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TECHNICAL ARTICLE—PEER-REVIEWED

Threaded Fastener Locking With Safety Wire and Cotter Pins

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Abstract This paper presents test results and analyses of safety lock wire and cotter pins with castellated nuts used as locking features in threaded fasteners. These mechanical components are used to secure hardware in aerospace and other applications as well as to assess whether a fastener has been tampered with. Tests are performed on aerospace threaded fasteners. Locking moments up to safety lock wire and cotter pin failure are measured. Calculations of locking moment to failure are provided and compared to measured results. For fasteners with safety lock wire, the angle of turn with applied moment is measured and computed. For fasteners with cotter pins and castellated nuts, the angles of turn from clearances and applied moment are computed. Since both safety lock wire and cotter pin with castellated nut applications result in some angle of turn, analysis for loss of preload with angle of turn is developed. Calculations show modest loss in preload for typical angles of turn for long bolts, but a significant loss of preload is found for short bolts.

Keywords Fastener · Bolt · Locking · Secondary · Loosening · Safety wire · Cotter pin · Castellated nut

Introduction and background

The use of threaded fasteners for assembly and attachment in structures, machines and mechanisms remains ubiquitous. Loosening of threaded fasteners has a history as long as threaded fasteners themselves [1]. Careful examination of the mechanics of a threaded fastener assembly reveals an inherent loosening moment even in the absence of external dynamic loads [2].

Efforts to avoid fastener loosening include careful design and installation as well as additional or secondary locking features [1, 2]. These include mechanical locking (e.g., cotter pin or safety wire), prevailing torque locking (e.g., deformed thread or polymer patch), adhesive locking (e.g., anaerobic adhesive) and free spinning locking (e.g., serrated bearing surface or lock washer).

In addition to being added to correct an observed or documented fastener loosening problem, secondary locking is generally used and often required in safety critical applications. For example, many aerospace applications require secondary locking [3]. For fasteners without preload, it provides the only form of locking.

This paper provides data specifically for the mechanical locking provided by safety lock wire and cotter pins with castellated nuts. Even though standards exist for specification [4–10] and installation [11] of these mechanical locking devices, very little quantitative locking data exist. As a result, the data and analyses in this paper help fill an existing void in the literature.

Although safety lock wire and cotter pins with castellated nuts do provide locking as examined in this paper, they have potential issues. These include additional parts and assembly steps, sharp edge hazards [3] and some angular motion due to clearances and deformation. This angular motion can result in loss of preload. This loss of preload depends on the angle of turn from zero to full preload which depends on several fastener and joint parameters. Light [12] shows example data with 30% loss in preload due to rotation to abutment. The angular motion in safety wire and cotter pin application is quantified in this

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Fig. 1 AN bolts with washers, aluminum plate and safety lock wire

paper. In addition, analyses and calculations are provided quantifying the associated loss in preload.

Safety lock wire tests

MS20995 [4] has been superseded by NASM20995 [5] which is the specification standard for safety lock wire. It includes specification for 0.015″ to 0.091″ wire diameter and six materials. Both Inconel and stainless steel safety lock wire have widespread application in aerospace and are therefore used in this work. A wire diameter of 0.032″ is appropriate for the 1/4-28 fasteners tested in this work. The specification [4] for the Inconel 600 (Ni–Cr–Fe alloy) safety lock wire used is MS20995N32 and for the 302/304 stainless steel safety lock wire used is MS20995C32.

All safety lock wire tests in this work use cadmium-plated steel AN4H3A (1/4-28 UNF, 15/32″ nominal length) drilled head bolts [13, 14] with AN960-416 standard and AN960-416L thin washers [15, 16]. A 1/2″ thick aluminum plate with 1/4-28 UNF tapped holes spaced 1.5″ center to center is used as a test fixture. Unassembled sample test components are shown in Fig. 1. The components are assembled without preload using the double twist method specified by NASM33540 [11]. Thin and standard washers are chosen to put bolt head holes approximately perpendicular to the centerline of the tapped holes in the fixture when seated. Sample assembled components are shown in Fig. 2.

