THE HOBSON Update

Volume 11

From the desk of Peter Hobson

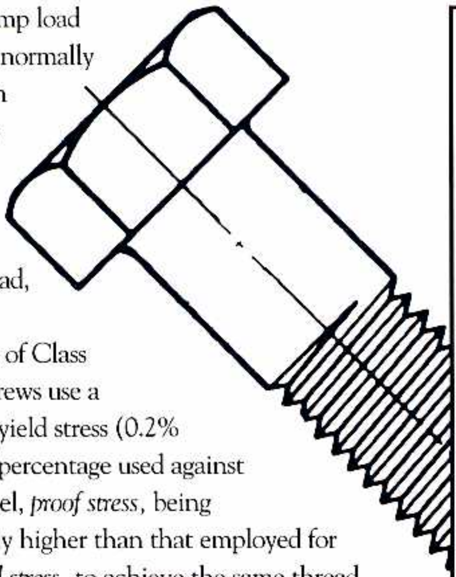
Inside...

Grade 8 Hexagon Bolts and Nuts

THESE products have been around for very many years, but their use, until relatively recently, has been somewhat restricted. Now this is changing as recognition of their advantages over the lower strength Grade 5 product broadens. This change is likely to accelerate in very much the same way and for the same logical reasons, as has already been experienced — over a sufficiently enough long period for a clear trend to be demonstrated — with Class 10.9 bolts in the metric series.

How is one to decide on whether to upgrade? The bottom line would seem to be to do a cost/benefit study. That is, to balance fastener cost against overall joint integrity. Joints are after all only as good as the bolts that hold them together and, the bolts on the other hand, are only as good as the clamp load they can deliver. As has been said in earlier issues of The Hobson Update, as well as previously in at least one other reputable locally produced and widely circulated publication, a joint loses its integrity as soon as its contact surfaces cease to remain very firmly in contact with one another. Obviously this would happen if there was any "give" in the bolts under loading, or, to express it in the more usual form, if the bolts lack adequate clamp load capacity.

A bolt's clamp load capacity, one way or another, can be said to be a function of its yield strength . The higher the yield strength the higher the clamp load capacity. One normally tightens a high tensile bolt up to a percentage of the stress under proof load, but some manufacturers of Class 12.9 socket screws use a percentage of yield stress (0.2% offset) — the percentage used against the former level, *proof stress*, being proportionately higher than that employed for the latter, *yield stress*, to achieve the same thread preload stress level. (It is a different procedure with structural bolts which typically are, because of the nature of their application, theoretically tightened into the yield zone.) The advantage of Grade 8 bolts over the lower strength Grade 5s, working within the diameter range per AS 2465, can be stated as:



Bolt Grade	Tensile Strength		Yield Strength		Index	
	≥¼ ≤1	≥1 ≤1½	≥¼ ≤1	≥1 ≤1½	Gr 5	Gr 8
Grade 5	120 000		92 000		100	141
		105 000		81 000	100	160
Grade 8	150 000	150 000	130 000	130 000		

The table shows Grade 8 bolts as having 41% more clamp load capacity than Grade 5s in the range 1/4 through to 1 inch and 60% more on sizes above 1 inch diameter. This means, when substituting Grade 8 bolts for Grade 5s, only 71% of the number of bolts would, in theory, be required for the same clamp load in

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the size range up to 1 inch and then only 62.5% of the number of bolts in sizes over 1 inch diameter.

So, if you are not looking beyond the first cost of the bolts, it could well pay to design around Grade 8. The same logic of course applies also when comparing Class 10.9 and Class 8.8 bolts. Notwithstanding this, as the cost of the fastener is generally only a very small component in the total cost of a joint (in any case, the costs today of Grade 8 and Class 10.9 bolts are competitive respectively against the lower strength Grade 5 and Class 8.8 options), argument most usually favours going for the extra clamp load rather than using fewer bolts. So, in proceeding thus, our cost/benefit study would reveal a gain of 41% or 60% (according to size) in joint integrity, when up-grading from Grade 5 to Grade 8 hexagon bolts. And at little or no cost disadvantage.

