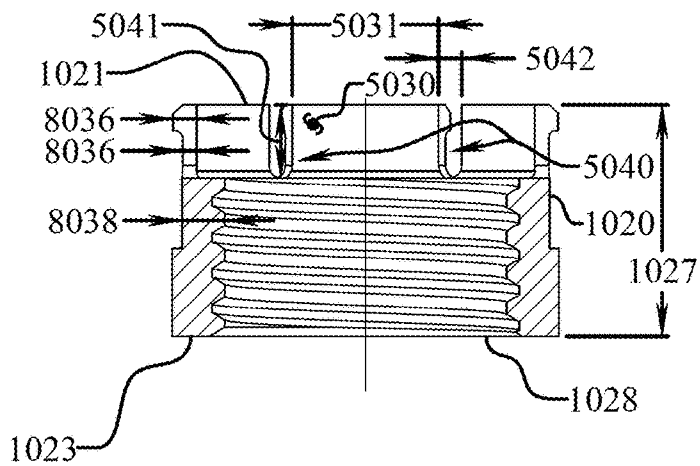

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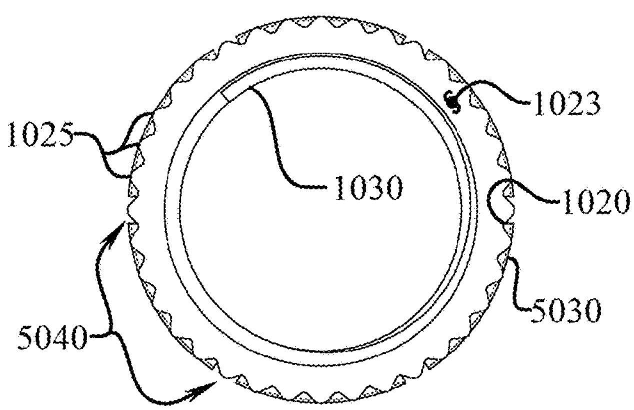

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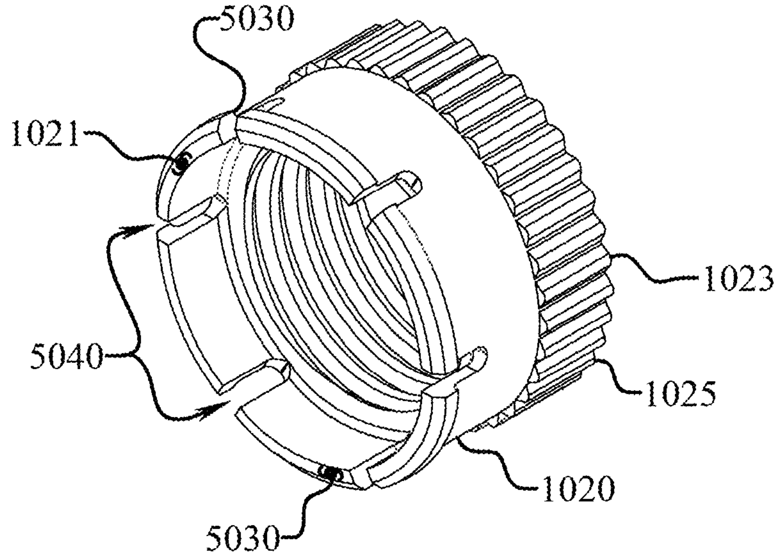
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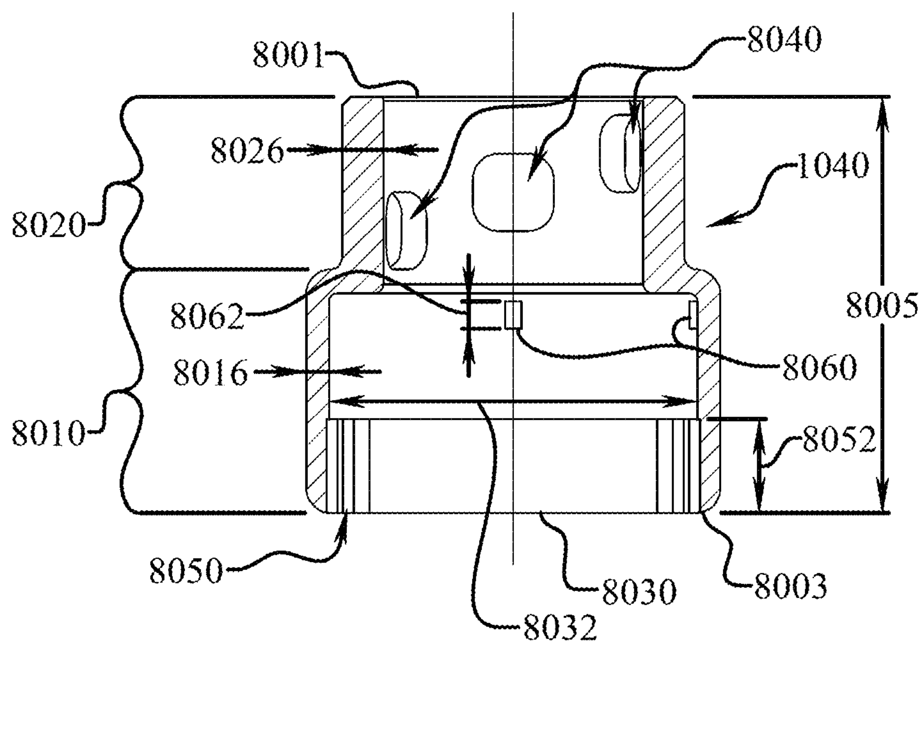
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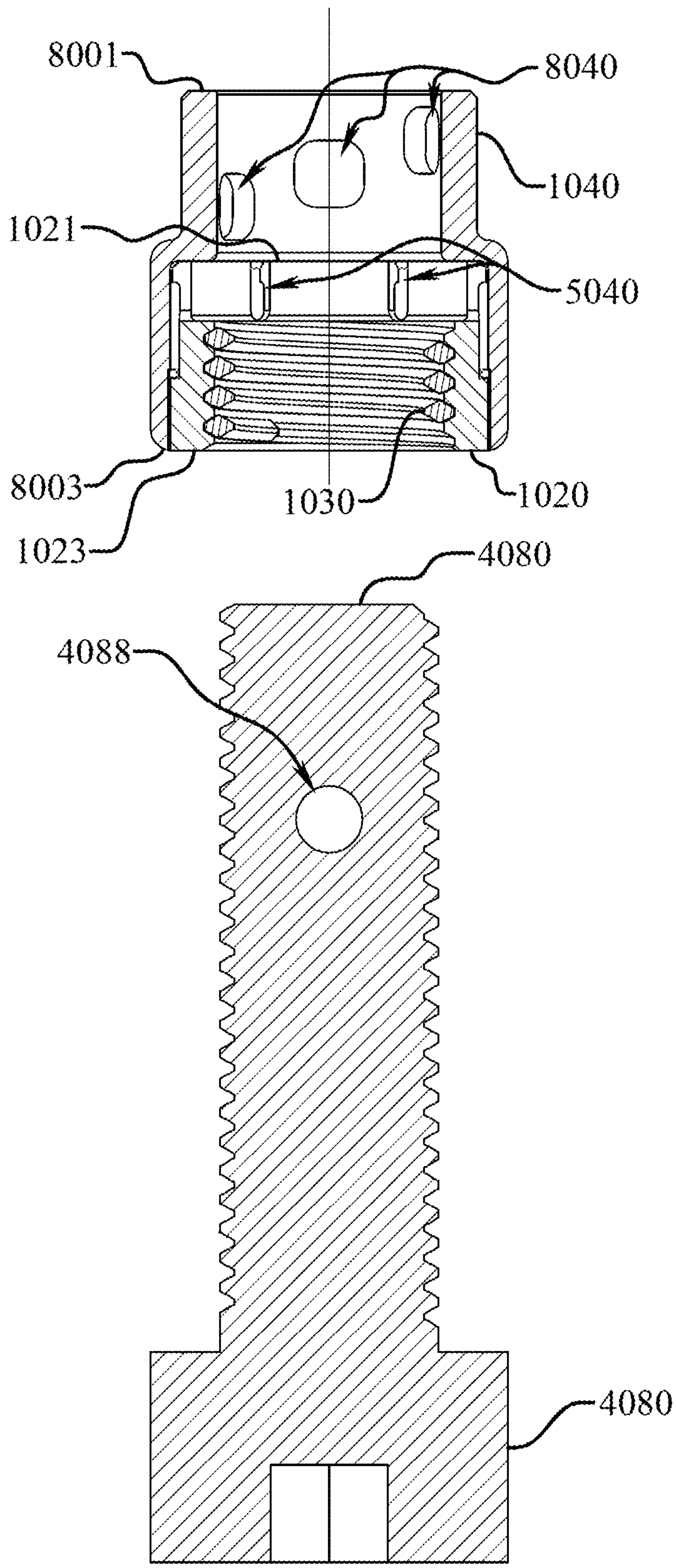
Fig. 18



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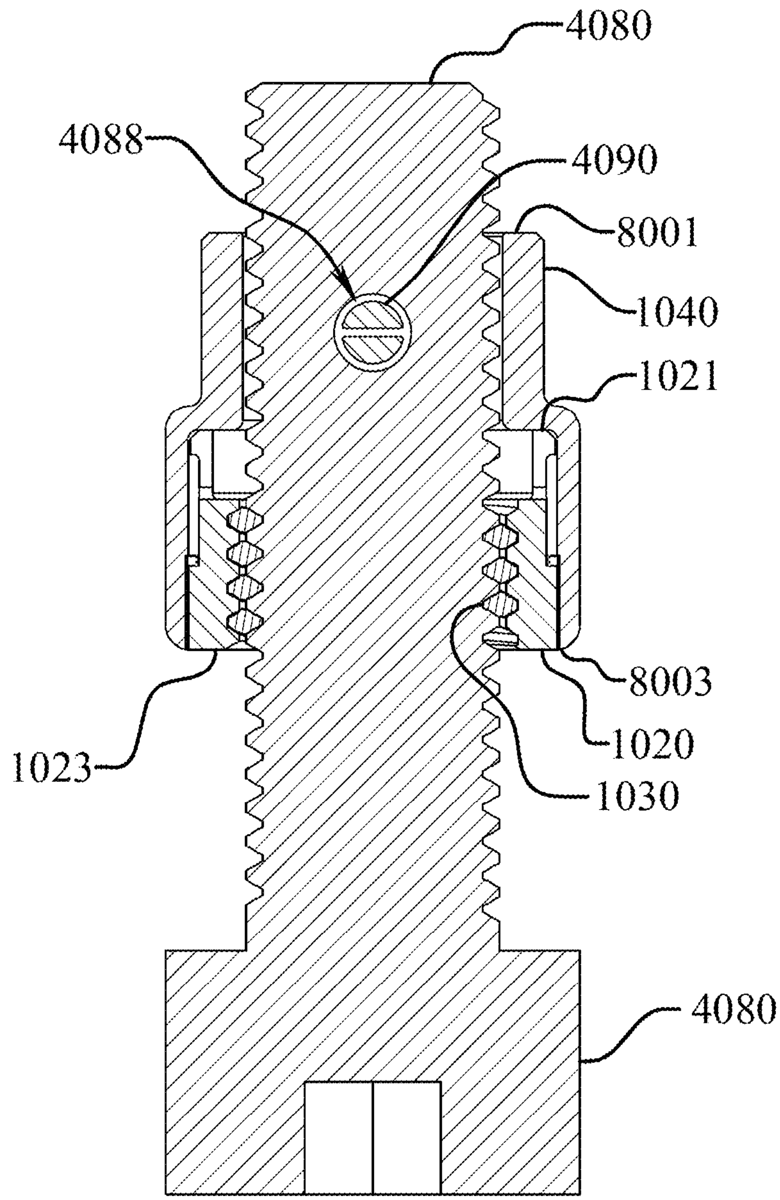

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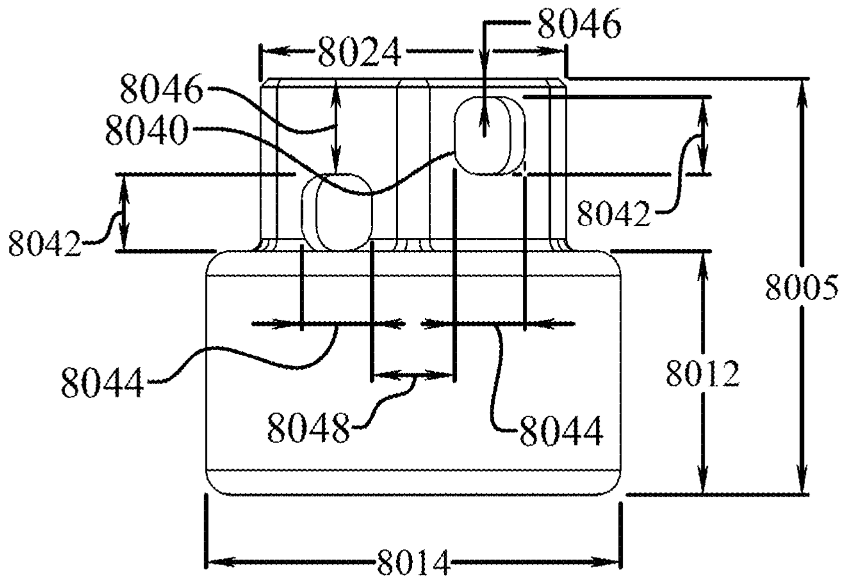
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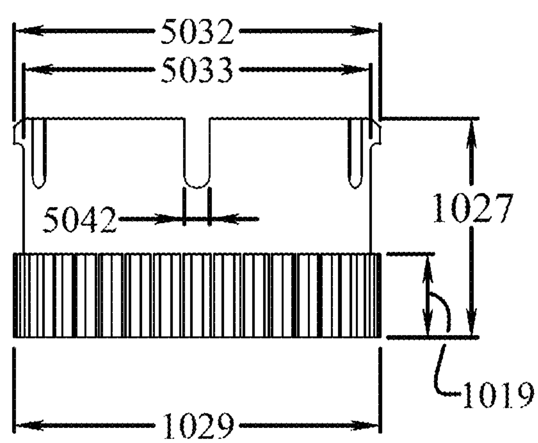
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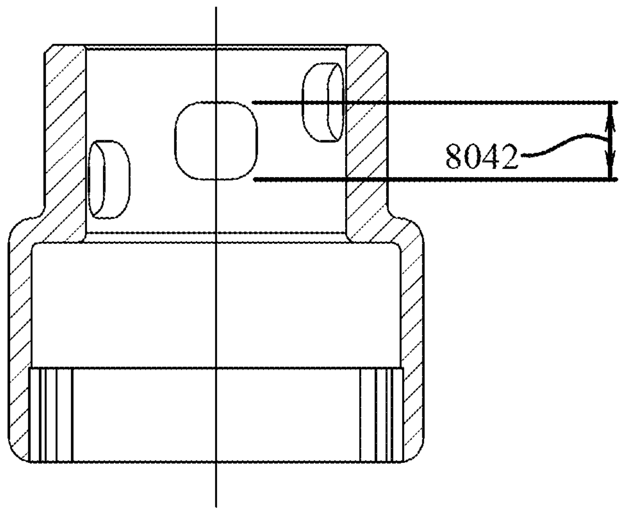

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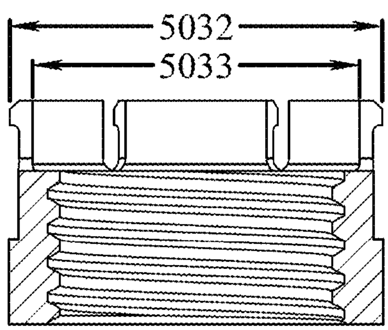