The tests are performed by securing the plate and applying a dial-type torque wrench to the top bolt in the loosening (i.e., counter-clockwise) direction. A torque wrench with 0–30 in-lb range followed by a torque wrench with a 0–75 in-lb range is used.

Fig. 2 Safety wired bolts using double twist method: Inconel and stainless steel

Table 1 Locking torque and relative angular position for Inconel safety lock wire

Test#						
19	Torque (in-lb)	10	20	30	35*	35**
	Angle (degrees)	5	10	15	20	
41	Torque (in-lb)	10	20	30	34*	37**
	Angle (degrees)	4	9	17	21	
43	Torque (in-lb)	10	20	30	35*	39**
	Angle (degrees)	3	8	14	19	
45	Torque (in-lb)	10	20	30	33*	37**
	Angle (degrees)	4	7	13	17	
53	Torque (in-lb)	10	20	30	34*	36**
	Angle (degrees)	4	10	14	18	
55	Torque (in-lb)	10	20	30	33*	37**
	Angle (degrees)	5	10	16	20	

*Onset of yield of safety wire
**Failure of safety wire

Six tests were performed for each safety lock wire material. The observed angular rotation of the top bolt head is measured for several applied torque values up to the onset of yield of the safety wire. The applied torque at safety lock wire yield and failure is measured. The onset of safety wire yielding is felt (i.e., when increase in angle occurs with no increase in applied torque), and the corresponding angle recorded. Tables 1 and 2 present the measured torque with angle of top bolt head turn.

Table 2 Locking torque and relative angular position for stainless steel safety lock wire

Test#						
20	Torque (in-lb)	10	20	30	40*	42**
	Angle (degrees)	4	9	14	19	
42	Torque (in-lb)	10	20	30	37*	41**
	Angle (degrees)	3	8	14	20	
44	Torque (in-lb)	10	20	30	36*	40**
	Angle (degrees)	5	9	16	20	
46	Torque (in-lb)	10	20	30	34*	41**
	Angle (degrees)	6	11	15	20	
54	Torque (in-lb)	10	20	30	36*	39**
	Angle (degrees)	5	9	15	19	
56	Torque (in-lb)	10	20	30	35*	38**
	Angle (degrees)	4	9	14	20	

*Onset of yield of safety wire
**Failure of safety wire

Fig. 3 Safety wired bolt assemblies after safety lock wire failure

The test results show that both safety lock wire materials provide 10, 20 and 30 in-lb of locking torque at about 5, 10 and 15 degrees of turn, respectively. In addition, the Inconel safety wire failed at 35–39 in-lb and the stainless steel safety wire failed at 38–42 in-lb. The angle of turn at yield was 17–21 degrees for Inconel and 19–20 degrees for stainless steel. The safety wire failed near the top bolt in some tests and near the bottom bolt in others as shown in Fig. 3.

Safety Lock Wire Analysis

Safety wire is expected to fail primarily in tension. The safety wire used in the tests is 0.032″ diameter. The tensile strength S_t of 0.032″ diameter annealed Inconel 600 wire is from 87,000 to 123,000 psi. The tensile strength S_t of 0.032″ diameter 302/304 annealed stainless steel wire is from 90,000 to 120,000 psi.

The cross-sectional area of 0.032″ diameter safety wire is

$$A_{s-\max} = \pi\left(\frac{0.032}{2}\right)^2 = 0.0008\ \text{in}^2 \qquad \text{(Eq 1)}$$

The tensile force at failure is the product of the total area and the tensile strength. For two wires as used in the double twist method, this force is

$$F_t = 2AS_t \qquad \text{(Eq 2)}$$

The moment required at failure is the product of this force and the moment arm from the bolt head. For an AN4H3A hex head bolt, the head width across the flats is 0.430″ to 0.439″ and the head width across the points is 0.51″. The moment arm w is defined as one-half of the head width and ranges from 0.215″ to 0.255″.