It is also interesting to look at fastener cost against installed fastener cost. That is, examine what all the other associated costs are, namely, those which go towards preparation of the joint prior to the actual bolting. How then does the cost of the bolt compare with the total of these? For a start, at least two holes have to be drilled (one on each side of the joint) for every bolt

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Some useful Grade 8 Bolt Details

Thread Size	Tensile Strength		Yield Strength (0.2% offset)		Shear Strength† (lbf)		Tightening Torque	
	ksi	lbf	ksi	lbf	Body	Thread	lbf.ft	Nm
1/4 – 20 UNC	150	4770	130	4130	4420	2860	11.7	15.9
5/16 – 18 UNC	150	7860	130	6810	6900	4720	24.2	32.8
3/8 – 16 UNC	150	11630	130	10080	9940	6980	42.9	58.1
7/16 – 14 UNC	150	15950	130	13820	13530	9570	68.6	93.1
1/2 – 13 UNC	150	21290	130	18450	17670	12770	105	142
9/16 – 12 UNC	150	27300	130	23660	22370	16380	150	205
5/8 – 11 UNC	150	33900	130	29380	27610	20340	210	280
3/4 – 10 UNC	150	50100	130	43420	39760	30060	370	500
7/8 – 9 UNC	150	69300	130	60060	54120	41580	600	810
1 – 8 UNC	150	90900	130	78780	70690	54540	890	1210
1-1/8 – 7 UNC	150	114450	130	99190	89460	68670	1270	1720
1-1/4 – 7 UNC	150	145350	130	125970	110450	87210	1790	2420
1-3/8 – 6 UNC	150	173250	130	150150	133640	103950	2340	3180
1-1/2 – 6 UNC	150	210750	130	182650	159040	126450	3110	4220
1-3/4 – 5 UNC	150	285000	130	247000	216480	171000	4910	6650
2 – 4-1/2 UNC	150	375000	130	325000	282740	225000	7380	10000
1/4 – 28 UNF	150	5460	130	4730	4420	3280	13.4	18.2
5/16 – 24 UNF	150	8700	130	7540	6900	5220	26.7	36.3
3/8 – 24 UNF	150	13170	130	11410	9940	7900	48.6	65.9
7/16 – 20 UNF	150	17810	130	15430	13530	10680	76.6	105
1/2 – 20 UNF	150	23990	130	20790	17670	14390	118	160
9/16 – 18 UNF	150	30450	130	26390	22370	18270	170	230
5/8 – 18 UNF	150	38400	130	33280	27610	23040	240	320
3/4 – 16 UNF	150	55950	130	48490	39760	33570	410	560
7/8 – 14 UNF	150	76350	130	66170	54120	45810	660	890
1 – 12 UNF	150	99450	130	86190	70690	59670	980	1330
1-1/8 – 12 UNF	150	128400	130	111280	89460	77040	1420	1930
1-1/4 – 12 UNF	150	160950	130	139490	110450	96570	1980	2680
1-3/8 – 12 UNF	150	197250	130	170950	133640	118350	2670	3620
1-1/2 – 12 UNF	150	273150	130	205530	159040	142290	3500	4740

† Single shear

Some useful Class 10.9 Bolt Details

Thread Size	Thread Pitch	Tensile Strength		Yield Strength (0.2% offset)		Shear Strength† (kN)		Recommended Tightening Torque	
	mm	MPa	kN	MPa	kN	Body	Thread	Nm	lbf.ft
M6	1	1040	20.9	940	18.9	17.7	12.6	14.7	10.8
M8	1.25	1040	38.1	940	34.4	31.4	22.8	35.8	26.4
M10	1.5	104	60.3	940	54.5	49.0	36.2	70.8	52.2
M12	1.75	1040	87.6	940	79.2	70.5	52.6	124	91.1
(M14)	2	1040	120	940	109	96.1	72.0	200	146
M16	2	1040	163	940	147	125	97.8	305	225
(M18)	2.5	1040	200	940	181	155	120	425	310
M20	2.5	1040	255	940	230	195	155	600	440
(M22)	2.5	1040	316	940	285	240	190	815	600
M24	3	1040	367	940	331	280	220	1030	760
(M27)	3	1040	478	940	432	360	290	1520	1120
M30	3.5	1040	583	940	527	440	350	2050	1510
(M33)	3.5	1040	721	940	652	530	430	2800	2060
M36	4	1040	849	940	768	640	510	3590	2650
(M39)	4	1040	1015	940	917	750	610	4650	3430
M42	4.5	1040	1166	940	1054	860	700	5750	4240
M48	5	1040	1532	940	1385	1130	920	8640	6370