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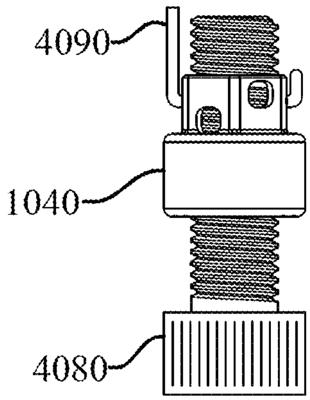
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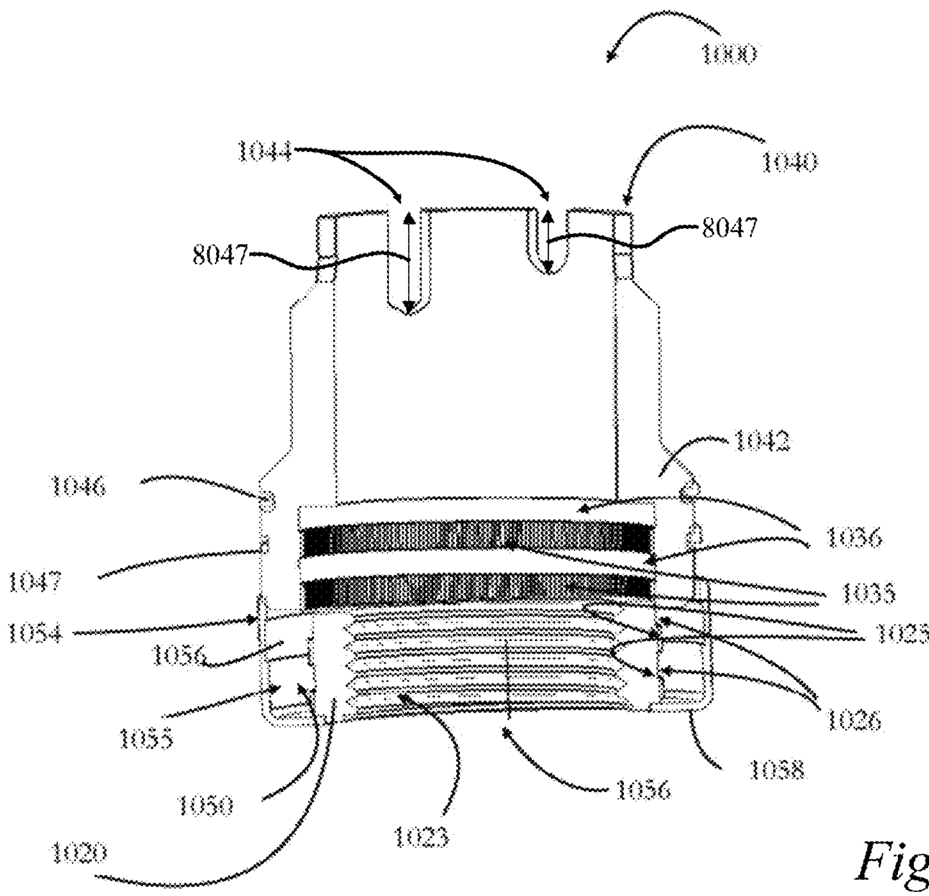
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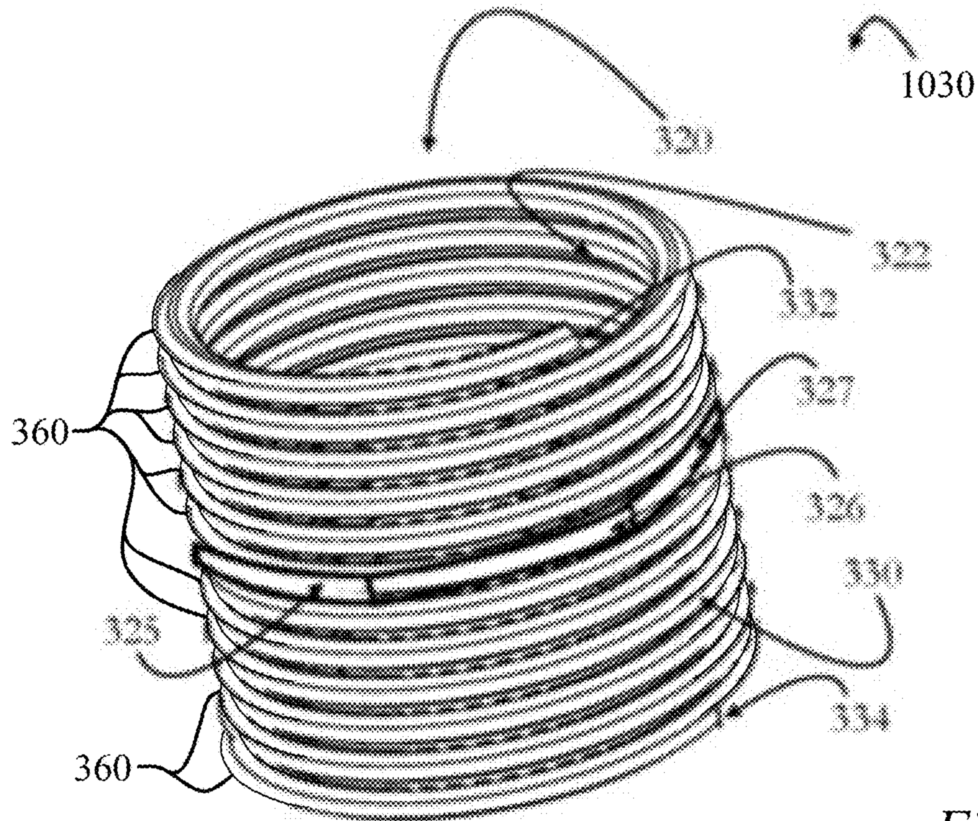
Fig. 26



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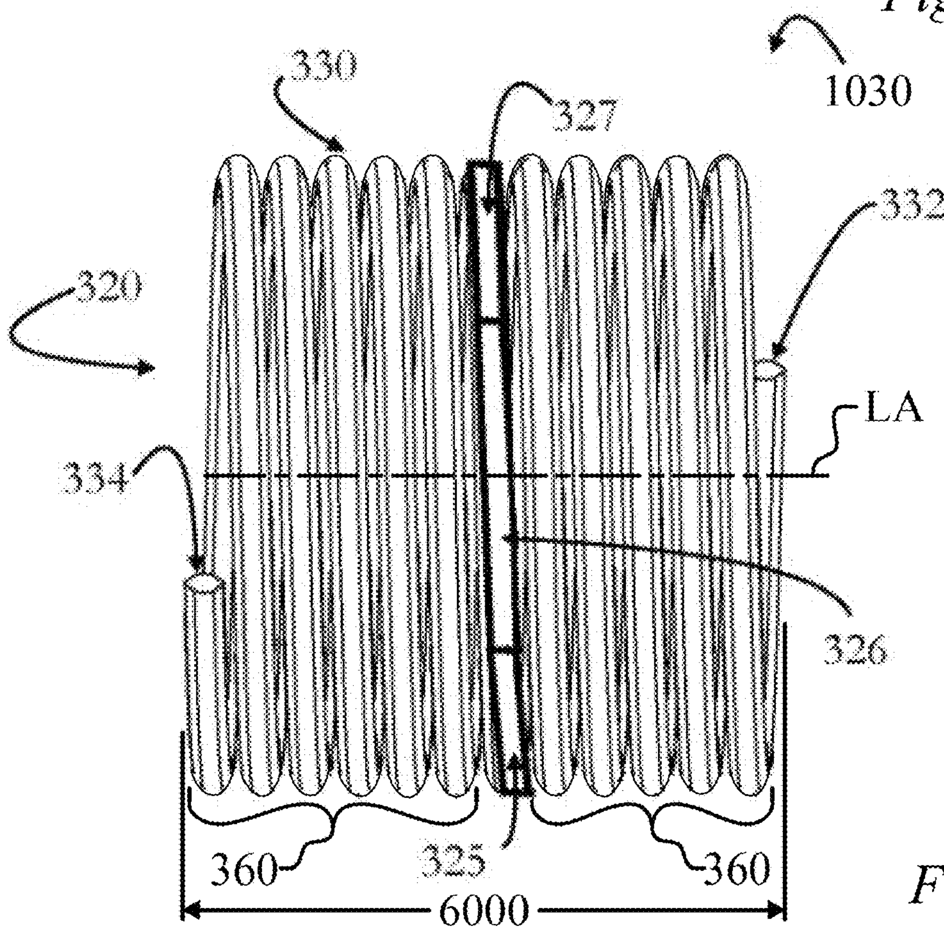

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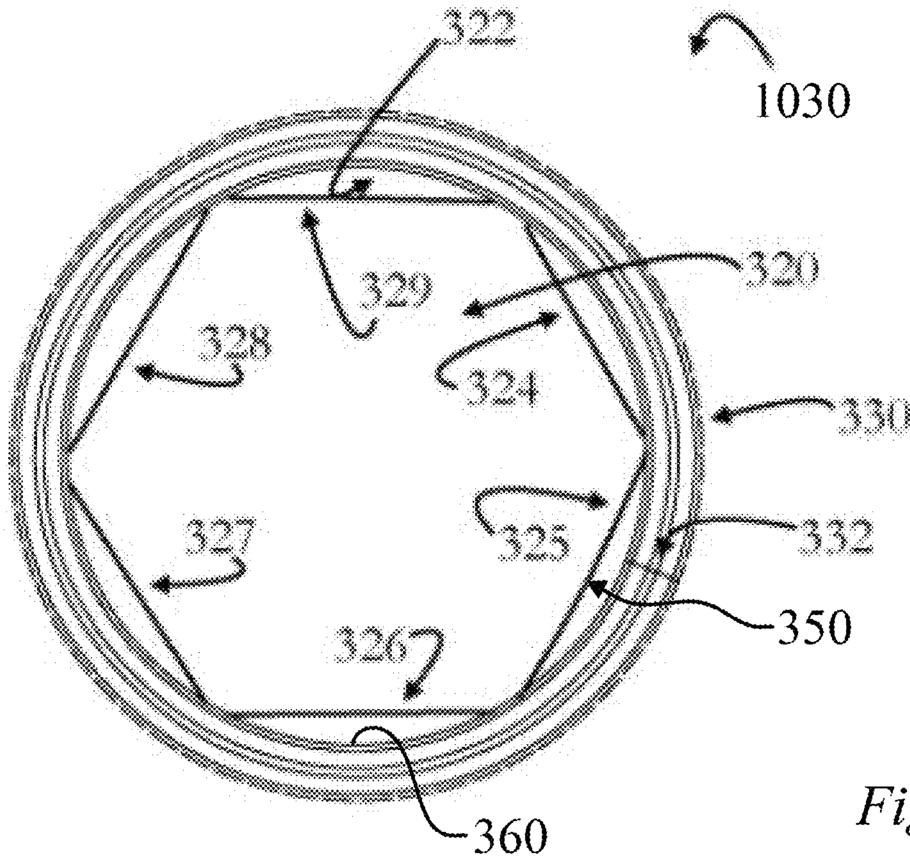

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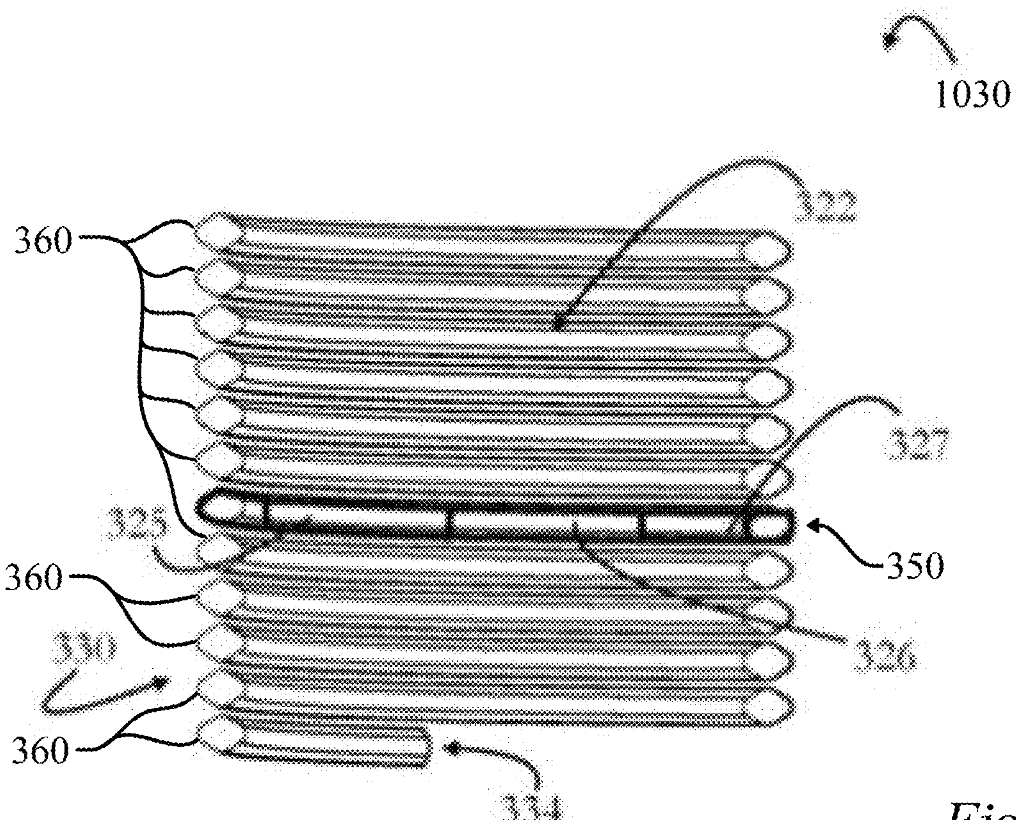

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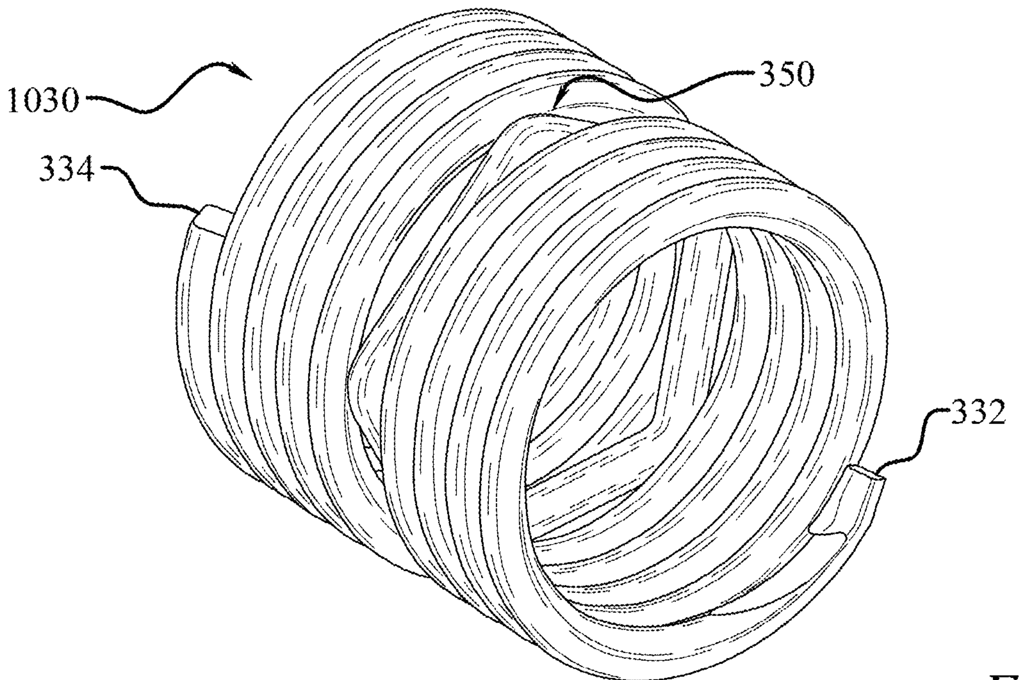

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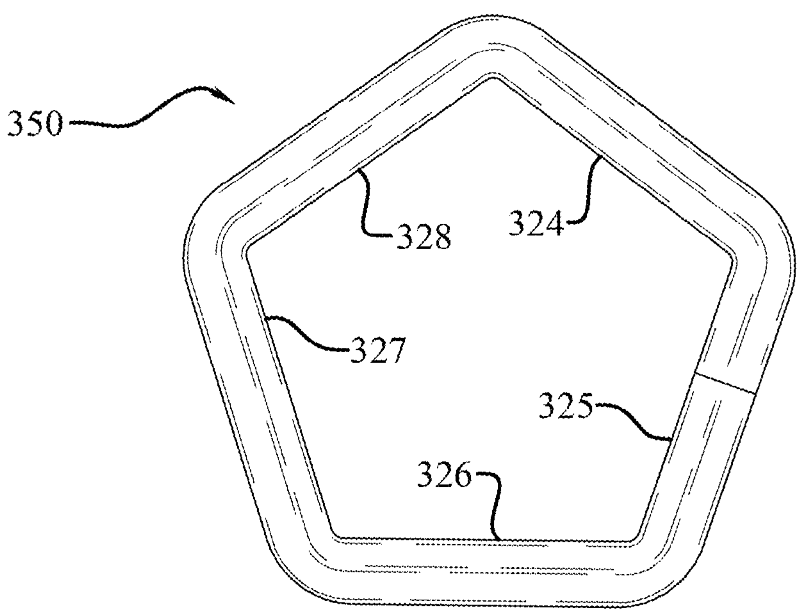

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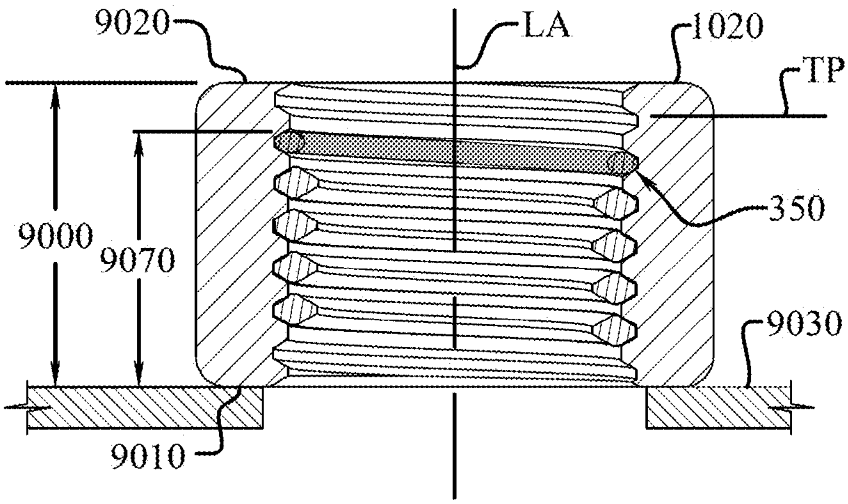
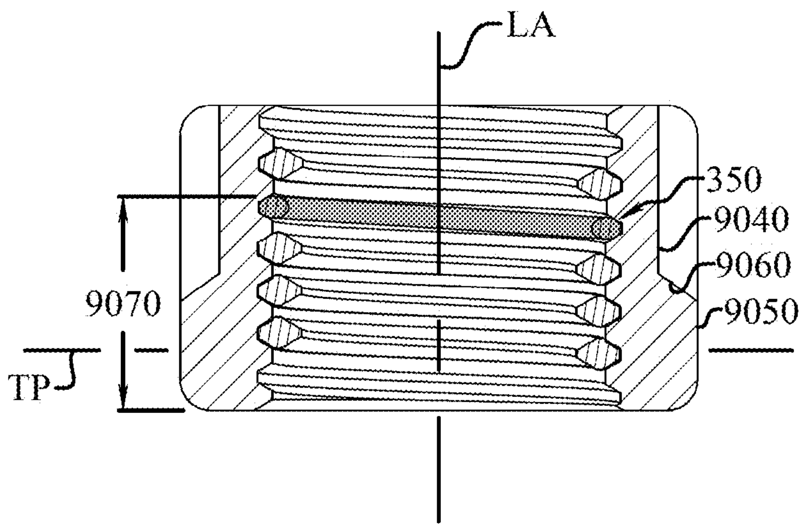