The minimum and maximum moments for tensile failure of the double twist Inconel 600 safety wire are

$$M_{\min} = 2AS_{t-\min}w_{\min} = 2(0.0008)(87{,}000)(0.215) = 29.9\ \text{in-lb} \qquad \text{(Eq 3)}$$

$$M_{\max} = 2AS_{t-\max}w_{\max} = 2(0.0008)(123{,}000)(0.255) = 50.2\ \text{in-lb} \qquad \text{(Eq 4)}$$

The minimum and maximum moments for tensile failure of the double twist 302/304 stainless steel safety wire are

$$M_{\min} = 2AS_{t-\min}w_{\min} = 2(0.0008)(90{,}000)(0.215) = 31.0\ \text{in-lb} \qquad \text{(Eq 5)}$$

$$M_{\max} = 2AS_{t-\max}w_{\max} = 2(0.0008)(120{,}000)(0.255) = 49.0\ \text{in-lb} \qquad \text{(Eq 6)}$$

These calculated ranges for moment to double twist safety wire failure are supported by the tests in which double twist safety wire failed at 35–39 in-lb for Inconel and 40–42 in-lb for stainless steel. This supports the expectation that safety lock wire fails primarily in tension.

The angle of turn of the bolt with the applied loosening moment has been measured and is listed in Tables 1 and 2. This angle is the result of a combination of safety wire stretch under tension, taking up of slack in the loops and

Fig. 4 AN test bolts, castellated nuts and cotter pins

Fig. 5 Castellated nut, cotter pin and bolt assemblies using preferred method

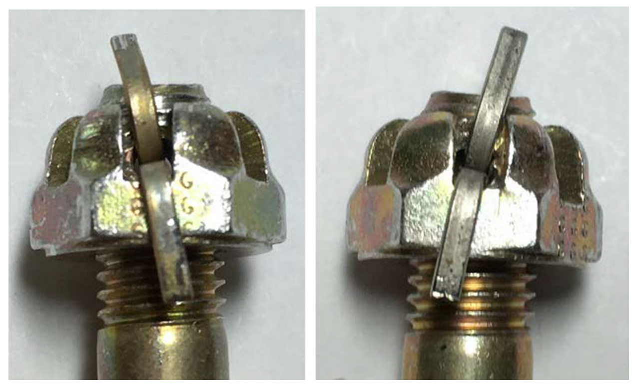

Fig. 6 Clearance between cotter pins and castellated nuts

twist of the safety wire, and turn of the seated lower bolt. Calculations of wire stretch and resulting angle from stretch are only 2–3 degrees. Therefore, most of the 17–21 degrees of bolt turn at yield are the result of taking up of slack in the safety wire and turn of the seated lower bolt.

Cotter pin with castellated nut tests

AN380 [6] and AN381 [7] have been superseded by MS24665 [8] and then NASM24665 [9] which is the specification standard for aerospace cotter pins. It includes specifications for cotter pins from 1/32″ to 3/4″ diameter and 1/4″ to 6″ length for various materials and coatings. Both cadmium-plated steel and stainless steel cotter pins have widespread use in aerospace and are therefore used in this work. A 1/16″ diameter with 1/2″ length cotter pin is appropriate for the 1/4-28 fasteners tested in this work. The specification for the cadmium-plated steel cotter pins used is AN380-2-2 or NASM24665-132 and for the stainless steel cotter pins used is AN381-2-8 or NASM24665-151.

All cotter pin tests in this work use cadmium-plated steel AN4-6 (1/4-28 UNF, 25/32″ nominal length) drilled shank bolts [13, 14] with AN310-4 castellated nuts [10]. Unassembled sample components are shown in Fig. 4. The components are assembled without preload using the preferred method specified by NASM33540 [11]. Sample assembled components are shown in Fig. 5.

Figure 6 shows the circumferential clearance between the cotter pins and castellated nuts. The relative angular play between the cotter pin and castellated nut can be determined from the dimensions and clearance of the test components. The width of the cotter pins, width of the nut slots and diameter of the nuts from flat to flat are measured using calipers. The measured width of the cotter pins is from 0.052″ to 0.053″, the width of the nut slot is from 0.090″ to 0.091″, and the diameter of the nut from flat to flat is 0.432″ to 0.434″.