† Single shear

required. Drilled holes don't come cheaply, and it doesn't end there as the edges of these holes most often need to be relieved by very light countersinking to follow good engineering practice. In some cases, the bolt has to be screwed into a hole tapped into one half of the joint. In other words, used as a hex head cap screw (or hex head set screw, if you prefer). Well, hole chamfering and tapped holes don't come cheaply either. After all that there is the assembly cost. There is no escaping this as assembly costs are very much the same whatever the strength grade of the bolt selected.

The rising interest in the use of Grade 8 (and Class 10.9) bolts in the developed world could well be explained as being the consequence of a growing number of users getting down to the simple and sensible task of cost/benefit analysing their joints.

Tensile, yield and shear strengths are shown in the preceding tables size by size, together with corresponding suggested tightening torques.

Our general recommendation on tightening torques, which broadly align with US published guidelines, provides for these to be based upon a preload stress in the threads approximating 75% of proof load stress. Proof load stresses for Grade 8 and Class 10.9 series bolts, being nominally around 90% of the tabulated yield stresses, are 120,000 lbf/in^2 (equivalent to 830 MPa) and 830 MPa, respectively. These strengths relate approximately to a hardness of R_c 34 (or HRC 34). Of course the actual induced stress (from tightening), taken as a percentage of a particular bolt's actual yield strength, will vary according to the hardness of that bolt. As both the relevant standards (AS 2465 and AS 1110), allow for variation in hardness upwards to R_c 39, it follows that the percentage of the induced stress (ie. from tightening) to the actual yield strength of any off-the-shelf, as-manufactured bolt, could vary downwards from a theoretical 65% of yield (equates nearly with 75% of proof load stress at specified minimum levels) to some lower level, as the actual yield strength of the bolt increases within the allowable hardness range. This is exampled in the following table.

Yield strength is used in the table and not stress under proof load, as it maintains its 90% relationship to tensile strength within the allowable hardness range. Stress under proof load is a fixed minimum stress to be met for the standard as a whole, irrespective of where

"Drilled holes don't come cheaply."

Grade/Class	Seating Stress At 65% Of Min. Yield	Theo. % Of Actual Yield Strength With Bolt Hardness Change	Approx Tensile Strength	Approx Yield Strength	Bolt Hardness Rockwell C Scale
Grade 8	84.5 ksi	65%	150 ksi	130 ksi	R_c 34
Class 10.9	610 MPa	65%	1040 MPa	940 MPa	R_c 34
Grade 8	84.5 ksi	60%	160 ksi	140 ksi	R_c 36
Class 10.9	610 MPa	62%	1100 MPa	990 MPa	R_c 36
Grade 8	84.5 ksi	56%	170 ksi	150 ksi	R_c 38
Class 10.9	610 MPa	58%	1170 MPa	1060 MPa	R_c 38

individual parts fall within the allowable hardness range. The table highlights the conservativeness of tightening to 65% of (min.) yield strength preload (equates with the 75% of proof load stress which we recommend for Grade 8 and Class 10.9 bolts). International leaders in socket screw manufacture, to quote from their technical publications, base their Class 12.9 socket head cap screw seating torque recommendations on a stress in the threads of a little over 70% of minimum yield stress — aligns with 73% of the minimum (0.2% offset) yield stress for Class 12.9 hex bolts to AS 1110. So why not 70% of yield for Class 10.9 and Grade 8 bolts? This is a matter for the designer to decide but we have no real concerns on that point. In the meantime, however, 65% of yield which, once again, corresponds approximately to 75% of proof load stress, is by derivation, our recommendation for general application. There are many variables to allow for in joints, for example, the mating surface material hardnesses and degrees of smoothness/roughness, which have to be allowed for when tightening by the torque method, where high levels of clamp load (bolt preload) are necessary to the avoidance of fatigue failure. Consequently, recommendations for the general case, while not being so positioned as to waste vital clamp load capacity, do have to make allowance for the many quite usual variables. We believe that about the foregoing preload stress levels in the threads for Grade 8 and Class 10.9 bolts is positioned to do this.