Fig. 30A



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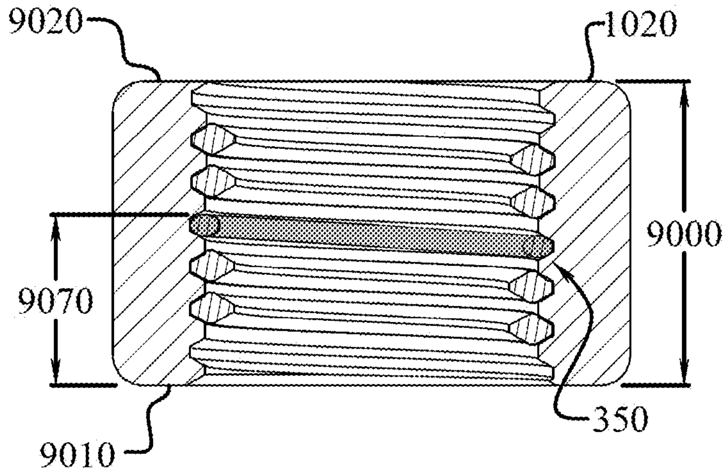

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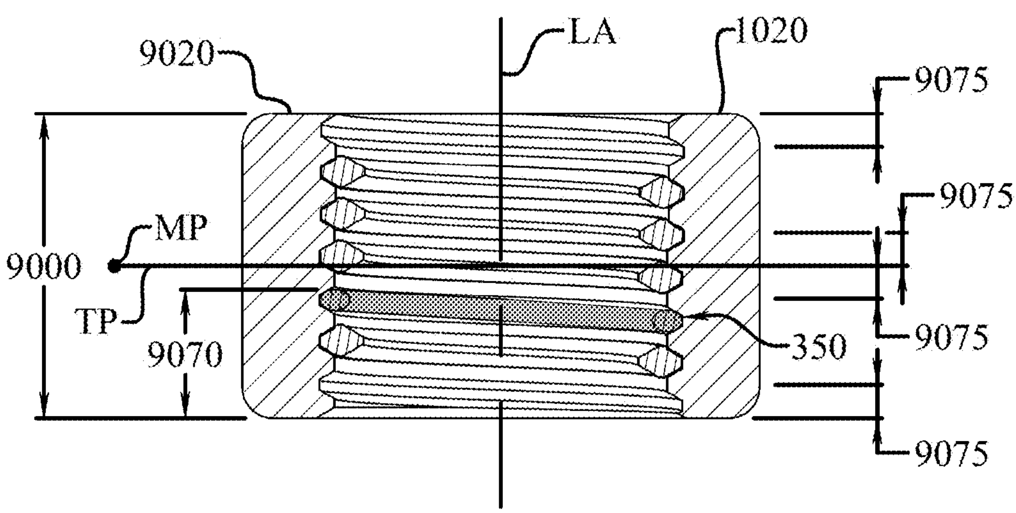

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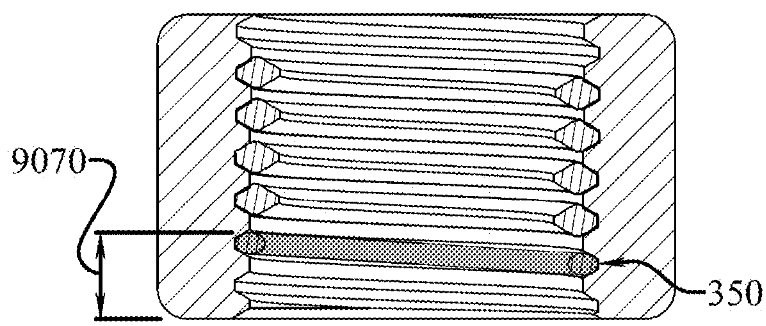

Fig. 30E

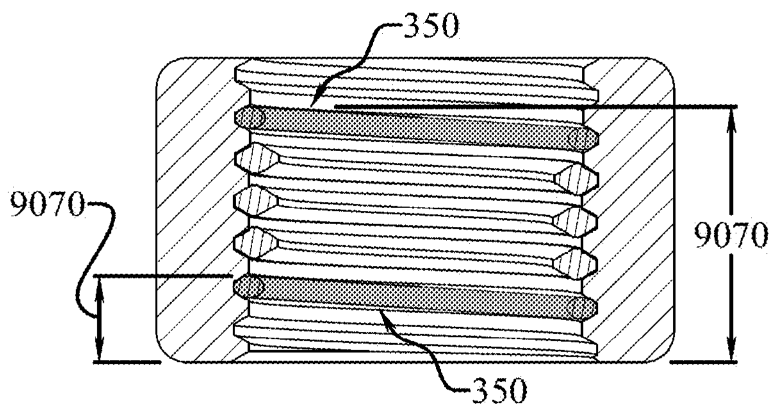

Fig. 30F

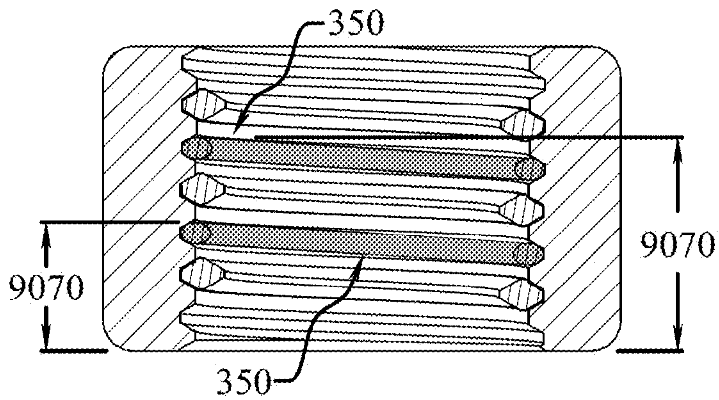

Fig. 30G

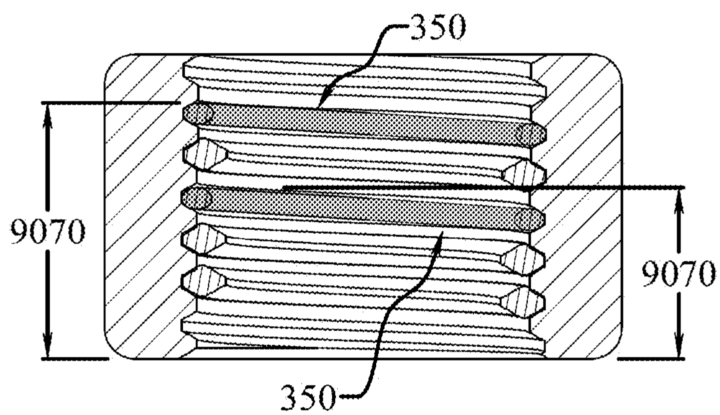

Fig. 30H

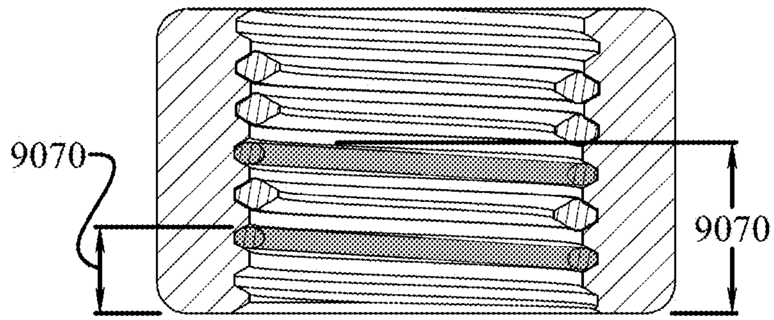

Fig. 30I

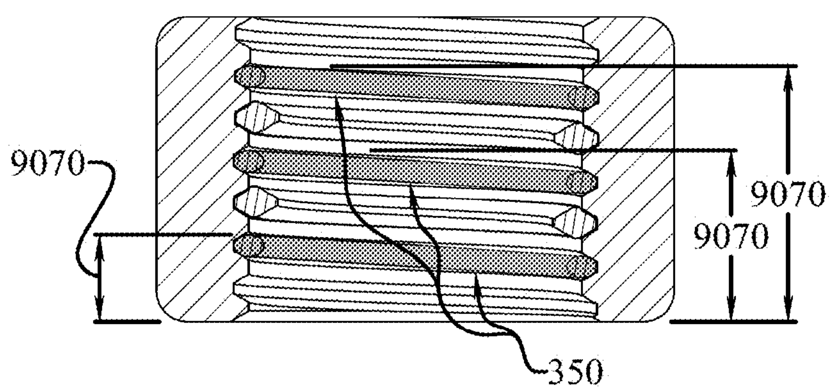

Fig. 30J

PRECISION TORQUE CONTROL POSITIVE LOCK NUT

CROSS-REFERENCE TO RELATED APPLICATIONS

This application is a continuation-in-part application of U.S. patent application Ser. No. 15/906,549 filed Feb. 27, 2018, which claims priority to U. S. patent application Ser. No. 15/595,620, filed May 15,2017, to U. S. Provisional Patent Application Ser. No. 62/553,190, filed Sep. 1, 2017, and to U. S. Provisional Patent Application Ser. No. 62/414, 423, filed Feb. 28, 2017, the disclosures of which are hereby incorporated by reference.

STATEMENT REGARDING FEDERAL GRANTS

Not Applicable.

BACKGROUND OF THE INVENTION

The present disclosure relates to a positive torque locked nut and fastener system with a longitudinally serrated lock nut and removable cap that is used as part of a fastener system to lock a fastener in place. In a preferred embodiment, the locking feature fastener system is utilized in vehicles, such as in aircraft. In an alternative embodiment a helical wire insert is used in conjunction with a dissimilar material to form the locking fastener, such as a polymer fastener body with a steel alloy helical thread system.

Locking fasteners are widely used in attaching equipment to an aircraft fuselage, and for other installations in vehicles, such as cars, agricultural equipment, construction equipment, railroad equipment and the like. In particular, aircraft jet engines are often attached to the airframe with barrel nuts that include a locking feature. Rotating shafts are commonly secured with a positive locking fastener, such as with a cotter pin. In addition, the same or similar fasteners are used in a variety of situations, such as industrial equipment, farm equipment and other equipment where vibration and motion control is required.

Castellated nuts and a compatible cross bore on a threaded shaft have been commonly used for some time to lock a shaft in position by insertion of a pin, (such as a cotter pin), a wire insert, or both. For instance, alternatively, a nut can have a pressed steel castellated cap placed over the nut, and the crennelations in the cap are aligned with a cross bore in the shaft.

Prevailing torque locking fasteners are available that provide for a prevailing torque lock through use of a disk of resilient material. For instance, Vespel (™) inserts are made from a polymide material and are often used with locking or self-locking fasteners. Currently available fastener systems are generally less than fully acceptable because the available locking inserts are expensive and installation of a disk for a locking insert, such as a Vespel (™) insert, is often difficult. Commonly Dupont Vespel SP polymide components are machined or cut into a disk shape and then inserted as a collar around a fastener nut. An additional difficulty in using such inserts is that the bolt fasteners must be driven into the insert to maintain the specified torque tolerance even when used in an environment that imposes a wide range of temperatures and vibration patterns.

Locking fasteners which use inserts such as resilient inserts formed of Vespel (™) have many limitations. Importantly, such inserts are expensive, as the plastic material must be approved by OEM users and the proprietary material in Vespel (™) cannot be substituted by unapproved alternatives from third parties. The use of resilient inserts also has many issues such as a) the inserts are easily-damaged during installation b) the inability to reuse resilient inserts for reinstallation of components, and c) the limitation to the shape of fasteners when using a resilient collar. These current systems are generally limited, and could be substantially improved with an alternative substitute to a resilient insert locking fastener. Another disadvantage of existing systems is the limited number of cycles of insertion and removal that are within specified limits. Furthermore, there is an undesired inconsistency between locking torque values between the early cycles of use, and when the fastener is finally replaced.

Other previous attempts in the aircraft industry to improve on locking fasteners have resulted in a variety of fasteners, each of which have certain limitations. For example, U.S. Pat. No. 5,127,782 issued Jul. 7, 1992 discloses a fastener system as a self locking castellated nut. For instance, the Shur-Lok "Sta-Lok" (T) system is approved for use in aircraft such as helicopters, and utilizes a series of small serrations to hold a fastener nut in place after being torqued to a give specification. For purposes of reliability and safety a positive locking mechanism is considered important, and in some situations essential.

An improved fastener system is desired by manufacturers and retrofitters to reduce the cost of current fasteners, and it is also desirable to enable labor savings along with improved assembly processes, and improved maintainability, reparability, overhauling, fastener reliability and strength.

BRIEF DESCRIPTION OF THE DRAWINGS

For a fuller understanding of the nature and advantages of the present invention, reference should be had to the following detailed description taken in connection with the accompanying drawings, in which:

FIG. **1**A shows an overhead assembled perspective view of a torque control locking fastener;

FIG. **1**B shows an exploded view of a torque control locking fastener;

FIG. **2**A shows alternative side elevation views of a toque control locking fastener;

FIG. **2**B shows a top plan view of a toque control locking fastener;

FIG. **2**C shows a bottom plan view of a toque control locking fastener;

FIG. **3**A shows a perspective view of a serrated nut body for use with the fastener system;

FIG. **3**B shows a top plan view of a serrated nut body for use with the fastener system;

FIG. **3**C shows a cross-sectional view of a serrated nut body for use with the fastener system;

FIG. **4** shows a perspective view of a serrated cap, and compatible fastener for a serrated locking fastener;

FIG. **5**A shows view of an alternative castellated nut for a positive locking fastener;

FIG. **5**B shows a cross-sectional view of a castellated nut for a positive locking fastener;

FIG. **5**C shows an exploded cross-sectional view of a castellated nut for a positive locking fastener;

FIG. **6**A shows a cross-sectional view of the components of a precise torque control fastener;

FIG. **6**B shows an exploded cross-sectional view of the components of a precise torque control fastener;

FIG. **7**A shows a cross-sectional view of an alternative embodiment of a precise torque control fastener;

FIG. **7**B shows a cross-sectional view of an alternative embodiment of a precise torque control fastener;

FIG. **8**A shows a side elevation assembly view of a fastener;

FIG. **8**B shows a side elevation assembly view of a fastener;

FIG. **8**C shows a side elevation assembly view of a fastener;

FIG. **8**D shows a side elevation assembly view of a fastener;

FIG. **8**E shows a side elevation assembly view of a fastener;

FIG. **8**F shows a side elevation assembly view of a fastener;

FIG. **9**A shows a side elevation assembly view of a fastener;

FIG. **9**B shows a side elevation assembly view of a fastener;

FIG. **9**C shows a side elevation assembly view of a fastener;

FIG. **9**D shows a side elevation assembly view of a fastener;

FIG. **9**E shows a side elevation assembly view of a fastener;

FIG. **9**F shows a side elevation assembly view of a fastener;

FIG. **10**A shows a cross-sectional assembly view of a fastener;

FIG. **10**B shows a cross-sectional assembly view of a fastener;

FIG. **10**C shows a cross-sectional assembly view of a fastener;

FIG. **11** shows a top plan view of a fastener;

FIG. **12** shows a side elevation view of a nut cap;

FIG. **13** shows a bottom plan view of a nut cap;

FIG. **14** shows a top plan view of a nut;

FIG. **15** shows a cross-sectional view of a nut;

FIG. **16** shows a bottom plan view of a nut;

FIG. **17** shows a perspective view of a nut;

FIG. **18** shows a cross-sectional view of a nut cap;

FIG. **19** shows a cross-sectional assembly view of a fastener;

FIG. **20** shows a cross-sectional assembly view of a fastener;

FIG. **21** shows a side elevation view of a nut cap;

FIG. **22** shows a side elevation view of a nut;

FIG. **23** shows a cross-sectional view of a nut cap;

FIG. **24** shows a cross-sectional view of a nut;

FIG. **25** shows a side elevation view of a fastener;

FIG. **26** shows a cross-sectional view of a fastener;

FIG. **27**A shows a perspective view of a helical thread insert;

FIG. **27**B shows a side elevation view of a helical thread insert;

FIG. **27**C shows a top plan view of a helical thread insert;

FIG. **27**D shows a cross-sectional view of a helical thread insert;

FIG. **28** shows a perspective view of a helical thread insert;

FIG. **29** shows a top plan view of a locking coil;

FIG. **30**A shows a cross-sectional view of a helical thread insert within a nut;

FIG. **30**B shows a cross-sectional view of a helical thread insert within a nut;

FIG. **30**C shows a cross-sectional view of a helical thread insert within a nut;

FIG. **30**D shows a cross-sectional view of a helical thread insert within a nut;

FIG. **30**E shows a cross-sectional view of a helical thread insert within a nut;

FIG. **30**F shows a cross-sectional view of a helical thread insert within a nut;

FIG. **30**G shows a cross-sectional view of a helical thread insert within a nut;

FIG. **30**H shows a cross-sectional view of a helical thread insert within a nut;

FIG. **30**I shows a cross-sectional view of a helical thread insert within a nut; and

FIG. **30**J shows a cross-sectional view of a helical thread insert within a nut.

SUMMARY OF THE INVENTION

A torque control fastener system having a fastener nut, a helical wire insert, and a shaft, where the nut and helical wire insert have unique hardness and coefficient of thermal expansion relationships that produce improved performance.

DETAILED DESCRIPTION OF THE INVENTION

Disclosed herein is a new apparatus and associated method for securing equipment to an underlying structural support. In particular, disclosed is a locking fastener useful for attaching components in vehicles, engines, and the like to structural members, where attachments may be subject to vibrational loosening.

As disclosed herein such fastener typically is configured to accept a male fastener, and provide for a desired set of torque tolerances including locking, unlocking, and during installation or removal.