The castle nut slot width minus cotter pin width defines the clearance. The relative angle of play between the cotter pin and castle nut is defined by the vertex angle of an isosceles triangle with the base being the clearance and each leg being one-half the flat-to-flat diameter of the nut. The minimum and maximum relative angle of play is (from a right triangle within the isosceles)

$$\phi_{\min} = 2\sin^{-1}\left(\frac{0.5(0.090 - 0.053)}{0.5(0.434)}\right) = 9.8 \text{ degrees} \tag{Eq 7}$$

$$\phi_{\max} = 2\sin^{-1}\left(\frac{0.5(0.091 - 0.052)}{0.5(0.432)}\right) = 10.4 \text{ degrees} \tag{Eq 8}$$

The torque measurements are taken by securing the castellated nut and applying a dial-type torque wrench in the loosening (i.e., counter-clockwise) direction. A torque

Table 3 Failure torque for castle nut with cotter pin

Cadmium-plated steel		Stainless steel	
Test#	Failure torque (in-lb)	Test#	Failure torque (in-lb)
17	50	18	60
37	53	38	59
39	51	40	57
47	50	48	61
49	55	50	62
51	52	52	60

wrench with 0–30 in-lb range followed by a torque wrench with a 0–75 in-lb range is used.

Six tests were performed for each cotter pin material. The applied torque is increased until cotter pin failure. The onset of cotter pin yield is detected by relative turn of nut and bolt with no increase in applied torque. The torque at cotter pin yield and failure in shear is about the same. Table 3 presents the measured torque at failure for all tests.

The test results show the cadmium-plated steel cotter pins failed with a torque from 50 to 55 in-lb, and the stainless steel cotter pins failed between 57 and 62 in-lb. Figure 7 shows the components after failure of the cotter pins. As expected, failure occurs in the cotter pins in two shear planes.

Cotter Pin with Castellated Nut Analysis

Cotter pins fail in two shear planes when used with castellated nuts. The two shear planes occur at the pitch diameter of the bolt and castle nut. The pitch diameter d_p is 0.2243–0.2268″ for an AN4 (1/4-28 UNF-3A) thread.

The hardness requirement for the stainless steel NASM24665 cotter pins of 1/16″ nominal diameter is 65–78 Rockwell 15-N or 200–350 Vickers 10 kg [9]. These hardness ranges correspond to approximate tensile strength from 92,000 to 163,000 psi and shear stress S_s from 55,000 to 98,000 psi. The material requirement [9] for the cadmium-plated steel cotter pins is simply carbon steel which is too vague to approximate shear strength.

The area of shear in each shear plane is determined from the cotter pin dimensions, specifically width B and height A shown in Fig. 8. NASM24665 [9] specifies the dimensions of NASM24665-132 and NASM24665-151 cotter pins. The width B has a specified min. of 0.044″ and max. of 0.060″. The height A has a specified min. of 0.056″ and max. of 0.060″. The test cotter pins dimensions were within these limits. The minimum area of shear in each shear plane is

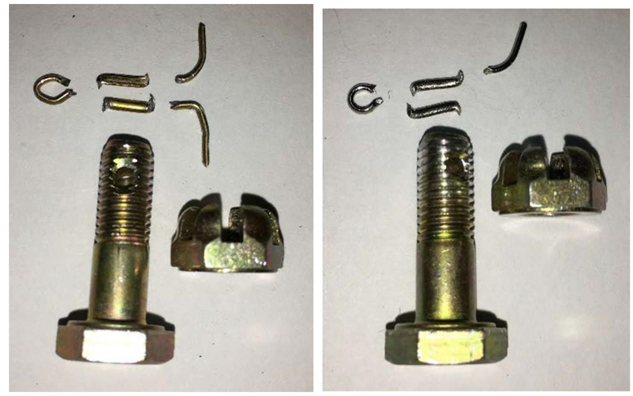

Fig. 7 Cotter pin assembly components after cotter pin failures

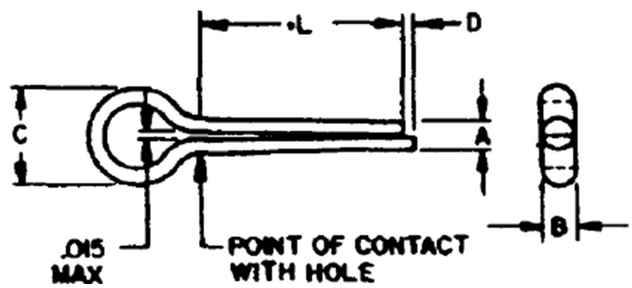

Fig. 8 Cotter pin dimensions (from [9])