> *"So, why not 70% of yield?"*

Detail	Index
Tensile Stress	= 100
Yield Stress	= 90
Proof Load Stress	= 80

If Class 10.9 & Grade 8 Tensile Stress is indexed at 100 then the Yield Stress Index would be 90 and the Proof Stress Index would be 80.

Notwithstanding all of this, there should not normally be a problem with an assembly (seating) stress in the threads at 70% of yield stress (0.2% offset) should designers require this when tightening by torque, since in the worst-case, at the upper level per the IFI published torque wrench tightening range (± 25%), preload would still only be lifted to 87.5% of yield stress, namely, to just below minimum proof stress (it being nominally 90% of yield), and therefore the preloaded bolt would still be within the elastic elongation range. Conversely, it has to be recognised, when tightening to 70% of yield and assuming a worst-case torque wrench accuracy right at the lower end of the ± 25% accuracy range, preload would then only be 52.5% of yield, but an even lower 49 % of yield if tightened to 65% of yield. For readers ease of reference, we repeat the preload measuring method table from page 2 of Hobson Update volume 7.

Preload Measuring Method	Accuracy %	Relative Cost
Feel (Operator Judgement)	± 35	1
Torque Wrench	± 25	1½
Turn-Of-Nut	± 15	3
Load Indicating Washers	± 10	7
Fastener Elongation	± 3 to ± 5	15
Strain Gauge	± 1	20

Table from Industrial Fastener Institute (USA) handbook.

This table covers a range of methods of measuring fastener preload and the differing degrees of accuracy attaching to each. It is particularly with regard to the TORQUE WRENCH method that the forgoing discussion has referred. Another of the alternatives covered in the table is the TURN-OF-NUT method. While this is widely used in the structural industry (HIGH STRENGTH STEEL BOLTS AND NUTS TO AS 1252), there are

"The trouble with our age is that it is all signpost and no destination." Louis Kronenberger, 1954 surprisingly not all that many calls for information from industry about using this method of tightening more generally. The approach here is to turn the nut through a number of degrees, after snug, sufficient for the bolt's elastic elongation to induce the desired internal stress in the thread. In the broad general case, with this method, it is usual for the threaded portion of the bolt not to be preloaded above or even up to it's minimum yield stress. On the other hand, the method's increased accuracy shown in the table, compared to tightening by torque, removes quite a lot of uncertainty.

It is highly recommended that, in the case of there being a need for a high preload stress in the thread, because of some critical application and where the torque wrench method of tightening is to be used, that a more exact measure of tightening torque be determined beforehand. This can be achieved by first torque tension testing the bolts on joint materials which are fully representative of what will be used and apply in the actual application.

But how many degrees of turn should be applied if the **TURN-OF-NUT** method is used? If the application is one of a typical or routine engineering nature (ie, not a structural bolt one or an out of the ordinary application), the following US published table could be followed.

Degress of Nut Rotation from "SNUG" Condition

Joint Thickness	SAE Grade 5†	SAE Grade 8†
Up to and including four bolt diameters	60°	90°
Over four bolt diameters and not exceeding eight bolt diameters	90°	120°
Over eight bolt diameters and not exceeding twelve bolt diameters	105°	135°

† UNC threads

As indicated, the data in the table is in fact for course thread (UNC) bolts. A 90° turn after snug of a 1-8 UNC nut would elongate the bolt by 25% of the thread pitch , or 0.25 x 0.125" = 0.0313 inches, whereas the same degree of turn of a 1-12 UNF nut would only elongate the bolt by (0.25 x 0.083") 0.021 inches. That is, an additional 45° of turn (or 135° in total) would be required for the UNF nut, in this example, to develop the same bolt extension (0.0313") and therefore (lbf or kN) force in the bolt.