FIGS. **1**A, **1**B, and **8**A-**8**F show an assembly and associated views of a locking fastener **1000** that provides an integrated retaining cap and allows for precise control of the applied torque and locking in a preferred position. Such a fastener **1000** can be provided with or without an optional helical thread insert (HTI) **1030**, seen in FIG. **3**C, that can provide for improved thread characteristics or additionally, a prevailing torque feature. The fastener **1000** shown in FIG. **1**B is embodied as a serrated nut **1020** with integrated cap **1040**. Fastener **1000** is essentially comprised of a nut body **1022**, of nut **1020**, and a nut cap **1040**. Fastener **1000** may have an externally circular shape with externally disposed longitudinal serrations **1025** on the exterior perimeter of the nut body **1022** (hidden in FIG. **1**A). A nut cap **1040** is provided with internally disposed serrations compatible with the external perimeter serrations of the nut body, such as the serrations **4010** shown in FIG. **4**.

As shown in FIG. **1**A nut cap **1040** may nest inside a cap retainer **1050**. The illustrated nut cap **1040** is provided with a series of crenellations, or notches, **1044** which allow for a retainer such as a pin or wire, **4090** seen in FIG. **4**, to pass through a cross bore **4088** in an inserted shaft **4080** and lock the nut **1020** in position relative to the shaft **4080**. The nut cap **1040** provides a wrenching interface **1045**, shown in the embodiment in FIG. **1**A as a twelve point nut body. Those skilled in the art will recognize that other nut shapes are applicable to the disclosure. The nut cap **1040** is formed with a shaft bore, **1040**, a portion of which may be formed as a thread bore. The thread bore may be tapped to conform to the desired thread pattern. Threads can be directly cut in a desired thread form or alternatively formed to fit typically available standard thread inserts, for example, STI threads.

FIG. **1**B shows an exploded perspective view of an embodiment of the fastener **1000**. Nut **1020** has a generally cylindrical nut body **1022**, with an internal bore with threads **1024**. Threads **1024** are shown as integrally formed, although STI thread or the like can be provided to allow installation of a helical thread insert **1030**, seen in FIG. **3**C. The exterior surface of the nut body **1022** may be formed with a series concentric serration **1026**, shown as two rings of serrations in FIG. **1**B, and separated by clear rings **1025**. In one embodiment the nut cap **1040** has compatible serrations on the internal surface, so that the internal serrations of nut cap **1040** engage with the external serrations of nut body **1020**.

In an embodiment, as seen in FIG. **1**B, the nut cap **1040** has concentric channels **1047** that hold rings **1046**. A retainer cup **1050** may have compatible rings that allow for the nut cap **1040** to be rotated relative the nut body **1020**.

In the embodiment of FIG. **3**C, as a bolt is inserted into the assembled locking fastener, such as fastener **1000**, rotation of the bolt engages helical insert **1030**, and expands the helical insert **1030** to bear against the inner surface of the thread bore body. As the bolt, aka shaft **4080** of FIG. **4**, advances into the helical insert **1030**, additional segments of the helical insert **1030** are expanded and bear against the thread bore body. In one embodiment, the bolt is advanced entirely through the helical insert **1030** and thread bore body, and is finally torqued to a specified torque. In such case, the characteristics of the helical insert **1030** will serve to retain the bolt in position and resist any backing-out of the bolt to a reverse torque specification determinable according to the characteristics of the helical insert **1030**. Properly installed, the helical insert **1030** will bear against both the bolt and the thread bore body, retaining the inserted bolt.

In an alternative embodiment, the bolt is inserted only a portion of the depth of the helical insert **1030**, and the portion of the helical insert **1030** that is bearing against the bolt and the thread bore body will retain the bolt to a determinable reversing torque. A variety of such specifications and applications are provided by the manufacturer of helical wire inserts, such as from Kato Fastening Systems, Inc.

described in regards to FIGS. **1**A and **1**B, castellated nuts and bearing nuts (for instance) may be provided with a pressed steel nut cap **1040** bearing castellations, utilized the "serration" in the form of the nut exterior perimeter, typically with six points, or possible also eight or twelve points. The nut cap **1040** may have six, eight, ten, or twelve notches. As described herein, the term serration refers to a uniform series of notches, indentation or tooth pockets, so that rotation of a nut **1020** and nut cap **1040** allows for continual matching of compatible serrations, i.e. teeth and notches, when in an engaged position. As shown in regards to FIG. **1**B, the distance between successive teeth around the circumferential perimeter of a nut **1020** is preferably about 1 mm, or about 2 mm. It is preferred that successive teeth be spaced less than about 5 mm apart.

By providing a series of compatible teeth and notches, it is thus not necessary to deviate from a preferred torque on the installed nut **1020**. The ease of the rotation of the nut cap **1040** to align a cross bore **4088**, seen in FIG. **4**, in a shaft **4080** with a castellation, such as **1044** and **4040**, in the nut cap **1040** allows for the preferred torque to be applied to the nut **1020** during installation, and then no backing off of the nut **1020** is necessary to install the lock (whether key, cotter pin, or wire) though the nut cap **1040** and the shaft cross bore **4088**.

A further improvement of the fastener **1000** is to install a helical thread insert **1030**, seen in detail in FIGS. **3**C and **28-30**, in the thread bore **1024** of the nut **1020**. Such a helical thread insert (HTI) **1030** can be either a locking (prevailing torque) or free running insert. Helical inserts, such as Kato brand "CoilThread" (™) inserts, have been used for some time in industry for applications that substantially differ from those disclosed, including, for instance, as a means to repair damaged threads. KATO CoilThread inserts are available in a variety of wound thread sizes including Unified Coarse (UNC), Unified Fine (UNF) and Metric thread sizes. As such the preceding are only some of the variety of helical coil inserts. When helical coil inserts are assembled in "STI" (Standard Thread Insert) tapped holes, helical coil inserts form standardized Unified Coarse (UC) or Unified Fine (UF) threads that conform to National Bureau of Standards Handbook H-28, and meet screw thread standards according to U.S. Federal classification. Helical coil inserts can also be produced that fit a variety of thread standards, such as for instance, will also accommodate UNJ, MIL-S-8879, and male threaded fasteners. Further examples are shown in the 2015 CoilThread Inserts and Tools product catalog of Kato Fastening Systems, Inc. of Newport New, Va. In one embodiment the helical thread insert **1030** undergoes a diameter reduction during installation. The outward spring-like force of the helical thread insert **1030** "locks" the insert into place. Each 360 degree coil of the helical thread insert **1030** can flex independently to contact the greatest amount of thread of the nut **1020** thereby improving both static and dynamic load bearing capabilities of the nut **1020**.

While in a conventional nut assembly the tensile load is spread over one or two threads, with the helical thread insert **1030** each engaged coil acts independently to spread the tensile load out over all of the engaged threads. The helical thread insert **1030** provides an increase in the bearing area and an increase in the ultimate tensile strength of the fastener. In fact, failure testing of a fastener **1000** having a nut **1020** with a 180 ksi tensile rating and utilizing a helical thread insert **1030** in combination with a shaft **4080** having a 220 ksi tensile rating, and a second shaft **4080** having a 260 ksi tensile rating, resulted in failures of the shaft **4080** in both instances, despite it having the higher strength. Further, incorporation of the helical thread insert **1030** produces a reusable prevailing torque fastener. In fact, in one embodiment a 50 cycle test of the locking torque and breakaway torque demonstrated that the breakaway torque remained above **18** in-lbs for all 50 cycles. In fact, in another embodiment the breakaway torque varied by less than 50% from the first cycle to the 50^{th} cycle, and less than 45% in another embodiment, and less than 40% in still a further embodiment. In one embodiment the breakaway torque of all cycles was within 20 in-lbs of the initial breakaway torque, and within 17.5 in-lbs in another embodiment, and within 15 in-lbs in still a further embodiment. The fastener **1000** incorporating the helical thread insert **1030** far exceeded the 30,000 cycle vibration test of NASM 1312-7 and the 800° F. soak torque test of NASM 25027. Additionally, testing of a fastener **1000** incorporating the helical thread insert **1030** in a ¼ aluminum nut **1020** produced a tensile strength exceeding the axial tensile requirement of alloy steel by 23% per NASM 25027 Table 1, at less than ½ of the weight of a comparably sized steel nut, 1.48 grams versus 3.76 grams, while still exceeding the 30,000 cycle vibration test of NASM 1312-7. In one embodiment the nut has a nut mass of no more than 3 grams, and no more than 2.5 grams, 2.0 grams, and 1.5 grams in further embodiment. In another embodiment the helical thread insert **1030** has an insert mass that is at least 20% of the nut mass, and at least 30%, 40%, and 50% in further embodiments. In another series of embodiments the insert mass is no more than the nut mass, and no more than 85% of the nut mass in another embodiment, and no more than 75%, 65%, and 55% in further embodiments.

As noted, testing has been performed of a nut **1020** having the helical thread insert **1030**. In the test samples, a sample nut **1020** was formed of 180 KSI material bored to size and then provided with a stainless steel helical thread insert **1030**. The strength to failure of the improved fastener was compared to two aircraft nut samples that are currently utilized by an aircraft manufacturer. The nut "BACN10HC" is formed of 220 KSI material, while the nut "BACN10ZC" is formed of 180 KSI material. As shown in Table I below, the tested sample fastener, formed as described in the present disclosure, exhibited an unexpectedly high force required to fail the sample nut **1020**. The 180 KSI material forming the sample nut **1020**, with the described helical thread insert **1030**, had a failure strength that was almost indistinguishable from the BACN10HC nut formed of 220 KSI material. In addition, the locking and breakaway torque forces were not substantially reduced over 15 cycles of tightening and loosening. Thus, not only was the strength performance better, but the improved system described herein provides a fastener that can be repeatedly used without substantially decreased performance. Thus, the prototype sample 180 KSI nuts **1020** performed as well as 220 KSI material nuts of BACN10HC. The consistency of the locking torque values over 15 cycles is far superior to the typical performance of Vespel material for the friction locking by common prevailing torque material.

TABLE 1

	BACN10HC (220 KSI)	BACN10ZC (180 KSI)	Sample Nut 1020 with Helical Thread Insert 1030 (180 KSI nut material)
Failure Load (PSI)	37,800	30,700	37,771 (average)
Locking Torque (in-lbs) MIN	150	150	Cycle 1: 81.22 Cycle 15: 61.88
Breakaway Torque (in-lbs) MIN	18	18	Cycle 1: 79.11 Cycle 15: 63.77

An additional series of testing was performed using 220 KSI bolts, aka shafts **4080**, inserted into the fasteners and tested to determine the force necessary to cause the threads to fail under load. Surprisingly, the 220 KSI shaft **4080** failed before the 180 KSI sample nut **1020** with the helical thread insert **1030**. Another test was performed using sample nuts **1020** made of Inconel **718** (material rated also at 180 KSI) and bolts, aka shafts **4080**, rated at 260 KSI with the belief that the failure profile of the sample nuts **1020** could be determined. Surprisingly, once again the 260 KSI rated bolts, aka shafts **4080**, failed before the threads in the Inconel **718** sample nut **1020**. The bolt failure occurred at 265 KSI (44% higher than the Inconel **718** sample nut **1020** rating), yet unexpectedly the 180 KSI Inconel **718** sample nut **1020** with helical thread insert **1030** still did not fail. It is believed that the fastener may still be functional, if the failed bolt shaft could be removed. Thus, testing reveals a significant improvement of the tensile strength performance.

In the above described testing, the breakaway and prevailing torque tests were performed before the tensile failure tests. As expected, the torque values were very consistent; much more consistent over 15 cycles than any other form of prevailing torque fastener currently available. Nuts **1020** manufactured as described with the helical thread insert **1030** maintain a locking torque of within about 50% of the average of the first five cycles, over the last of 15 cycles in one embodiment, and within 40% and 30% in further embodiments. Similarly, nuts **1020** manufactured as described with the helical thread insert **1030** are predicted to maintain a breakaway torque of within about 50% of the average of the first five cycles, over the last of 15 cycles in one embodiment, and within 40% and 30% in further embodiments.

As seen in FIGS. **27**A-**30**J, the helical thread insert **1030** may further include at least one non-round locking coil **350** that has a series of straight segments, seen as **324-329** in FIG. **27**C. A regular coil **360**, or round coil, of the helical thread insert **1030** is defined as one having a full 360 degree circumference of uniform curvature, as seen in FIGS. **27**A-**27**D. Upon entry of the shaft **4080** the straight segments **324-329** of the locking coil **350** are flexed outwardly, creating pressure on the shaft **4080** so that contact area is increased and unit pressure is minimized. FIGS. **27**A and **28** show a perspective views of such a helical thread insert **1030**. Helical thread insert **1030** is formed of polygonal wire, so that the insert **1030** presents external threads **330** and an internally threaded threadbore **320** with internal threads **322**, and locking coil **350** embodiments have a series of at least three faces, **324-329** (for example), which project into the final threadbore **320**, with the embodiment of FIG. **27**C illustrating six faces and the embodiment of FIG. **29** illustrating five faces, however the locking coil **350** may include three or more such faces, including five or more in an embodiment, six or more in another embodiment, eight or more in a further embodiment, and ten or more in still a further embodiment. Performance is diminished, and manufacturing complexity increased, when the locking coil **350** has more than twenty-four faces in one embodiment, and no more than eighteen faces in another embodiment, and no more than twelve faces in a further embodiment, and no more than eight faces in still another embodiment.

The helical thread insert **1030** has as an insert longitudinal axis LA, seen in FIG. **27**B, an insert distal end **332**, and an insert proximal end, which defines an insert length **6000** measured parallel to the insert longitudinal axis from the insert distal end **332** to the insert proximal end **334**. Further, the nut **1020** has a nut longitudinal axis LA, seen in FIG. **30**A, and has a nut threaded length **9000** measured parallel to the nut longitudinal axis from the extreme ends of the threaded portion of the nut **1020**.

The location of the locking coil **350** has unexpectedly been linked to improved performance. With reference again to FIG. **30**A, the nut **1020** has a load bearing end **9010** and a free end **9020**. The shaft **4080** enters the nut **1020** from the load bearing end **9010**. As seen in FIG. **30**A, the nut **1020** may have a cross-sectional shape in a transverse plane TP, which is perpendicular to the longitudinal axis LA, that is constant from the load bearing end **9010** to the free end **9020**. Alternatively, as seen in FIG. **30**B, the nut **1020** may have a variable cross-sectional shape in a transverse plane TP, often incorporating a tool-engagement portion **9040**, often a hex shape, separated from a load bearing portion **9050**, often a round shape, by a transition portion **9060**, or transition from the tool-engagement portion **9040** to the load bearing portion **9050**.

In such an embodiment, locating the locking coil **350** so that it is not within the transition portion **9060** has been shown to improve the stress distribution in the nut **1020**. Stated another way, locating the locking coil **350** so that it is in a location in which the nut **1020** has a transverse cross-sectional shape that doesn't change from a first transverse plane to a second transverse place, where the first transverse plane is located at the point where it just contacts the point on the locking coil **350** that is furthest away from the load bearing end **9010**, which corresponds to the location of the leader associated with **9070** in FIG. **30**A, and the second transverse plane is located at the point where it just contacts the point on the locking coil **350** that is closest to the load bearing end **9010**, which in FIG. **30**A would be just below the arrow head associated with the label **350**. Thus, in such an embodiment the locking coil **350** would be located so that no portion of it falls within the transition portion **9060** of FIG. **30**B. In the illustrated embodiment of FIG. **30**B the locking coil **350** is located entirely between the free end **9020** and the start of the transition portion **9060**, however in another embodiment the locking coil **350** is located entirely between the load bearing end **9010** and the transition portion **9060**.

In a further embodiment having transverse planes TP possessing different cross-sectional shapes, the nut **1020** has a portion that does possess a constant cross-sectional shape in the transverse planes over a constant shape length, and the locking coil **350** is located within the constant shape length. In the embodiment of FIG. **30**B, both the tool-engagement portion **9040** and the load bearing portion **9050** possess a constant shape length, and thus the locking coil **350** is located in either the tool-engagement portion **9040** or the load bearing portion **9050**.

In further embodiments applicable to both constant and variable cross-sectional shape configurations of the nut **1020**, the locking coil **350** is located closer to the free end **9020** than the load bearing end **9010**. In fact, in another embodiment a portion of the locking coil **350** is located a locking offset distance **9070**, measured from the load bearing end **9010**, that is at least 55% of the nut length **9000**, and at least 60%, 65%, and 70% in further embodiments. Unless noted otherwise, the locking offset distance **9070** is the distance from the load bearing end **9010**, or nut proximal end, to the farthest away portion of the locking coil **350**, as seen in FIGS. **30**A-**30**E. In another series of embodiment the locking offset distance **9070** is no more than 95% of the nut length **9000**, and no more than 90%, 85%, 80%, and 75% in further embodiments.

In still further embodiments no portion of the locking coil **350** is within a lock-free zone, which is defined by a predetermined lock-free distance **9075** measured parallel to the longitudinal axis LA from the load bearing end **9010**, the free end **9020**, and/or a transverse plane TP passing through the midpoint of the nut length **9000**, as illustrated in FIG. **30**D. In one embodiment the predetermined lock-free distance **9075** is 5% of the nut length **9000**, and **150**, 15%, 20%, and 25% in additional embodiments. In another series of embodiments the lock-free distance **9075** is no more than 50% of the nut length **9000**, and no more than 40%, 30%, and 25% in further embodiments. The load bearing end **9010** and the free end **9020** may be the nut proximal end or the nut distal end depending on the installation configuration; however to be consistent with the general nomenclature of the illustrated embodiments and FIG. **8** the load bearing end **9010** is the nut proximal end.

The locating of the locking coil **350** within the nut **1020** is further improved, as is the associated stress distribution in the nut **1020**, when the helical thread insert **1030** has a different number of regular coils **360**, i.e. coils having a circular end profile, also referred to as circular coils, on opposite sides of the locking coil **350**. For instance, in one embodiment, one side has at least one additional regular coil **360**, and at least two additional regular coils **360** in another embodiment, as seen in FIG. **30**B, and at least three additional regular coils **360** in still a further embodiment. There are at least two regular coils **360** on each side of the locking coil **350** in another embodiment, and at least one side has at least four regular coils **360** in a further embodiment, and in yet another embodiment one, or both, sides have no more than ten regular coils **360**.