$$A_{s-min} = \pi\left(\frac{0.044}{2}\right)\left(\frac{0.056}{2}\right) = 0.0019\ \text{in}^2 \tag{Eq 9}$$

The maximum area of shear in each shear plane is

$$A_{s-max} = \pi\left(\frac{0.06}{2}\right)^2 = 0.0028\ \text{in}^2 \tag{Eq 10}$$

The shear force at failure is the product of the total area of shear and the shear strength

$$F_s = (A_s + A_s)S_s \tag{Eq 11}$$

The moment required to shear the cotter pin in the castle nut is the sum of the moments at each shear plane

$$M = A_sS_s\left(\frac{d_p}{2}\right) + A_sS_s\left(\frac{d_p}{2}\right) = A_sS_sd_p \tag{Eq 12}$$

Here the moment arm for the force at each shear plane with respect to the axis of the bolt and nut is $d_p/2$. The minimum and maximum moments to shear the stainless steel cotter pin are

$$M_{min} = A_{s-min}S_{s-min}d_{p-min} = (0.0019)(55{,}000)(0.2243) = 23.4\ \text{in-lb} \tag{Eq 13}$$

$$M_{max} = A_{s-max}S_{s-max}d_{p-max} = (0.0028)(98{,}000)(0.2268) = 62.2\ \text{in-lb} \tag{Eq 14}$$

This calculated range for moment of cotter pin in castellated nut failure is supported by the tests in which the stainless steel cotter pins failed in shear between 57 and 62 in-lb as listed in Table 3.

The cadmium-plated cotter pins failed in shear with applied torques between 50 and 55 in-lb as listed in Table 3. This is about 10–20% lower than that measured with the stainless steel cotter pins, indicating the shear strength of the tested cadmium-plated steel cotter pins has a shear strength about 10–20% lower than the tested stainless steel cotter pins.

The relative angular play between the cotter pin and castle nut can be determined from the specification dimensions and clearance of the components. As stated above, the test cotter pins have a width from 0.044″ to 0.060″. The AN310-4 castle nut slots have a width of 5/64″ plus 1/32″ or minus 0″ (0.0781″ to 0.1094″) and a flat-to-flat diameter of 0.428″ to 0.440″. The castle nut slot width minus cotter pin width defines the clearance. The relative angle of play between the cotter pin and castle nut is defined by the vertex angle of an isosceles triangle with the base being the clearance and each leg being one-half the flat-to-flat diameter of the nut. The minimum and maximum relative angle of play is (from a right triangle within the isosceles)

$$\phi_{\min} = 2\sin^{-1}\left(\frac{0.5(0.0781 - 0.060)}{0.5(0.440)}\right) = 4.7 \text{ degrees} \quad \text{(Eq 15)}$$

$$\phi_{\max} = 2\sin^{-1}\left(\frac{0.5(0.1094 - 0.044)}{0.5(0.428)}\right) = 17.6 \text{ degrees} \quad \text{(Eq 16)}$$

As an example, a cotter pin set dead center in a castle nut slot would have 2.4–8.8 degrees angle of play in the loosening direction.

Self-loosening Moment

Threaded fasteners exhibit an inherent self-loosening moment that has been previously developed and presented [2]. This self-loosening moment is defined by

$$M_{\text{self}-l} = \frac{F_{\text{p}}p}{2\pi} \quad \text{(Eq 17)}$$

Here F_{p} is the preload and p is the thread pitch. This results from the bolt stretch torque and associated potential energy in the bolt. It is inherent to the threaded fastener and is proportional to preload and thread pitch.