We think that it is reasonable to apply the same degrees of turn to (course thread series) Class 10.9 bolts, as those shown in the table for Grade 8 ones. The two are at the same strength level, basically, and there is little difference in their thread pitches as limited sample in the following table shows.

Size	Pitch mm	Size	Pitch mm
M10	1.5	⅜-16	1.6
M16	2.0	⅝-11	2.3
M20	2.5	¾-10	2.5
M24	3.0	1-8	3.2
M30	3.5	1¼-7	3.6

On the other hand, with structural bolting, where the minimum allowable stress in the bolt is the stress under proof load, it is common practice for the degree of turn to be sufficient to aim at taking the bolt thread section part of the way into the plastic region. This provides for higher clamp loads and, as the bolt thread stress has been moved into the relatively flat portion of the preload/tightening angle curve, reduced preload variation (preload scatter) from bolt to bolt, as well. Although the requirement here is for bolt tension to at least be equal to the minimum proof load specified under AS 4100 – 1990, it will not always be that the threaded section of the bolt is actually stressed above the yield point. This would happen only theoretically, albeit maybe more often than not.

We say theoretically because the structural bolt standard (AS1252) permits hardness to vary from R_c 23 to R_c 34. It is therefore quite possible that the threaded section of the bolt will still not be tightened above it's *actual* yield stress. The minimum yield stress (more properly 0.2% permanent set, going by the standard) for structural bolts is 640 MPa (equates with R_c 24), whereas the yield stress at a hardness level of say R_c 32 would be very approximately 785 MPa. This is to say that if a given angle of nut turn after snug, as determined by testing, would take the threaded section of the bolt some 10% beyond the minimum specified yield strength, this would mean loading the thread roughly to 705 MPa. Reference to hardness/tensile strength tables shows, that under this stress in the thread,

"How many degrees of turn should be applied?"

only those bolts with hardnesses below R_c 28 would have their threads with tightening induced preloads in fact exceeding their actual yield stress. This is demonstrated in the following table.

Structural Bolts to AS 1252

Minimum Yield Strength	Actual Yield Strength	Minimum Tensile Strength	Actual Tensile Strength	Bolt Hardness Rockwell C Scale
640 MPa	640 MPa	800 MPa	800 MPa	R_c 24
640 MPa	750 MPa	800 MPa	940 MPa	R_c 30
640 MPa	800 MPa	800 MPa	1010 MPa	R_c 33

"It will not be the whole bolt that will be stressed above yield."

Of course, it will not be the whole bolt that will be stressed above yield. The full body section will not be because of its larger cross-sectional area. Taking as an example an M20 x 100 structural bolt. The body proportions are nominally: bolt length — 100 mm; thread length — 46 mm, body length — 54 mm; thread stress area — 245 mm^2; and body area — 314 mm^2. The load needed to raise the stress in the thread of the bolt up to minimum yield (stress/1000 x area) would be (640/1000 x 245) or say 157 kN. A load of 173 kN(10% more), therefore, will elevate the stress to 10% above the specified minimum, viz. say to 705 MPa. At this level, the stress in the full body of the bolt will only be 705 MPa x 245 mm^2 ÷ 314 mm^2, or 550 MPa. In fact, it would take a load of 200 kN to lift the stress in the *full body* to the 640 MPa and this load would raise the stress in the thread to 816 MPa. This would then be above the 800 MPa minimum tensile strength for structural bolts. Consequently, whether or not a bolt is actually tightened beyond its (thread section) yield stress, will depend not only on the degree of over-torque, but on the bolt's hardness as well.

The angles of turn and minimum bolt tensions for structural bolts specified under AS4100 – 1990 are detailed in the two following tables.