More than one locking coil **350** may be incorporated in a helical thread insert **1030**, as seen in FIGS. **30**F-**30**J, with some embodiments having at least two locking coils **350**, and another embodiment having at least three locking coils **350**; however, further embodiments cap the number of locking coils **350** at no more than one locking coil **350** for every two regular coils **360**, and no more than one locking coil **350** for every three regular coils **360**, and no more than one locking coil **350** for every four, five, six, or seven regular coils **360** in still further embodiments. Further, an insert longitudinal plane exists and contains the insert longitudinal axis LA, seen in FIG. **27**B, and in some embodiments a locking coil cross-sectional shape in the insert longitudinal plane is different than a regular coil cross-sectional shape in the same plane, as illustrated in FIG. **30**B.

A problem overcome by these embodiments originates from the fact that in a conventional nut, without a helical thread insert **1030**, the first thread typically takes on 38% of the load, with the second thread bearing 25% of the load, and the third thread bearing 18% of the load; thus, 81% of the load is distributed across the first three threads. This is not ideal and often necessitates a nut material having a hardness above a Rockwell C hardness of 39 RWC, which would make it prone to hydrogen embrittlement. Additionally, the likelihood of failure of the first thread is increased due to dilation of the nut flange and thread bending due to undesirable stress distribution in a conventional nut. Dilation and thread bending cause the major diameter of the nut thread to increase thus decreasing the shear stress area of the already critically loaded first three threads. Theoretical and practical studies of this phenomenon indicate that the top face of the nut contracts in a radial direction while its bearing surface expands. Thus, without high temper materials to withstand the effects of extreme shear stress on the first three threads, conventional MS21042 nuts will split due to nut dilation and/or stripping. Reference to MS21042 nuts refers to nuts meeting Military Specification MS21042 for use in aircraft. Further, high hardness values require precise processing methods to eliminate hydrogen embrittlement and reduce the fracture toughness.

In fact, testing of standard ¼" 28 thread/inch MS21042 nut was performed with a 180 KSI rated bolt and a 6500 lbf axial load, revealing a first thread load of 2462.4 lbs and a first thread stress of 119534 psi, requiring a Rockwell C hardness of 42 RWC to achieve a shear strength of 110,200 psi, and a ductility of only 13%. Conversely, a test fastener referred to as SCF610-4 having an alloy steel nut, also a ¼" 28 thread/inch nut, tempered to only a Rockwell C hardness of 37 RWC, while utilizing the disclosed helical thread insert **1030**, had a distinctly different failure mechanism when subjected to the same 6500 lbf axial load with a 180 KSI rated bolt. The first thread load of the SCF610-4 test nut was reduced to 1944 lbs with a first thread stress of only 96280 psi, while having an improved ductility of 17% and the

Rockwell C hardness of 37 RWC being well below the hydrogen embrittlement threshold. In contrast to the MS21042 nut, the SCF610-4 test nut flange showed no dilation. The SCF610-4 test nut exhibited a significant reduction in shear stress area occurred near the middle of the nut instead of at the base, as in the MS21042 nut. This is clear indicator of preferred load distribution in the threads of the SCF610-4 test nut. In the SCF610-4 test nut, as the load increased, the middle threads plastically deformed first then the lower threads. Since the SCF610 test nut material is softer and more ductile than MS21042 nut, brittle failure did not occur in the SCF610 test nut, and the lower threads took on more load.

The SCF610 test nuts surpassed the NASM21042 tensile load requirement across the range of critical tolerances. This is important in that it allows a more forgiving manufacturing process that is consequently easier to control. Further, the helical thread insert **1030** provided a tensile strength advantage due to load sharing. Such load sharing allows the SCF610 test nut material to be softer and more ductile than comparable MS21042 nuts, thereby eliminating hydrogen embrittlement issues and the resulting catastrophic failure and FOD issues. Incorporation of the helical thread insert **1030**, in nuts of similar size and performance requirements, also allows corrosion resistant materials such as A286, an iron-nickel-chromium alloy with additions of molybdenum and titanium, which is one of the most popular high temperature alloys, since as an austenitic alloy it maintains good strength and oxidation resistance at temperatures up to 1300° F. Thus, the helical thread insert **1030** facilitates the use of materials having lower tensile strengths in corrosive environments in place of hard alloy steels with expensive cadmium plating and hydrogen bake out processes, which is a tremendous benefit. Therefore, in one embodiment the nut **1020** has no plating.

The tensile strength, hardness, thermal coefficient of expansion, and/or percent elongation relationships among the various components plays a significant role in improved performance, durability, and, in some cases, reusability. Further, the unique combinations and relationships achieve specific performance goals and are much more than just routine experimentation, and, as one skilled in the art will appreciate, often requires careful and deliberate heat treatments processes to achieve the relationships. The disclosed relationships are related to test coupons formed of the same material and subjected to the same heat treatments, hardening, and/or working as the associated component and tested per ASTM E8.

In one embodiment the Rockwell C hardness of the helical thread insert **1030** is greater than the Rockwell C hardness of the nut **1020** and/or the shaft **4080**. In fact, in a further embodiment the Rockwell C hardness of the helical thread insert **1030** is at least 2 units greater than the Rockwell C hardness of the nut **1020** and/or the shaft **4080**, and at least 3 units greater in another embodiment, and at least 4 units greater in still another embodiment. However, further embodiments limit the differential in Rockwell C hardness units to avoid negative effects. Specifically, in one embodiment the difference in Rockwell C hardness units is no greater than 13, and no greater than 10 in another embodiment, and no greater than 7 in still a further embodiment. The helical coil insert **1030** has a Rockwell C hardness of no more than 50 RWC in one embodiment, and no more than 47 RWC in another embodiment, and no more than 45 RWC in still a further embodiment. Whereas in another embodiment the helical coil insert **1030** has a Rockwell C hardness of at least 42 RWC in one embodiment, and at least 43 RWC in another embodiment, and at least 44 RWC, and 46 RWC in still a further embodiments. The nut **1020** and/or shaft **4080** has a Rockwell C hardness of no more than 42 RWC in one embodiment, and no more than 40 RWC in another embodiment, and no more than 38 RWC in still a further embodiment.

Further, in another embodiment the coefficient of thermal expansion of the helical thread insert **1030** is greater than the coefficient of thermal expansion of the nut **1020** and/or the shaft **4080**. In fact, in one embodiment the coefficient of thermal expansion of the helical thread insert **1030** is at least 1.5×10^{-6}/° C. greater than the coefficient of thermal expansion of the nut **1020** and/or the shaft **4080**, and at least 3×10^{-6}/° C. greater in another embodiment, and at least 4.5×10^{-6}/° C. greater in still a further embodiment. However, further embodiments limit the differential in coefficient of thermal expansion to avoid negative effects. Specifically, in one embodiment the difference in coefficient of thermal expansion is no greater than 9×10^{-6}/° C., and no greater than 7×10^{-6}/° C. in another embodiment, and no greater than 5×10^{-6}/° C. in still a further embodiment. The coefficient of thermal expansion of the helical thread insert **1030** is at least 13×10^{-6}/° C. in one embodiment, and at least 15×10^{-6}/° C. in another embodiment, and at least 16×10^{-6}/° C. in still a further embodiment. The coefficient of thermal expansion of the nut **1020** and/or shaft **4080** is no more than 15×10^{-6}/° C. in one embodiment, and no more than 13×10^{-6}/° C. in another embodiment, and no more than 11×10^{-6}/° C. in still a further embodiment.

Additionally, in another embodiment the tensile strength of the helical thread insert **1030** is greater than the tensile strength of the nut **1020** and/or the shaft **4080**. In fact, in one embodiment the tensile strength of the helical thread insert **1030** is at least 10 ksi greater than the tensile strength of the nut **1020** and/or the shaft **4080**, and at least 20 ksi greater in another embodiment, and at least 30 ksi greater in still a further embodiment. However, further embodiments limit the differential in tensile strength to avoid negative effects. Specifically, in one embodiment the difference in tensile strength is no greater than 60 ksi, and no greater than 50 ksi in another embodiment, and no greater than 40 ksi in still a further embodiment. The tensile strength of the helical thread insert **1030** is at least 195 ksi in one embodiment, at least 205 ksi in another embodiment, and at least 215 ksi in still a further embodiment. The helical thread insert **1030** is made of 304 stainless steel in one embodiment, and is made of cold-rolled stainless steel wire in another embodiment.

Still further, the percent elongation of the helical thread insert **1030** is less than the percent elongation of the nut **1020** and/or the shaft **4080**. In fact, in one embodiment the percent elongation of the helical thread insert **1030** is at least 3 percentage units less than the percent elongation of the nut **1020** and/or the shaft **4080**, and at least 5 percentage units less in another embodiment, and at least 7 percentage units less in another embodiment. The term percentage units is used to be clear that the terms refers to a difference between measured % values, not a percentage of one of the measured values. For example, if the percent elongation of the helical thread insert **1030** test specimen is 9% and the percent elongation of the nut **1020** test specimen is 12%, the differential is 3 percentage units. However, further embodiments limit the differential in percentage units to avoid negative effects. Specifically, in one embodiment the difference in percentage units is no greater than 20 percentage units, and no greater than 15 percentage units in another embodiment, and no greater than 12.5 percentage units in still a further embodiment.

The side view in FIG. **27**B shows a generally cylindrical helical thread insert **1030**, with the threads **322** being interrupted by segments of the locking coil **350**, as indicated at **325-327**. An end view of the helical thread insert **1030** is shown in FIG. **27**C, and a cross sectional view in FIG. **27**D. From the top view shown in FIG. **27**C, the internal threads **322** of threadbore **320** combine to create a generally circular end profile, with the locking coil faces **324-329** interrupting the smooth profile of the internal threads **322**. In practice, the characteristics of a helical thread insert **1030** installed in a nut **1020** will serve to retain an inserted shaft **4080** in position and resist any backing-out of the shaft **4080** to a reverse torque specification determinable according to the characteristics of the helical thread insert **1030**. Properly installed, the helical thread insert **1030** will bear against both the inserted shaft **4080** and the threads within nut **1020**, thus resisting the initial back-off movement and further retaining the inserted shaft **4080** against a reverse prevailing torque.

Additional performance benefits have been found to be attributed to the surface roughness of the helical thread insert **1030**. Conventional thinking results in the production of inserts that are exceedingly smooth and described as virtually eliminating friction-induced thread erosion with an average Ra roughness value of 32 μin. For perspective, a cold rolled, heat treated, skin passed stainless steel with a 2B surface finish per publication BS EN 10088-2:2014 "Stainless steels—Technical delivery conditions for sheet/plate and strip of corrosion resisting steels for general purposes," of the British Stainless Steel Association, has a roughness of 0.3-0.5 μm, or approximately 12-20 μin. Further, a bead blasted finish produces an average Ra of 1.00-6.00 μm, or 39-236 μin.

The ultimate test for vibration resistance relative to proof load is the Junkers test. Developed in the late 1960s by German engineer Gerhard Junker, the mechanical testing device measures preload in nut and bolt by means of a load cell. The nut and bolt are subjected to shear loading by means of transverse vibration and proof load is constantly measured. Testing has shown a typical NAS9926 nut with an initial compressive load of approximately 2000 lbf and initial torque of approximately 11 ft-lbf retains only 57-60% of the initial compressive load after 400 vibrational cycles (12.5 Hz, +/−0.026" transverse displacement, 75° F.), and only 37-56% of the initial compressive load after 2000 vibrational cycles. However, introduction of a helical thread insert **1030** in the same size and material nut, and same test conditions, retains 85% of the initial compressive load after 400 cycles (12.5 Hz, +/−0.026" transverse displacement, 75° F.), and 87% of the initial compressive load after 2000 cycles.

Testing has shown thread cycling, defined as assembly and disassembly of the nut and bolt—threaded in, threaded out, significantly impacts the % of the retained load. Specimens were tested at 10, 25 and 50 thread cycles. Specimens that did not contain the helical thread insert **1030** had widely variable retained energy, which is the area under a curve with clamping load retention (%) on the y-axis and vibrational cycles on the x-axis (0-2000). At zero thread cycles the nut containing the helical thread insert **1030** yielded a Junkers test retained energy percentage of 0.86, while the NAS9926 nuts were 0.45-0.59. At 10, 25, and 50 thread cycles the nut containing the helical thread insert **1030** maintained a relatively consistent retained energy percentage of 0.89-0.91, while the NAS9926 nuts had widely variable retained energy percentages of 0.58-0.78. The roughness imparted on the threads of the nut or the helical thread insert **1030** by thread cycling tends to improve the retained energy percentage.

Thus, in one embodiment the threads of the nut **1020** are treated so that a portion of the threads have a roughness of at least 39 μin, and at least 50 μin in another embodiment, and at least 60 μin in still further embodiment. However, another series of embodiments balances the potential negative performance attributes associated with increased roughness by capping the range, thus in one embodiment no portion of the threads has a roughness greater than 200 μin, and no greater than 150 μin, 125 μin, 100 μin, and 80 μin in further embodiments. Likewise, in one embodiment a surface of the helical thread insert **1030** is treated so that a portion has a roughness of at least 39 μin, and at least 50 μin in another embodiment, and at least 60 μin in still further embodiment. However, another series of embodiments balances the potential negative performance attributes associated with increased roughness by capping the range, thus in one embodiment no portion of the helical thread insert **1030** has a roughness greater than 200 μin, and no greater than 150 μin, 125 μin, 100 μin, and 80 μin in further embodiments. In one embodiment the portion of the nut threads having the disclosed roughness is at least 25% of the total surface area of the threads, while in further embodiments it is at least 35%, 45%, 55%, 65%, 75%, 85%, and 95%. Similarly, in one embodiment the portion of the helical thread insert **1030** having the disclosed roughness is at least 25% of the total surface area of the helical thread insert **1030**, while in further embodiments it is at least 35%, 45%, 55%, 65%, 75%, 85%, and 95%. In one embodiment the method of creating the disclosed roughness is via chemical milling, electrical discharge machining, milling, broaching, reaming, electron beam texturing, laser etching and/or texturing, plasma etching, electro-chemical, sanding and/or blasting, ultrasonic polishing, and/or magnetic polishing.

Inserts have been used for some time in industry for applications that substantially differ from those disclosed, including, for instance, as a means to repair damaged threads, and have not recognized the desirable performance benefits associated with the disclosed relationships. The primary purpose of such conventional inserts has been to provide renewed threads after thread damage has occurred. Helical thread inserts are not generally used at all in nut bodies in the new and unique manner disclosed herein to achieve the desired goals. The use separate helical thread inserts **1030** inserted in a nut **1020**, not cast in a body, prior to the present disclosure has been generally disfavored, as the helical thread insert **1030** adds additional complexity, another separate component, and additional cost, not to mention the material treatments necessary to achieve the disclosed relationships. The present disclosure provides a rationale and adaptable design for implementing helical thread inserts **1030** in a nut **1020** to provide increased strength nuts **1020**, with renewable threads, and provides a mechanism for providing a locking or retaining system for nuts **1020** that previously suffered from a number of limitations. The improved nut **1020** disclosed is a heretofore unutilized application of helical thread inserts **1030** to allow for manufacture of a nut **1020** that both provides for a prevailing torque locking fastener, and that increases the useful life of a nut **1020**. Importantly, implementation of the improved nut **1020** with helical thread insert **1030** allows for increased strength of the fastener, in excess of what would be predicted based on the previous understanding of the performance of threaded fasteners, and use of softer more ductile materials. The improved fastener even further allows for nuts **1020** of new and/or uncommon materials, providing weight savings and additional performance enhancements. The present helical thread insert **1030** is not subject to the vagaries of wear commonly encountered with both resilient disk fasteners and with crimped locking fasteners. A nut **1020** with the helical thread insert **1030** experience significantly less permanent alteration when used in service, such that these fasteners can be repeatedly used until a rated cycle life is exceeded. Further, the helical thread insert **1030** may in certain applications allow for renewal after a given number of insertions, or cycles of operation in place. When the design life is due to be exceeded, the helical thread insert **1030** can be removed and renewed without excessive expense.

FIGS. **2**A, **2**B, and **2**C show three directional views of the fastener shown in FIG. **1**. FIG. **2**A shows a side view of fastener **1000**. FIG. **2**B shows a top view, revealing the threadbore, and FIG. **2**C shows a bottom view. In this embodiment the nut cap **1040** nests inside concentric cap retainer **1050**. The nut cap is provided with a series of notches **1044**, and the nut cap **1040** provides the wrenching interface **1045**. The nut **1020** is formed with a shaft bore, **1042**, a portion of which will be formed as a thread bore lined with threads **1024**. Threads **1024** can be formed to fit typically desired standard thread, for example, UNF threads. FIG. **2**C shows a bottom view of the fastener **1000**. Nut **1020** has a generally cylindrical nut body, with an internal bore with threads **1024**. Threads **1024** are shown as integrally formed, although STI thread or the like can be provided to allow installation of a helical thread insert. Retainer cap **1050** has compatible rings that allow for the nut cap to be rotated relative the nut body.

FIGS. **3**A, **3**B, and **3**C show detailed views of nut **1020**. In FIG. **3**A, threads **1024** of nut body **1022** line the thread bore **1028**. Nut body **1022** is provided with at least one ring **1025** of locking serrations, or teeth and notches around the nut body perimeter. The illustrated embodiment has two rings **1025** that are separated by a smooth rotation ring **1026**. The illustrated embodiment has two smooth rotation rings **1026** that are slightly smaller diameter than the locking serration rings **1025**, so that when the nut cap serration rings are aligned with the nut body rotation rings, the nut cap **1040** can be rotated. FIG. **3**B shows a top view of nut **1020**, along with the notches and teeth ring **1025**, threadbore **1028**, lined by threads **1024**. FIG. **3**C shows a cross section of nut body **1020**. FIG. **3**C shows nut body **1020**, with external locking serration rings **1025** and rotation rings **1026**. Fastener nut body **1020** is threaded, as shown in FIG. **3**C, with STI threads **1024** and the thread bore **1028** is provided with a helical insert **1030** that occupies the threads of the fastener. Fastener nut body **1022** thread bore **1028** is shown filled by a helical insert **1030**. Helical insert **1030** may be selected from a wide range of available helical inserts to occupy the depth of thread bore **1028**. The helical insert **1030** can be either a prevailing torque insert (as shown) or free running. An installed helical insert **1030** can be held in position by a retainer, as shown by detents **1032**.