Loosening Moment from External Loads

In addition to the inherent self-loosening moment in fasteners due to bolt stretch, external loads introduce loosening moments, $M_{\text{ext-}l}$. The loosening moment from external cyclic transverse loads on a bolted joint has been developed and presented previously [2]. Similar loosening moments can result from other external loads such as axial loads, bending loads and combined loads. The total loosening moment acting on a threaded fastener is determined by adding the loosening moments created by external loads with the self-loosening moment.

Primary Locking

The friction terms in the tightening and removal torque equations define the primary locking moment in a bolted joint [2]. This primary locking moment is dependent on preload and friction. This primary locking is absent in fasteners without preload. Even with preload, if sustained cyclic slip occurs in a bolted joint due to an external load, this primary locking moment can be ineffective and locking must be provided by a secondary locking feature.

Secondary Locking Requirement

The total loosening moment in a threaded fastener is the self-loosening moment plus any external load loosening moment. In the absence of thread and nut friction, this total loosening moment defines the locking moment required from a secondary locking feature. The secondary locking feature locking moment requirement becomes

$$M_{\text{locking}} \geq M_{\text{self}-l} + M_{\text{ext}-l} \quad \text{(Eq 18)}$$

As an example, consider a 0.25–28 UNF thread fastener in a joint with a 2190-lb preload and an external cyclic transverse load creating a loosening moment of 10 in-lb. The self-loosening moment is

$$M_{\text{self}-l} = \frac{F_{\text{p}}p}{2\pi} = \frac{2,190}{2\pi(28)} = 12.5 \text{ in-lb} \quad \text{(Eq 19)}$$

The external load loosening moment of 10 in-lb combines with this self-loosening moment for a total loosening moment of 22.5 in-lb. If a locking feature is used with a locking torque of 25 in-lb, then a small margin against loosening is provided by the locking feature. Note that in the absence of the external load loosening moment, the total loosening moment is 12.5 in-lb. In this case, a locking feature with a locking torque of 15 in-lb is sufficient and provides a small margin against loosening.

Loss of Preload with Angle

During assembly of a bolted joint, the relative angle of turn between the threaded components results in bolt stretch. With rigid clamped components, one complete turn of three hundred and sixty degrees results in one pitch of stretch. Bolt stretch times the effective bolt stiffness equals the preload. The longer the bolt for a given diameter, the lower the stiffness and higher the stretch.

Since safety lock wire and cotter pins with castellated nuts can have some angle of turn due to clearances and deformation, it is useful to calculate the loss of preload due to this angle.

Bolt stretch is generally determined from Hooke's law and treating the unthreaded body of the bolt with a portion of the bolt head thickness in series with the threaded portion of the bolt up to the nut with a portion of the nut thickness [17] as

$$\Delta L = \frac{F_p}{E}\left(\frac{0.5t_h + L_s}{A} + \frac{L_t + 0.5t_n}{A_m}\right) \quad \text{(Eq 20)}$$

Here F_p is the preload, E is the modulus of elasticity, t_h is the thickness of the bolt head, L_s is the length of the bolt shank, A is the cross-sectional area of the bolt shank, L_t is length of threaded section up to the nut, t_n is the thickness of the nut, and A_m is the cross-sectional area of the bolt at the minor diameter. The angle of turn for N percent loss of preload is

$$\theta = \frac{3.6N\Delta L}{p} \quad \text{(Eq 21)}$$

And the percent loss of preload in terms of angle of turn is

$$N = \frac{\theta p}{3.6\Delta L} \quad \text{(Eq 22)}$$

As an example, consider an AN4-30 bolt with an AN310-4 nut. Since this bolt yields at 3130 lb [13, 14], a preload F_p at 70% yield is 2190 lb. In addition, $E = 30 \times 10^6$ psi, $t_h = 0.156''$, $L_s = 2.5625''$, $A = 0.0487$ in^2, $L_t = 0.1875''$, $t_n = 0.2813''$, $A_m = 0.0335$ in^2 and $p = 1/28$ [10, 13]. Using Eq 20, the bolt stretch at preload is 0.0045″. Using Eq 21, an angle of turn of 45 degrees results in 100% loss of preload. However, from Eq 22, 22% loss of preload occurs with 10 degrees of turn.