Minimum Bolt Tension – Structural Bolts AS 4100 – 1990

Nominal diameter of bolt	Minimum bolt tension, kN
M16	95
M20	145
M24	210
M30	335
M36	490

NOTE: The minimum bolt tensions given in this table are approximately equivalent to the minimum proof loads given in AS 1252.

The contrast between preload scatter favouring tightening into the plastic area is well demonstrated in the following diagram. Although somewhat exaggerated for ease of following (the straight line portion of the 'curve' actually runs more toward the vertical than has been drawn), this shows the marked extent to which scatter diminishes for the same degree of

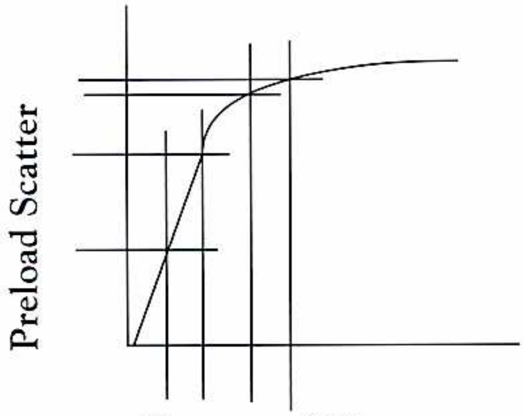
Degrees of Nut Rotation from "SNUG" Condition — Structural Bolts AS 4100 – 1990

Bolt Length (underside of head to end of bolt)	**Disposition of Outer Face of Bolted Parts** (See notes 1,2,4 & 4)		
	Both faces normal to bolt axis	One face normal to bolt axis and other sloped	Both faces sloped
Up to and including 4 diameters	1/3 turn	1/2 turn	2/3 turn
Over 4 diameters but not exceeding 8 diameters	1/2 turn	2/3 turn	5/6 turn
Over 8 diameters but not exceeding 12 diameters	2/3 turn	5/6 turn	1 turn
(see Note 5)			

NOTES:

1. Tolerance on rotation: for 1/2 turn or less, one-twelfth of a turn (30°) over and nil under tolerance; for 2/3 turn or more, one-eighth of a turn (45°) over and nil under tolerance.
2. The bolt tension achieved with the amount of nut rotation specified in the table will be at least equal to the minimum bolt tension specified in the following table.
3. Nut rotation is the rotation relative to the bolt, regardless of the component turned.
4. Nut rotations specified are only applicable to connections in which all material within the grip of the bolt is steel.
5. No research has been performed to establish the turn-of-nut procedure for bolt lengths exceeding 12 diameters. Therefore, the required rotation should be determined by actual test in a suitable tension measuring device which simulates conditions of solidly fitted steel.

nut turn, once the plastic zone has been entered. It is therefore clear from the diagram just how much less the preload scatter is for the same degrees-of-turn accuracy range when tightening inside the plastic zone.



We receive calls, from time to time, for assistance in calculating the turn-of-nut angle on standard (AS1110) Class 10.9 and (AS2465) Grade 8 bolts. So far, these calls have been largely associated with bolts in the larger diameters or otherwise when users have wanted the added accuracy over the **TORQUE WRENCH** method. There is no reason why the **TURN-OF-NUT** method should not be used more widely and this article may encourage more users to do so where there is concern about final clamp load.

Interestingly, some experts have said, there is no reason why, in special circumstances, tightening right up to the theoretical yield point should not be employed with Grade 8 and Class 10.9 bolts. The limiting factor here is the degree to which the relationship between torque and bolt tension and/or turn-of-nut and bolt tension can be relied upon. Today however, highly accurate electronically controlled torque wrenching equipment is available in many industrialised countries. This equipment is already in use, but so far within a given bolt diameter range only. The equipment is produced in both hand held wrench and multi-spindle production line unit types. It is highly reliable for running high strength bolts accurately right up to the yield point, or even a little beyond, if set for required angles of turn after snug. Alternatively, one can use load indicating washers with their ± 10% accuracy. So, if for example in not wanting to actually exceed yield stress, and if load indicating washers were available to suit, one could theoretically aim to tighten to 90% of yield point. Within the range of accuracy of the load indicating washer, the stress in the thread would then be somewhere in the range 81% — 99% of yield.