FIG. **4** shows an exploded perspective view of the set of components utilized with the disclosed system. Fastener **4000** is comprised of is a compatibly serrated removable cap **4050** for use with a version of the fastener, and a threaded locknut nut **4020**. (See also fastener **1000** of FIG. **1**). Cap **4050** has an externally hexagonal shape as is common for machine nuts, with faces **4002**-**4007**. Cap **4050** is formed with a bore **4010**. Bore **4010** is formed with vertical internal serrations **4040**. Bore **4010** also has tab extensions **4030** with lock slots **4042**, **4044** between tabs **4030**.

In this embodiment the nut **4020** is externally serrated, as at **4060**, and externally compatible with cap **4050**, and is internally threaded to be compatible with a mounting shaft, such as shaft **4080**, or with a bolt or stud. A distal bearing face **4056** bordered by clear ring **4052** will bear against a surface associated with shaft **4080** (or with a pulley, or support brace, for instance) and proximal bearing face **4054** bears against the inside of cap **4000**.

Threads **4058** of nut **4020** are compatible with threads **4082** on shaft **4080**. Shaft **4080** is also provided with a cross bore, such as bore **4088**, so that a lock can be passed through the shaft to lock it into place. In order to lock the shaft **4080** and fastener **4000**, the fastener **4000** (of nut **4020** and cap **4050**) is driven onto shaft **4080**, and tightened to a specified torque. The cap **4050** is then slipped off the nut **1020**, and the cross bore **4088** is aligned with the lock slots (i.e. **4040**) on the cap **4050**. As shown in FIG. **4**, two lock slots **40404042** are provided. Those skilled in the art will recognize that 4 or more lock slots are considered useful for particular applications, and are in keeping with the present disclosure. Following proper torque application, a pin or otherwise compatible lock or wire lock, such as the pin end **4092** of cotter pin **4090**, can fit through the aligned lock slot **4040**, pass through aligned bore **4088** and lock slot **4042**, lock the cap **4050** in place, relative to the fastener **4000** and the shaft **4080**.

The internal serration effective diameter of cap **4050** will match the external serration effective diameter of nut **4020** such that cap **4050** will fit over nut **4020** with sufficient clearance between serrations **4010** on the cap and serrations **4060** on the nut **4020** to allow for ease of installation, and removal. Tolerance is minimized to prevent advancement over the serrations. In one embodiment of the disclosure, cap **4050** is placed over nut **4020** and installed together, to thread, wrench, and properly torque fastener **4000**. Cap **4050** is then removed and remounted to properly align slots **4040** with a keyhole in the shaft **4080** onto which fastener **4000** is threaded. Cotter pin **4090** is then inserted through slots **4040**, the keyhole bore **4088** in the shaft, and then slot **4042** (for instance) on the opposite side of cap **4050**. Thus, a precise torque can be imposed on the nut, and the torqued nut need not be advanced or loosened in order to add a positive lock. Previous positive locking nut cap combinations do not allow for application of a precise torque because the locking crennelations seldom are coincident with the specified torque position.

For certain applications, two pin slots **4040** are desired. In other applications, 4, 6 or 12 slots may be desirable, with a limit to the number of slots **4040** reached when the tabs **4030** are no longer sufficiently robust to limit the reverse torque that may be applied to the mounted fastener **4000**.

FIGS. **5**A, **5**B, and **5**C show views of an alternative embodiment fastener **5000** without a separate serrated removable cap as shown in regards to fastener **1000** of FIG. **1**. FIG. **5**A shows a tip perspective view, FIG. **5**B shows a cross section through plane B-B, and FIG. **5**C shows an exploded cross section. Fastener **5000** has an externally hexagonal shape as is common for machine nuts, with faces **5002**-**5007**. Again, those skilled in the art will recognize that other nut shapes are applicable to the disclosure. Fastener **5000** is formed of nut body **5020** with a bore **5010**. Bore **5010** of nut body **5020** also has tab extensions **5030** with slots **5040** between tabs though which a cotter pin or retaining wire can fit. Tab extensions **5030** are formed as part of the nut body **5020**, and must be strong enough to resist a nominal reverse torque to safely lock the fastener in place when a key, pin or wire is installed. As shown in FIG. **5**A, twelve tab extensions are provided, and tab extensions alone are preferably twelve, sixteen or eighteen in number, and should be aligned in an opposite manner so that a pin can pass straight through a slot, a cross bore and an opposite slot.

The internal threads of fastener **5000** can be formed to be compatible with a helical thread insert, such as helical insert **5012**. Helical insert **5012** is shown as a free running insert, and may be trapped in the threadbore **5010** by thread perturbations, such as detents **5014**, **5016**. As in FIG. **4**, a cotter pin can be inserted through slots **5040**, the keyhole in the shaft, and then an opposite slot on the opposing side of fastener **5000**. Thus, the nut is precisely torqued and then locked in place.

Existing locking fasteners are often characterized as either "positive locking" or a "prevailing torque" locking fastener. In a positive locking fastener, the threaded on portion of the fastener, typically a nut, is mechanically held in its prescribed position by some type of mechanical locking feature. As shown above, or the nut to be released, or backed off from its specified final position, in a positive locking fastener, some mechanical failure must occur, such as shearing of metal, or displacement of retainer pin for the nut to move.

A prevailing torque mechanical fastener utilizes a specified torque or opposed frictional force to lock the fastener in place. Plastic inserts, such as a Vespel insert in a nut, offset locking washers, or crimped deformation fasteners are common examples of prevailing torque locking fasteners. As disclosed herein, the helical insert **5012** functions as a prevailing torque locking fastener. It can also be combined to add a positive lock as shown. Such lock can be a cotter pin or alternatively a lock or wire.

FIGS. **6**A and **6**B show cross sections to demonstrate the retained cap embodiment for use with a fastener such as fastener **1000** of FIG. **1**. Cap **1042** has an externally hexagonal shape. Cap **1042** is formed with bore and formed with vertical serrations **1035**, and clearance rings **1036**, seen in FIG. **6**B. Bore **5010** also has twelve tab extensions **1040** with slots **1044** between tabs **1040** though which a retaining pin can fit.

FIG. **6**A shows a cross section of the components of the precise torque control locking fastener with the cap **1042** of the fastener seated on the nut body, so that the serrations are engaged. FIG. **6**B shows the fastener **1000**, with the cap retracted, releasing the cap for rotation relative to the nut **1020**. Retainer **1058** can be implemented to prevent the cap from separating from the nut body, limiting the possibility of foreign object damage when a part is separated. Gasket **1046**, along with ring **1047** are formed to limit passage of lip **1054**. Clear space **1056**, along with land **1055** allow the cap **1040** to rotate while being retained by retainer **1058**.

Another existing type of locking nut fastener comprises a nut that has been provided with a thread barrel that is a shape other than round, in particular, an oval thread barrel. One current method of creating an oval thread barrel is to distort, or "crimp" a circular cross section nut barrel to a specified torque, distorting the round cross section to an oval cross section. Such crimped fasteners can function as a prevailing torque locking fastener, but have a number of limitations. These limitations include the difficulty in starting the crimped nut on the thread of a bolt, due to the distortion of the circular cross section. Nuts which are crimped at the time of use may be essentially destroyed by improper or over crimping. Furthermore, it is difficult to reproducibly create a desired fastener that performs within a narrow desired range of prevailing torque. In these fasteners, the amount of back-off resistance (i.e. the prevailing torque of the fastener) is difficult to control and lacks consistency between different lots of crimped fasteners, and between installation events or between different technician installers. See for instance, Barrett, R. T., "Fastener Design Manual," NASA Reference Publication 1228, Mar. 1990.

Another type crimped fastener utilizes three-point crimping (usually used on a larger sizes of nuts). Theoretically more points for crimping are possible (for example four or more).

It is a further embodiment of the disclosed apparatus or device is use of a helical insert as a locking feature for female self-locking fasteners in lieu of other traditional methods such as crimping (oval and three or more points) in order to deliver more consistent torque performance of the fasteners within the specimens of a given production batch. Such use of the new system provides for a reduced scrap rate of fasteners, better maintainability of installed fasteners, and less risk of material performance issues such as micro crack or hydrogen embrittlement for instance. Implementation of the disclosure allows for the elimination negative production issues, such as double crimping, unnecessary additional sorting or the like.

The helical wire insert **5012** of the current disclosure can be a full substitute for crimped locking fasteners, and minimize the existing problems with starting the fastener on a threaded shaft caused by the tolerances resulting from crimping of the fastener into an oval shape.

A further embodiment of the disclosure is the use of helical locking inserts in applications that require high strength fasteners, such as 220 KSI rated12 point nuts. In substituting for the six point fastener shown in FIG. **1**, a 12 point nut can be provided with a helical coil insert **5012** that nests within a provided coil pocket of the thread barrel. FIG. **4** shows a perspective view of the helical insert locking nut **4000**, with the nut cap **4050** formed as a six sided nut with six driving faces **4002**-**4007**. It will be apparent to those skilled in the art that a variety of other shapes of driving faces can be provided, such as 8, 12, or some other variation from a regular polygonal shape.

FIGS. **7**A and **7**B show an alternate embodiment of the torque control locking nut. In FIG. **7**, the locking nut **7000** is comprised of three primary structural components, nut body **7020**, lock cap **7040**, and retainer **7050**. Nut body **7020** is an internally threaded (as at **7023**) hollow cylinder, and is configured with external serrations generally at **7025**.

The nut body **7020** surrounds and generally defines a bore, with said bore accommodating a shaft, such as an externally threaded shaft compatible with threads **7023**.

Lock cap **7040** has a sleeve portion and a seat body **7046**, with internal serrations **7035**. Internal serrations **7035** mate in a nesting fashion with the external serrations **7025** of nut body **7020**. Said serrations do not necessarily have an identically compatible structure, but must allow for slidable engagement of the compatible serrations. Nut body serrations **7025** may number 24 or 48 individual teeth, proportionally arrayed about the out surface of the nut body. The internal serrations of the lock cap thus must be compatibly arranged, for instance with 48 or 96 proportionally arrayed individual teeth. In such a manner, either the nut body **7020** or the cap **7040** could be provided with more teeth to allow very fine control over the positioning of a nut about a threaded shaft, while not sacrificing overall locking nut strength.

Lock cap sleeve portion is provided with a cross bore **7044** which allows for insertion of a locking pin, such as a

Cotter key, retaining wire or the like. As previously described, insertion of a threaded shaft into bore allows the lock cap to be threaded onto the shaft generally longitudinally to arrow **7002** in FIG. **7**A, with the nut body freely advancing until the nut body seat is in contact with a bearing surface, whereupon the torque required to advance the nut body along a shaft increases. With the lock cap **7040** engaged about the serrations mating between the nut body **7020**, the fastener can be advanced until a precise predetermined torque specification is reached. There is no immediate need to align the cross bore **7044** with a compatible cross bore in the inserted shaft. (Refer to the disclosure in relation to FIG. **5**) Once the predetermined torque is reached, the lock cap **7040** is retracted from engagement with the serrations on nut body **7020** by retracting the lock cap in the direction of arrow **7004**. Because the nut body **7020** is fully seated and torqued to specification, the nut body **7020** remains in place and the lock cap **7040** and nut body **7020** separate from one another.

As shown in FIG. **7**B, and the lock cap **7040** can be rotated about the shaft axis until the lock cap cross bore **7044** aligns with a cross bore in the inserted shaft. When the shaft cross bore and the lock cap cross bore **7044** align, the lock cap **7040** is advanced in the direction of arrow **7002** in FIG. **7**A, engaging the serrations **7025** and **7035**. Once the locking pin is inserted and secured, the fastener **7000** is held at precisely the predetermined torque until such time as the locking pin is removed and the fastener is counter rotated.

The third component of the locking nut **7000** is retaining collar **7050**. Retaining collar **7050** retains the lock cap **7040** and nut body **7020** in association with one another. When lock cap **7040** is urged in the direction of arrow **7002** in FIG. **7**A, the nut body **7020** nests within the seat body of lock cap **7040**. When the lock cap is withdrawn from association with nut body **7020**, by moving the lock cap **7040** in the direction of arrow **7004** in FIG. **7**B, the retaining collar **7050** allows disengagement of the nut body **7020** and lock cap **7040**. As the disengagement occurs, the withdrawing lock cap **7040** is prevented from full separation from the nut body **7020** by the lock cap shoulder **7046** encountering the retaining collar shoulder **7056**. Retaining collar **7050** is further retained in association with the nut body **7020** and the lock cap **7040** by retaining collar heel **7058**, which is concentrically circumferential with the bore, the nut body **7020**, and the lock cap **7040**.

In yet another embodiment of the present disclosure, the helical insert can be utilized with barrel nuts. Barrel nuts are widely used in attaching equipment to an aircraft fuselage. In particular, aircraft jet engines are often attached to the airframe with barrel nuts that include a locking feature. In another example, isolator mounts are produced by the Lord Corporation of Cary, North Carolina In addition, the same or similar fasteners are used in a variety of situations, such as industrial equipment, farm equipment and other equipment where vibration and motion control is required.

These current systems are generally unacceptable because of the expense of the locking inserts and difficulty in installing the locking Vespel insert. An additional difficulty in using such inserts is the need for the bolt fasteners driven into the insert to maintain the specified torque tolerance when in use in an environment that imposes a wide range of temperatures and vibration patterns. As such a locking mechanism is considered important. Currently, the only effective locking or retaining system available for floating inserts is a collar made of resilient material, such as Dupont Vespel (™).

Insertion of a threaded male fastener into the compressed helical insert will cause the spring nature of the compressed insert to resist the anti-rotation of the inserted fastener. Selection of appropriate threads and wire insert can be used to meet particular torque specifications.

It should be recognized that the fasteners system disclosed is applicable to a method of attaching components by providing a fastener that includes a thread bore internally threaded to accept a helical wire insert, inserting a helical wire insert with an external thread that mates with the internal threads of the thread bore, and internal threads of the insert that are compatible with an externally threaded bolt and capable of being driven by a given torque into the helical wire insert, with the helical wire insert resisting the backing out of the driven insert with a torque greater that the given torque for driving the threaded shaft into the helical wire insert.

FIGS. **8**A-**30** show additional embodiments of the fastener system, which is (a) a combination positive locking and prevailing torque fastener system when a helical insert **1030** is used in conjunction with a positive locking nut cap **1040**, versus (b) a positive locking fastener system when used without the helical insert **1030**, versus (c) simply a prevailing torque fastener system when used with the helical insert **1030** but without the positive locking nut cap **1040**, or with a nut cap that does not incorporate the positive locking features. Thus, all disclosure applies to all variations and need not include each and every one of the disclosed components or features. As a preliminary note, references to a distal end of a component or the system refer to those aspects nearest the top of FIG. **8**A, while references to a proximal end of a component or the system refer to those aspects nearest the bottom of FIG. **8**A, unless noted otherwise.

With reference first to the nut **1020**, seen best in FIGS. **17** and **15**, it has a nut distal end **1021** and a nut proximal end **1023**, thereby defining a nut length **1027**. The nut **1020** has a nut bore **1028** extending into the nut **1020** from the nut proximal end **1023**, and may extend all the way through the nut **1020** to the nut distal end **1021**, or only a portion of the nut length **1027**, and at least a portion of the nut bore **1028** contains a plurality of nut bore threads. The nut **1020** may include a plurality of nut slots **5040** extending from the nut distal end **1021** toward the nut proximal end **1023**, and having a nut slot depth **5041** and nut slot width **5042**, as seen in FIG. **15**. One such embodiment includes at least two nut slots **5040**, while another embodiment includes at least four, and the illustrated embodiment includes at least six. One embodiment has a slot separation angle **5043**, seen in the top plan view of FIG. **14**, that is no more than 90 degrees, and is 30-60 degrees in a further embodiment.

The plurality of nut slots **5040** provide a mechanism to rotationally lock the nut **1020** to the nut cap **1040** when the two are engaged in a predetermined longitudinal position with respect to one another, or when the nut **1020** is used without a nut cap **1040** the nut slots **5040** may receive a portion of the pin **4090** to mechanically secure the nut **1020** to the shaft **4080** in a manner similar to the embodiment of FIG. **5**A. In the former such embodiments, the nut cap bore **8030** may include nut cap projections **8060**, such as those seen in FIG. **18**, to cooperate with at least one of the nut slots **5040**. Alternatively, or in addition to the nut slots **5040**, the mechanism to rotationally lock the nut **1020** to the nut cap **1040** may include a plurality of nut locking serrations **1025** on the exterior surface of the nut **1020**, which may be configured in a single perimeter ring extending from the nut proximal end **1023**, as seen in FIG. **17**, or a plurality of spaced apart rings, as seen in FIG. 1B. As previously disclosed, such nut locking serrations **1025** are configured to cooperate with a plurality of cap serrations **8050** on the interior surface of the nut cap **1040**, which likewise may be configured as a single ring extending from the nut cap proximal end **8003**, as seen in FIG. **18**, or a plurality of spaced apart rings, as seen in FIG. **6**B. In one embodiment the plurality of nut locking serrations **1025** includes at least 16 individual serrations spaced about the perimeter, and at least 20, 24, 28, 32, 36, and 40 in further embodiments. In a further embodiment the plurality of nut locking serrations **1025** includes no more than 72 individual serrations spaced about the perimeter, and no more 68, 64, 60, 56, and **52** in further embodiments. Similarly, in another embodiment the plurality of cap serrations **8050** includes at least 16 individual serrations spaced about the perimeter, and at least 20, 24, 28, 32, 36, and 40 in further embodiments. In a further embodiment the plurality of cap serrations **8050** includes no more than 72 individual serrations spaced about the perimeter, and no more 68, 64, 60, 56, and 52 in further embodiments. In another embodiment the greatest radial dimension from a peak to a valley of the nut locking serrations **1025**, and/or the cap serrations **8050**, is 1.50 mm, and 1.25 mm, 1.00 mm, 0.75 mm, and 0.50 mm in further embodiments. Further, the peak to peak separation distance between adjacent nut locking serrations **1025**, and/or the cap serrations **8050**, is no more than 5.0 mm, and no more than 4.0 mm, 3.0 mm, 2.0 mm, and 1.0 mm in further embodiments. In still another embodiment the greatest radial dimension from a peak to a valley of the nut locking serrations **1025**, and/or the cap serrations **8050**, is less than the peak to peak separation distance between adjacent nut locking serrations **1025**, and/or the cap serrations **8050**.