Table 4 presents results for this example for various AN4 bolt lengths. All of the parameters remain the same except for the bolt shank length L_s. Percent preload loss calculations are provided for 5, 10, 15 and 20 degrees of angle of turn. Also, the angle of turn for 100% preload loss is included for all bolt lengths.

Table 4 Percent loss of preload with angle for different length AN4 bolts

Bolt	Shank length L_s (in)	Bolt stretch at preload (in)	Percent preload loss with angle (deg) of				Angle for 100% preload loss
			5	10	15	20	
AN4-6	0.3125	0.0012	41	83	100	100	13
AN4-20	1.5625	0.0030	17	33	50	66	31
AN4-30	2.5625	0.0045	11	22	33	44	45
AN4-40	3.5625	0.0059	8	17	25	34	60
AN4-50	49/16	0.0074	7	13	20	27	74
AN4-60	59/16	0.0088	6	11	17	23	89
AN4-70	69/16	0.0102	5	9	15	19	103
AN4-80	79/16	0.0117	4	8	13	17	118

The results in Eqs 7 and 8 show about 10 degrees of turn due to clearance with the test AN310-4 castellated nuts and AN380-2-2 or AN381-2-8 cotter pins. Table 4 shows for the shortest bolt examined this results in 83% loss in percent. This reduces to 8% preload loss for the longest bolt examined. This suggests for short bolts that cotter pins with castellated nuts should not be used for maintaining preload.

Results similar to those in Table 4 are found for AN4HA bolts (as used with safety lock wire) of various lengths in tapped holes or inserts. Measurements in Tables 1 and 2 show some locking moment (comparable with prevailing torque lock nuts [2]) with safety lock wire at an angle of turn at about 5 degrees. This locking moment increases with angle of turn. Table 4 shows for the shortest bolt examined, even 5 degrees of turn results in 41% preload loss. This reduces to as little as 4% preload loss with the longest bolt examined.

Conclusions

Test results and analyses from 1/4-28 UNF threaded fasteners with 0.032″ diameter Inconel and stainless steel safety lock wire using the double twist method were presented. Locking moments with angle of turn were measured. Locking moments comparable to that obtained with aerospace prevailing torque lock nuts were found for safety lock wire at an angle of turn of about 5 degrees. Locking moments were found to increase to 33–40 in-lb with angle of turn until safety lock wire yield at about 20 degrees. The stainless steel safety wire was found to provide slightly higher locking torque than the Inconel safety wire.

Calculations of bolt torque at safety lock wire tensile failure were found to agree with the locking torque measurements at failure. Calculations of wire stretch and resulting angle were found to be only 2–3 degrees, suggesting most of the measured bolt turn at yield was the result of taking up of slack in the safety wire and turn of the seated lower bolt.

Test results and analyses from 1/4-28 UNF threaded fasteners with 1/16″ cadmium-plated steel and stainless steel cotter pins with castellated nuts were presented. Angle of play between the test cotter pins and castellated nuts was determined to be about 10 degrees. The locking moments at cotter pin failure were measured from 50 to 55 in-lb for cadmium-plated steel and from 57 to 62 in-lb for stainless steel. The stainless steel cotter pins were found to provide 10–20% higher locking torque than the cadmium-plated cotter pins.

Calculations of nut torque at shear failure of the stainless steel cotter pins were found to agree with the measured values. Calculations of the angle of play between the cotter pins and castellated nuts for specification dimensions ranged from 5 to 18 degrees which support the results from the test fasteners.

Since both safety lock wire and cotter pin with castellated nut applications result in some angle of turn due to slack in lock wire assembly or clearance between pin and nut slot, analyses for loss of preload with angle of turn were developed for preload bolts. Sample calculations showed modest loss in preload with 5–10 degrees of angle of turn for long bolts and rigid clamped components. However, a significant loss of preload with angle of turn was found for short bolts, indicating that short bolts with safety lock wire or cotter pins and castellated nuts should not be used for maintaining preload.

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