Although far from approaching the typical every day situation, it would surely be a matter of how special and critical the application is that would warrant the procedure. In any justifiable case, when bolts are tightened right up to yield and certainly into the plastic zone, it is important that routine maintenance and/or disassembly/reassembly (where the latter would be a factor, as with movable plant and equipment) controls be in place sufficient to preclude the possibility of bolts being used more than once.

SNUG

REFERENCE has been made through this article to the condition — **SNUG**. This denotes the bringing firmly and solidly together of the joint faces, prior to the commencement of the measurement of nut turn. It is generally described in the structural field as Podger spanner tight. Obviously there will be variation from person to person in using the wrench. One way which has been recommended to minimise person-to-person and even bolt-to-bolt (with the same person) variation in snug, is to bring the components together into the snug position by pulling the wrench with the fingers hooked only at the first joints. When a significant increase in effort is felt, the joint is snug and the turn-of- nut process can begin. One may well need to go beyond relying completely on this procedure on large diameter bolts.

CLOCK-FACE CONTROL METHOD FOR TURN-OF-NUT TIGHTENING

WITH the nut at **SNUG**, lines or marks are made, one at a corner on the nut and another on the joint in line with that corner. This is taken to be 12 o'clock. The positions at the opposite corner would be taken as 6 o'clock and the positions in between as 2 o'clock and 4 o'clock, respectively. As the angle turned though to these positions represent 2/12ths and 4/12ths of the circle or 60° and 120°, respectively, it follows that 1 o'clock would mean a 30° turn, 3 o'clock a 90° turn, 5 o'clock a 150°

“This is taken to be 12 o'clock”

turn and of course 6 o'clock a 180° turn and so on. The following table may be useful.

Turn	Degrees	Clock Reading (Hrs. & Mins.)
0.083 (1/12)	30	1.00
0.125 (1/8)	45	1.30
0.167 (1/6)	60	2.00
0.208	75	2.30
0.25 (1/4)	90	3.00
0.292	105	3.30
0.333 (1/3)	120	4.00
0.375 (3/8)	135	4.30
0.417	150	5.00
0.5 (1/2)	180	6.00
0.667 (2/3)	240	8.00
0.833 (5/6)	300	10.00
1	360	12.00

"Not the same critical need for very high strength bolts on these early planes."

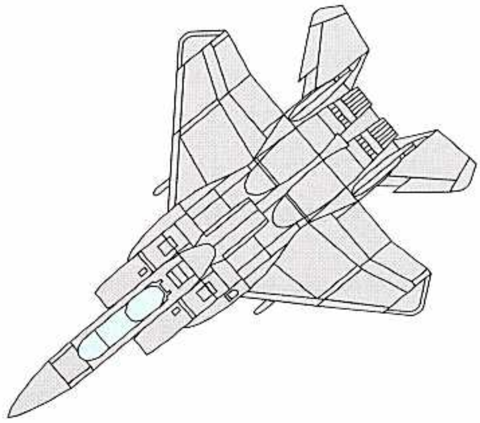
FINAL NOTE ON TIGHTENING PROCEDURES

WE certainly hope that readers will find this article interesting and helpful. It must be said, however, that although Hobson Engineering always tries to be of service in as many ways as possible, we cannot however take responsibility for whatever values users may elect to apply in practice for the tightening of bolts, screws or studs.

The world of engineering has come a long way since the horse and buggy days and the early bi-planes that took to the skies at the beginning of this century. Most forms of machinery then were not subjected to the high stresses which are an unavoidable aspect of the high performance equipment in use today — be it industrial equipment, machine tools, road transport vehicles, ships or aircraft in all their various forms. There is no alternative to high strength bolts for safety and reliability nowadays. And as the need for increased strength has risen, so has the requirement for taking full advantage of that strength when tightening up.

Not the same critical need for very high strength bolts on these early planes.

Only very high strength bolts can support the stresses of modern day equipment, and this is certainly not confined to the advanced aerial fighting machines in world-wide use.



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