The nut slot depth **5041** is preferably at least 10% of the nut length **1027**, and at least 15%, and 20% in further embodiments. However, additional embodiments limit the nut slot depth **5041** to balance the tradeoffs of rotational resistance strength, weight savings, durability, fatigue resistance, and longitudinal load bearing capability, by limiting the nut slot depth **5041** to no more than 60% of the nut length, and no more than 50%, 40%, and 30% in further embodiments. Further embodiments recognize desirable and unique performance, while balancing the mentioned tradeoffs, by having the nut slot width **5042** less than the nut slot depth **5041**, and at least 10% less, 20% less, and 30% less in further embodiments. Similarly, a longitudinal length of the threaded portion of the nut bore **1028** is greater than the nut slot depth **5041**, and is at least 20% greater, and at least 40% greater in further embodiments.

The portion of the nut **1020** between the nut slots **5040** create a plurality of nut tab extensions **5030**, as seen in FIGS. **15**, **17**, and **5**B. Each nut tab extension **5030** has a tab length **5031**, seen in FIG. **15**, and a nut tab sidewall thickness **8036**. Again, balancing the aforementioned tradeoffs, desirable and unique performance is found when the tab length **5031** is greater than the nut slot depth **5041**, and at least 20% greater, 40% greater, and 60% greater in further embodiments. In a further embodiment desirable benefits are achieved with the tab wall thickness **8036** is less than a maximum nut sidewall thickness **8038**, and at least 10% less, 25% less, and 40% less in further embodiments; however, additional embodiments further balance the tradeoffs by limiting the relationship so that the tab wall thickness **8036** is no more than 80% less than the maximum nut sidewall thickness **8038**, and no more than 70% less, and 60% less in further embodiments. Further, the tab wall thickness **8036** may vary to provide additional rigidity and a contact surface near the nut distal end **1021** that may also engage an inner surface of the nut cap **1040**. Thus, in one embodiment the nut tab extension **5030** has a portion nearer the nut proximal end **1023** with a tab sidewall thickness **8036** that is less than the tab sidewall thickness **8036** at a point hearer the nut distal end **1021**, such as that seen in FIG. **15**.

As seen in the bottom plan view of FIG. **16**, in one embodiment the outermost surface of the plurality of serrations do not extend beyond an outermost surface of the nut tab extensions **5030**. Thus, in the illustrated round embodiment, with reference to FIG. **22**, a maximum tab portion dimension **5032** is at least 2% greater than a minimum tab portion dimension **5033**, and at least 3.5% greater, and at least 5% greater in further embodiments. Further, in another embodiment, a maximum serrated portion dimension **1029** is no greater than the maximum tab portion dimension **5032**, however the maximum serrated portion dimension **1029** is greater than the minimum tab portion dimension **5033**, and at least 2% greater, 3.5% greater, and 5% greater in additional embodiments. Additionally, a serrated portion length **1019**, seen in FIG. **22**, is no more than 80% of the nut length **1027**, and no more than 60%, and no more than 40% in further embodiments; however, in further embodiments the serrated portion length **1019** is at least 10% of the nut length **1027**, and at least 20%, and 30% in additional embodiments. Further, the sum of the serrated portion length **1019** and the nut slot depth **5041** are no more than 80% of the nut length **1027**, and no more than 70%, and no more than 60% in further embodiments. Additionally, in another embodiment the serrated portion length **1019** is no more than 50% greater than the nut slot depth **5041**. Incorporation of the plurality of nut slots **5040** at one end of the nut **1020**, and the plurality nut locking serrations **8032** at the other end of the nut **1020**, provides better durability, fatigue resistance, stress distribution, and a failsafe mechanism at the rotational locking interface between the nut **1020** and the nut cap **1040**. Similarly, in another embodiment the nut slot width **5042** is no more than twice the separation distance between adjacent nut locking serrations **8032**, and is no more than 70% greater than the separation distance between adjacent nut locking serrations **8032**, and no more than 50% greater in still further embodiments.

Now focusing on the nut cap **1040** embodiments of FIGS. **18** and **21**, the nut cap **1040** has a nut cap proximal end **8003** and a nut cap distal end **8001**, thereby defining a nut cap length **8005**. As seen in FIG. **21**, the nut cap **1040** has a nut engagement portion **8010**, having a nut engagement portion length **8012** and a nut engagement portion width **8014**, and a positive locking portion **8020**, having a positive locking portion length **8022** and a positive locking portion width **8024**. The nut cap **1040** has a nut cap bore **8030** extending into the nut cap **1040** from the nut cap proximal end **8003**, and in some embodiments extending all the way through the nut cap **1040** to the nut cap distal end **8001**. The nut cap bore **8030** has a nut cap bore width **8032**, which may be a diameter in the case of a round nut cap bore **8030**. The nut cap distal end **8001** may be open to allow the shaft **4090** to pass all the way through the nut cap **1040**, as illustrated, or it may be closed.

A portion of the nut cap bore **8030**, within the nut engagement portion **8010**, includes a rotation prevention structure that engages the nut **1020** to prevent rotation when engaged, but allows for rotational movement when disengaged by longitudinal movement of the nut **1020** and nut cap **1040**. The rotation prevention structure may include a plurality of cap serrations **8050** and/or at least one nut cap projection **8060**, as previously mentioned with respect to FIG. **18**. The plurality of cap serrations **8050** have a cap serration length **8052**, and the nut cap projection **8060** has a nut cap projection length **8062**. In one embodiment the cap serration length **8052** is at least 10% of the nut engagement portion length **8012**, and at least 20%, and at least 30% in further embodiments. Similarly, in another embodiment the cap serration length **8052** is at least 5% of the nut cap length **8005**, and at least 10%, and at least 15% in further embodiments. Additionally, another series of embodiments limits the cap serration length **8052** so as to not require complete separation of the nut **1020** and nut cap **1040** in order to rotationally reposition them with respect to one another, as previously explained in the disclosure of the cap retainer **1050**. Thus, in one embodiment the cap serration length **8052** is no more than 80% of the nut engagement portion length **8012**, and no more than 65%, and no more than 50% in further embodiments. Similarly, in another embodiment the cap serration length **8052** is no more than 60% of the nut cap length **8005**, and no more than 50%, and no more than 40% in further embodiments.

With reference to FIGS. **18** and **21**, the positive locking portion **8020** may be formed with a plurality of locking apertures **8040**, as opposed to, or in addition to, the previously disclosed slots **1044**, **4040** seen in FIGS. **1-7**. In fact, all of the disclosure related to the slots **1044**, **4040** applies equally to the locking apertures **8040**. Each locking aperture **8040** has an aperture length **8042** and an aperture width **8044**, seen in FIG. **21**, which may be equal. Further, each locking aperture **8040** is located an aperture offset **8046** that is the distance, parallel to a longitudinal axis of the nut cap bore **8030**, from the nut cap distal end **8001** to the nearest point of the locking aperture **8040**, as seen in FIG. **23**. Each locking aperture **8040** is separated from the adjacent locking aperture **8040** by an aperture separation distance **8048**, shown in FIG. **21**. The aperture separation distance **8048** is at least 50% of the aperture width **8044**, and in further embodiments the aperture separation distance **8048** is at least 70% of the aperture width **8044**, 90%, 100%, and 110%. In a further embodiment the aperture separation distance **8048** is at least 50% of the aperture length **8042**, and in further embodiments the aperture separation distance **8048** is at least 70% of the aperture length **8042**, 90%, 100%, and 110%. In another embodiment the aperture separation distance **8048** is at least 5% of the nut engagement portion width **8014**, and at least 10%, 15%, and 25% in additional embodiments. Similarly, in another embodiment the aperture separation distance **8048** is at least 10% of the positive locking portion width **8024**, and at least 15%, 20%, and 25% in additional embodiments.

One embodiment includes at least two pairs of locking apertures **8040**. Within each pair the locking apertures **8040** are located 180 degrees from one another and each locking aperture **8040** within the pair has the same aperture offset **8046**. However, each pair has a different aperture offset **8046**. Thus, a first pair has a first aperture offset and the second pair has a second aperture offset that is greater than the first aperture offset. In one embodiment the second aperture offset is at least 15% greater than the first aperture offset, and at least 20%, 25%, 30%, 40%, 50%, 60%, and 70% greater in additional embodiments. In another series of embodiments the second aperture offset is no more than 400% greater than the first aperture offset, and no more than 300%, 250%, 200%, 150%, 100%, and 80% greater in additional embodiments. In another embodiment the first pair is radially offset no more than 90 degrees from the second pair, and in a further embodiment the first pair is radially offset no more than 60 degrees from the second pair, and in yet another embodiment the first pair is radially offset no more than 45 degrees from the second pair.

Another embodiment further includes a third pair of locking apertures, having a third aperture offset that is greater than the second aperture offset. In one embodiment the third aperture offset is at least 15% greater than the second aperture offset, and at least 20%, 25%, 30%, 40%, 50%, 60%, and 70% greater in additional embodiments. In another series of embodiments the third aperture offset is no more than 400% greater than the second aperture offset, and no more than 300%, 250%, 200%, 150%, 100%, and 80% greater in additional embodiments. In another embodiment the second pair is radially offset no more than 90 degrees from the third pair, and in a further embodiment the second pair is radially offset no more than 60 degrees from the third pair, and in yet another embodiment the second pair is radially offset no more than 45 degrees from the third pair. Further embodiments include at least four pair of locking apertures **8040**, and five pair, and six pair in additional embodiments.

As one skilled in the art will appreciate, the aperture offset **8046** disclosure applies equally to embodiments containing slots rather than apertures. This is illustrated for convenience in FIG. **26** with respect to the embodiment of FIG. **6**B but applies to all disclosed embodiments. Such slot embodiments have a slot bottom offset **8047**, seen in FIG. **26**, rather than an aperture offset **8046**, and the slot bottom offset **8047** value is measured in the same manner as the aperture offset **8046** but to the deepest portion of the slot **1044**; which also applies to slots **5040** formed in nuts such as embodiments like FIGS. **5**A-**5**C. The disclosure of aperture length **8042**, aperture width **8044**, aperture offset **8046**, aperture separation distance **8048**, and aperture pairs, and all the associated relationships, applies equally to the slots.

In some embodiments the aperture length **8042** is not equal to the aperture width **8044**. In further embodiments the aperture length **8042** is greater than the aperture width **8044**, and at least 10%, 20%, and 30% greater in further embodiments. Additional embodiments cap this relationship to provide additional fail safes and in such embodiments the aperture length **8042** is no more than 70% greater than the aperture width **8044**, and no more than 60%, 50%, and 40% in further embodiments. In still another embodiment the aperture length **8042** is no more than 80% of the positive locking portion length **8022**, and in further embodiments the aperture length **8042** is no more than 70% of the positive locking portion length **8022**, 60%, and 50%. Similarly, in further embodiments the aperture length **8042** is no more than 40% of the nut cap length **8005**, and in further embodiments the aperture length **8042** is no more than 30% of the nut cap length **8005**, 25%, and 20%. These same relationships of the aperture length **8042** to the positive locking portion length **8022** and the nut cap length **8005** apply equally to relationships of the aperture width **8044** to the positive locking portion length **8022** and the nut cap length **8005**.

The nut cap length **8005** may be at least 25% greater than the nut length **1027**, and at least 35%, 45%, 55%, 65%, and 75% greater in further embodiments. An additional series of embodiments limits this relationship such that the nut cap length **8005** is no more than 200% greater than the nut length **1027**, and no more than 175%, 150%, and 125% greater in further embodiments. In another embodiment the nut engagement portion length **8012** is greater than the positive locking portion length **8022**, and in another embodiment the nut engagement portion length **8012** is at least 5% greater than the positive locking portion length **8022**, and 10%, 15%, and 20% greater in further embodiments.

With reference now to FIG. **21**, while the positive locking portion width **8024** may be equal to, or even greater than, the nut engagement portion width **8014**, in one embodiment, as illustrated, the positive locking portion width **8024** is less than the nut engagement portion width **8014**. In fact, in one embodiment at least a portion of the positive locking portion **8020** has a positive locking portion width **8024** is at least 2.5% less than the nut engagement portion width **8014**, and at least 5%, 7.5%, 10%, and 15% less in additional embodiments. Such embodiments facilitate the use of a compact pin **4090** that when installed and positioned in a locked position, such as that shown in FIG. **25**, the pin **4090** does not extend beyond the widest portion of the locking cap **1040**. Thus, in one embodiment at least a portion of the nut engagement portion **8010** has a nut engagement portion width **8014** that is greater than a greatest parallel dimension between the outermost points on pin **4090**, thereby reducing the likelihood of the pin **4090** becoming a snag nuisance. Further, in the illustrated embodiment of FIGS. **18** and **21** the nut cap bore width **8032** is greater than the positive locking portion width **8024** of at least a portion of the positive locking portion **8020**.

As with all of the disclosed relationships, they involve a delicate balance of tradeoffs that involves more than merely minimizing or maximizing a value, rather embodiments with closed ended ranges recognize points of diminishing returns and avoid negative consequences affecting durability, ease of use, strength, stress and stress distribution, fatigue and vibration resistance, and weight.

FIGS. **8**A-**8**F illustrate the initial positioning of the nut **1020** and nut cap **1040** onto the shaft **4080**. As seen in FIG. **8**A the nut **1020** and nut cap **1040** may be independent and separable, however further embodiments may join the nut **1020** and nut cap **1040** together, while still allowing a predetermined amount of longitudinal movement to facilitate an engaged position and a disengaged position, as previously disclosed with respect to a cap retainer **1050**, which one skilled in the art will recognize may be included in the embodiments of FIGS. **8**A-**26**. Thus, the system may include individual components that are assembled together as seen in FIGS. **8**A-**8**B, or the nut **1020** and nut cap **1040** may be previously joined together, such as the position of FIG. **8**C, via a cap retainer **1050**. Regardless, the following disclosure applies to all embodiments. Thus, with reference again to FIG. **8**A, the nut cap **1040** is aligned with the nut **1020**, and they are longitudinally positioned with respect to on another as shown in FIGS. **8**B-**8**D so they are rotationally engaged and must rotate as a single unit. The assembly is then threaded onto the shaft **4090** as seen in FIGS. **8**E and **8**F. Obviously the external component that is to be fastened is not illustrated.

Once the nut **1020** is positioned on the shaft **4080** as desired, and/or the desired torque is reached, then the positive locking feature of the nut cap **1040** must be aligned with the cross bore **4088**. Thus, FIGS. **8**D-**8**F, **10**B, **10**C, and **20** show the nut **1020** and nut cap **1040** in the engaged position whereby the rotationally locking features are fully engaged and the nut **1020** and nut cap **1040** must rotate in unison. FIG. **9**A shows the movement of the nut cap **1040** in an opposite longitudinal direction away from the fixed nut **1020** to a disengaged position whereby the nut cap **1040** can rotate independent of the nut **1020**. FIG. **9**B shows the nut cap **1040** rotated to an aligned position whereby at least one pair of locking apertures **8040**, or slots **1044**, **5040**, are aligned with the cross bore **4088**. Then, as shown in FIG. **9**C, the nut cap **1040** is moved longitudinally in the opposite direction from the aligned position to the engaged and aligned position, shown in FIG. **9**D, whereby the cross bore **4088** is not only radially aligned with the pair of apertures **8040**, or slots, but also longitudinally aligned with the pair of apertures **8040**, or slots, so that the pin **4090** may pass through one aperture **8040**, or slot, then through the shaft **4080** via the cross bore **4088**, and then through the second aperture **8040**, or slot, thereby locking the nut cap **1040**, and thus the nut **1020**, in the engaged and aligned position relative to the shaft **4080**. The flexibility provided by the different aperture offset **8046**, or slot bottom offset **8047**, or at least two pair of locking apertures **8040**, or slots **1044**, **5040**, and all the associated length, width, radial, longitudinal, and separation distance relationships, increases the likelihood of the installer utilizing a position that maximizes the engagement between the nut cap **1040** and the nut **1020**, and being afforded all the performance benefits of reliability, strength, reusability, and vibration and fatigue resistance, while also greatly simplifying the ease of use and labor savings, which are key goals of the present invention. If not obvious, FIGS. **9**E and **9**F simply illustrate the configuration of FIG. **9**D rotated 90 degrees to illustrate the installation of the pin **4090**. As shown in FIG. **9**F and **25**, in some embodiments one end, or both ends, of the pin **4090** may be bent to secure the pin **4090** in position.

The desired key goals are provided by a delicate interplay of relationships of the various components, variables within each component as well as relationships across the components. The disclosed relationships are more than mere optimization, maximization, or minimization of a single characteristic or variable, and are often contrary to conventional design thinking, yet have been found to achieve a unique balance of the trade-offs associated with competing criteria such as durability, vibration and fatigue resistance, weight, and ease of use. It is important to recognize that all the associated disclosure and relationships apply equally to all embodiments and should not be interpreted as being limited to the particular embodiment being discussed when a relationship is mentioned. Further, the aforementioned balances require trade-offs among the competing characteristics recognizing key points of diminishing returns, as often disclosed with respect to open and closed ranges for particular variables and relationships. Proper functioning of each component, and the overall system, on each and every engagement can be a matter of life or death. Therefore, this disclosure contains a unique combination of components and relationships that produce reliable joining and separation of components, and constrained freedom of movement of the components. While the relationships of the various features and dimensions of a single component play an essential role in achieving the goals, the relationships of features across multiple components are just as critical, if not more critical, to achieving the goals.

Additionally, the relative length, width, thickness, geometry, and material properties of various components, and their relationships to one another and the other design variables disclosed herein, influence the durability, ease of use, security, and safety of the system to achieve the goals. Now to put the disclosed ranges and relationships into perspective with an embodiment of nut cap **1040** has a nut cap length **8005** of 0.375"-0.750", and 0.400"-0.600" in another embodiment, and 0.450"-0.550" in still a further embodiment. The positive locking portion length **8022** is 0.150"-0.500" in an embodiment, and 0.175"-0.400" in another embodiment, and 0.200"-0.300" in still a further embodiment. The positive locking portion width **8024** is

0.250"-0.750" in an embodiment, and 0.300"-0.650" in another embodiment, and 0.325"-0.550" in still a further embodiment. With reference now to FIG. **18**, an average locking portion sidewall thickness **8026** of the positive locking portion **8020** is 0.5-2.5 mm, and 0.7-2.0 mm in another embodiment, and 1.0-1.7 mm in still a further embodiment. The average engagement portion sidewall thickness **8016** of the nut engagement portion **8010** is 0.3-2.0 mm, and 0.4-1.5 mm in another embodiment, and 0.5-1.0 mm in still a further embodiment. Further, the nut length **1027** is 0.150"-0.500" in one embodiment, and 0.200"-0.400" in another embodiment, and 0.250"-0.350" in still a further embodiment. In another embodiment, the threaded portion of the nut **1020** is at least 40% of the nut length **1027**, and at least 50%, 60%, and 70% in further embodiments.

Additionally, the nut slot depth **5041** is 0.050"-0.200" in an embodiment, 0.070"-0.150" in another embodiment, and 0.085"-0.125" in still a further embodiment. In additional embodiments the nut cap bore width **8032**, the maximum tab portion dimension **5032**, and/or the maximum serrated portion dimension **1029** is 0.250"-0.750", and 0.350"-0.650" in another embodiment, and 0.400"-0.550" in still another embodiment. The nut sidewall thickness **8038** is 0.5-2.5 mm, and 0.7-2.0 mm in another embodiment, and 1.0-1.7 mm in still a further embodiment. The tab wall thickness **8036** is 0.3-2.0 mm, and 0.4-1.5 mm in another embodiment, and 0.5-1.0 mm in still a further embodiment. In another embodiment the tab wall thickness **8036** varies such that a maximum tab wall thickness **8036** is at least 25% greater than a minimum tab wall thickness **8036**, as seen in FIG. **15**, and at least 30%, and 35% in further embodiments; while in a further series of embodiments the maximum tab wall thickness **8036** is no more than 150% greater than a minimum tab wall thickness **8036**, and no more than 125%, 100%, and 75% in further embodiments. In one embodiment the nut length **9000** is less than 0.65", and 0.55", 0.45", and 0.35" in further embodiments. In a further embodiment the nut has less than 60 threads per inch.

In addition to the previously disclosed hardness relationships, in another embodiment the helical coil insert **1030** has a density greater than the density of the nut **1020**, the nut cap **1040**, and/or the shaft **4080**. In fact, in another embodiment the helical coil insert **1030** has a density that is at least twice the density of the nut **1020**, the nut cap **1040**, and/or the shaft **4080**.

In a further embodiment at least one of the nut **1020**, nut cap **1040**, shaft **4080**, and pin **4090** are composed of, but not limited to, at least one of the following: an aluminum alloy, an anodized aluminum alloy, a copper containing alloy, a zinc alloy, a stainless steel alloy, a carbon steel alloy, a carbon epoxy compound, or a glass epoxy compound. Additionally, in the embodiments that are composed of various metals a corrosion resisting coating may also be used such as, but not limited to: a cadmium coating, a chromate coating, a polymer coating or a combination thereof. Furthermore, in one embodiment any of the threaded surfaces may have a lubricant to help facilitate ease of installing, including dry film lubricants such as molybdenum disulfide. Further, any of the components may include corrosion resistant coatings and/or cadmium plating.

Some examples of metal alloys that can be used to form any of the components include, without limitation, magnesium alloys, aluminum/aluminum alloys (e.g., 3000 series alloys, 5000 series alloys, 6000 series alloys, such as 6061-T6, and 7000 series alloys, such as 7075, just to name a few), titanium alloys (e.g., 3-2.5, 6-4, SP700, 15-3-3-3, 10-2-3, and other alpha/near alpha, alpha-beta, and beta/near beta titanium alloys, just to name a few), carbon steels (e.g., 1020 and 8620 carbon steel, just to name a few), stainless steels (e.g., A286, 301, 302, 303, 304, 309, 316 and 410 stainless steel), PH (precipitation-hardenable) alloys (e.g., 17-4, C450, and C455 alloys, just to name a few), copper alloys, brass alloys, bronze alloys, nickel alloys, austenitic nickel-chromium-based superalloys such as Inconel, a registered trademark of Special Metals Corporation, high-temperature low creep superalloys such as Nimonic 90, is a registered trademark of Special Metals Corporation, and iron-base superalloys such as heat and corrosion resistant austenitic iron-base material Type A286 alloy (S66286).

Additionally, in some embodiments the nut **1020**, nut cap **1040**, shaft **4080**, and/or pin **4090** may be formed of nonmetallic materials such as plastics, composites, thermoplastics, and resin based composites. In one embodiment the nonmetallic material is a carbon fiber reinforced plastic material. Another embodiment the nonmetallic material is a polyamide resin, while in a further embodiment the polyamide resin includes fiber reinforcement, and in yet another embodiment the polyamide resin includes at least 35% fiber reinforcement. In one such embodiment the fiber reinforcement includes long-glass fibers having a length of at least 10 millimeters pre-molding and produce a finished component having fiber lengths of at least 3 millimeters, while another embodiment includes fiber reinforcement having short-glass fibers with a length of at least 0.5-2.0 millimeters pre-molding. Incorporation of the fiber reinforcement increases the tensile strength of the component, however it may also reduce the primary portion elongation to break therefore a careful balance must be struck to maintain sufficient elongation. Therefore, one embodiment includes 35-55% long fiber reinforcement, while in an even further embodiment has 40-50% long fiber reinforcement. One specific example is a long-glass fiber reinforced polyamide 66 compound with 40% carbon fiber reinforcement, such as the XuanWu W5801 resin having a tensile strength of 245 megapascal and 7% elongation at break. Long fiber reinforced polyamides, and the resulting melt properties, produce a more isotropic material than that of short fiber reinforced polyamides, primarily due to the three-dimensional network formed by the long fibers developed during injection molding. Another advantage of long-fiber material is the almost linear behavior through to fracture resulting in less deformation at higher stresses.

In a still further embodiment the nut **1020**, nut cap **1040**, shaft **4080**, and/or pin **4090** may be formed of a nonmetallic material having a density of less than 2 g/cc and an elongation to break of at least 3% in one embodiment, and at least 4%, 5%, 6%, 7%, and 8% in further embodiments. In a further embodiment the nonmetallic material has a density of less than 1.80 g/cc, and less than 1.60 g/cc, and less than 1.40 g/cc, and less than 1.2 g/cc in additional embodiments. In an embodiment the nonmetallic material is a thermoplastic material, and a Polyetherimide (PEI) in a further embodiment, and, in still more embodiments, any of the following materials that meet the claimed mechanical properties: polycaprolactam, a polyhexamethylene adipinamide, or a copolymer of hexamethylene diamine adipic acid and caprolactam, however other embodiments may include polypropylene (PP), nylon 6 (polyamide 6), polybutylene terephthalates (PBT), thermoplastic polyurethane (TPU), PC/ABS alloy, PPS, PEEK, and semi-crystalline engineering resin systems that meet the claimed mechanical properties. In one embodiment the nonmetallic material has one, or more, of the following properties: a tensile strength of at least 20 Ksi, a tensile modulus of at least 1000 Ksi, a flexural strength of at least 30 Ksi, a flexural modulus of at least 900 Ksi, a compressive strength of at least 20 Ksi, a compressive modulus of at least 450 Ksi, a shear strength of at least 13 Ksi, and a Rockwell M scale hardness of at least 105.

In still another embodiment at least one of the nut **1020** and nut cap **1040** are formed of a metallic material with a density of less than 4.6 g/cc in one embodiment, and less than 3 g/cc in yet another embodiment: and in another embodiment the material has one, or more, of the following properties: an ultimate tensile strength of at least 68 Ksi, and at least 80 Ksi in anther embodiment; a tensile yield strength of at least 47 Ksi, and at least 70 Ksi in another embodiment; an elongation to break of at least 9% in one embodiment, and at least 11% in another embodiment, and at least 13%, 15%, 17%, and 19% in still further embodiments; and/or a modulus of elasticity of at least 9000 Ksi in one embodiment, and at least 10000 Ksi in another embodiment.

Another embodiment tunes the galvanic compatibility of the components. Thus, in one embodiment there is no more than a 0.50 V difference in the "Anodic Index" between any two of the components that come in contact with one another, while in another embodiment there is no more than a 0.25 V difference in the "Anodic Index" between any two of the components that come in contact with one another, and in yet another embodiment there is no more than a 0.15 V difference in the "Anodic Index" between any two of the components that come in contact with one another; per the galvanic data from MIL-STD-889.

Numerous alterations, modifications, and variations of the embodiments disclosed herein will be apparent to those skilled in the art and they are all anticipated and contemplated to be within the spirit and scope of the instant invention. For example, although specific embodiments have been described in detail, those with skill in the art will understand that the preceding embodiments and variations can be modified to incorporate various types of substitute and or additional or alternative materials, relative arrangement of elements, and dimensional configurations. Accordingly, even though only few variations of the present invention are described herein, it is to be understood that the practice of such additional modifications and variations and the equivalents thereof, are within the spirit and scope of the invention as defined in the following claims. Additional benefits and features of the fastener system will be apparent to those skilled in the art.

While the invention has been described with reference to preferred embodiments, those skilled in the art will understand that various changes may be made and equivalents may be substituted for elements thereof without departing from the scope of the invention. In addition, many modifications may be made to adapt a particular situation or material to the teachings of the invention without departing from the essential scope thereof. Since certain changes may be made in the above system without departing from the scope of the invention herein involved, it is intended that all matter contained in the above descriptions and examples or shown in the accompanying drawings shall be interpreted as illustrative and not in a limiting sense. Also, all citations referred herein are expressly incorporated herein by reference. All terms not specifically defined herein are considered to be defined according to Webster's New Twentieth Century Dictionary Unabridged, Second Edition. The disclosures of all of the citations provided are being expressly incorporated herein by reference. The disclosed invention advances the state of the art and its many advantages include those described and claimed.

I claim:

1. A torque control fastener system, comprising:

a fastener nut having a nut proximal end and a nut distal end, a nut bore extending from the nut proximal end into the fastener nut and defining a nut bore axis, a plurality of nut bore threads on at least a portion of the nut bore, and formed of a nut material having a nut Rockwell C hardness of no greater than 42 RWC and a nut coefficient of thermal expansion;

a helical wire insert formed to create a plurality of insert coils forming insert external threads and a plurality of insert internal threads, and formed of an insert material having an insert Rockwell C hardness of at least 43 RWC and an insert coefficient of thermal expansion greater than the nut coefficient of thermal expansion, wherein a portion of the insert external threads mate with a portion of the nut bore threads in a portion of the nut bore and engage the fastener nut;

a shaft having a plurality of shaft threads that mate with a portion of the insert internal threads such that an insertion torque is required to create relative movement between the shaft threads and the insert internal threads in a first direction, and a removal torque is required to create relative movement between the shaft threads and the insert internal threads in a second direction opposite the first direction;

wherein a hardness differential between the insert Rockwell C hardness and the nut Rockwell C hardness is 2-13.

2. The torque control fastener system of claim **1**, wherein the insert coefficient of thermal expansion is at least $1.5\times10^{-6}/°$ C. greater than the nut coefficient of thermal expansion, the nut Rockwell C hardness is no greater than 40 RWC, and the hardness differential is no greater than 10.

3. The torque control fastener system of claim **1**, wherein the insert coefficient of thermal expansion is at least $3.0\times10^{-6}/°$ C. greater than the nut coefficient of thermal expansion, the nut Rockwell C hardness is no greater than 38 RWC, and the hardness differential is no greater than 7.

4. The torque control fastener system of claim **3**, wherein the insert coefficient of thermal expansion is no more than $9.0\times10^{-6}/°$ C. greater than the nut coefficient of thermal expansion.

5. The torque control fastener system of claim **4**, wherein the insert coefficient of thermal expansion is at least $13.0\times10^{-6}/°$ C.

6. The torque control fastener system of claim **5**, wherein majority of the plurality of insert coils are regular coils having a circular end profile, and at least one of the plurality of insert coils is a locking coil having a non-circular end profile including at least three straight segments, and the locking coil is located between regular coils.

7. The torque control fastener system of claim **6**, wherein the helical wire insert has an insert distal end and an insert proximal end, a first quantity of regular coils are located between the insert distal end and the locking coil, a second quantity of regular coils are located between the insert proximal end and the locking coil, and the first quantity is different than the second quantity.

8. The torque control fastener system of claim **7**, wherein the first quantity is at least 2 greater than the second quantity.

9. The torque control fastener system of claim **8**, wherein the first quantity is 4-10 and the second quantity is 2-10.

10. The torque control fastener system of claim **8**, wherein the locking coil is located a locking offset distance from the nut proximal end, and the locking offset distance is at least 60% of the nut length.

11. The torque control fastener system of claim 7, wherein the nut has a lock-free zone and no portion of the locking coil is within the lock-free zone, wherein the lock-free zone is defined by a predetermined lock-free distance measured parallel to a nut longitudinal axis from a transverse plane perpendicular to the nut longitudinal axis and passing through a midpoint of the nut length, and the lock-free zone extends the lock-free distance toward the nut proximal end and extends the lock-free distance toward the nut distal end, and the lock-free distance is at least 5% of the nut length.

12. The torque control fastener system of claim 11, wherein the nut has a second lock-free zone and no portion of the locking coil is within the second lock-free zone, wherein the second lock-free zone is defined by the predetermined lock-free distance measured parallel to a nut longitudinal axis from the nut distal end.

13. The torque control fastener system of claim 12, wherein the nut has a third lock-free zone and no portion of the locking coil is within the third lock-free zone, wherein the third lock-free zone is defined by the predetermined lock-free distance measured parallel to a nut longitudinal axis from the nut proximal end.

14. The torque control fastener system of claim 12, wherein the predetermined lock-free distance is no more than 30% of the nut length.

15. The torque control fastener system of claim 14, wherein the nut includes a tool-engagement portion and a load bearing portion, separated by a transition portion, and no portion of the locking coil is located in the transition portion.

16. The torque control fastener system of claim 15, wherein locking offset distance is 60-90% of the nut length.

17. The torque control fastener system of claim 14, wherein the nut material is aluminum alloy and the nut has a nut mass of less than 1.5 grams, and the insert material is stainless steel alloy and the insert has an insert mass of no more than 85% of the nut mass.

18. A torque control fastener system, comprising:

a fastener nut having a nut proximal end and a nut distal end, a nut bore extending from the nut proximal end into the fastener nut and defining a nut bore axis, a plurality of nut bore threads on at least a portion of the nut bore, and formed of a nut material having a nut Rockwell C hardness of no greater than 40 RWC and a nut coefficient of thermal expansion;

a helical wire insert formed to create a plurality of insert coils forming insert external threads and a plurality of insert internal threads, and formed of an insert material having an insert Rockwell C hardness of at least 43 RWC and an insert coefficient of thermal expansion greater than the nut coefficient of thermal expansion, wherein a portion of the insert external threads mate with a portion of the nut bore threads in a portion of the nut bore and engage the fastener nut, wherein majority of the plurality of insert coils are regular coils having a circular end profile, and at least one of the plurality of insert coils is a locking coil having a non-circular end profile including at least three straight segments, and the locking coil is located between regular coils, and wherein the helical wire insert has an insert distal end and an insert proximal end, a first quantity of regular coils are located between the insert distal end and the locking coil, a second quantity of regular coils are located between the insert proximal end and the locking coil, and the first quantity is different than the second quantity;

a shaft having a plurality of shaft threads that mate with a portion of the insert internal threads such that an insertion torque is required to create relative movement between the shaft threads and the insert internal threads in a first direction, and a removal torque is required to create relative movement between the shaft threads and the insert internal threads in a second direction opposite the first direction;

wherein a hardness differential between the insert Rockwell C hardness and the nut Rockwell C hardness is 2-10, and the insert coefficient of thermal expansion is at least $1.5\times10^{-6}/^\circ$ C. greater than the nut coefficient of thermal expansion.

19. The torque control fastener system of claim 18, wherein the insert coefficient of thermal expansion is at least $3.0\times10^{-6}/^\circ$ C. greater than the nut coefficient of thermal expansion, the nut Rockwell C hardness is no greater than 38 RWC, the hardness differential is no greater than 7, the insert coefficient of thermal expansion is no more than $9.0\times10^{-6}/^\circ$ C. greater than the nut coefficient of thermal expansion, and the insert coefficient of thermal expansion is at least $13.0\times10^{-6}/^\circ$ C.

20. The torque control fastener system of claim 18, wherein the first quantity is 4-10 and the second quantity is 2-10, and the nut has a lock-free zone and no portion of the locking coil is within the lock-free zone, wherein the lock-free zone is defined by a predetermined lock-free distance measured parallel to a nut longitudinal axis from a transverse plane perpendicular to the nut longitudinal axis and passing through a midpoint of the nut length, and the lock-free zone extends the lock-free distance toward the nut proximal end and extends the lock-free distance toward the nut distal end, and the lock-free distance is at least 5% of the nut length.

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