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JANUARY-MARCH 2017



Exhaust Headers
Getting the heat out

Critical Fasteners
Reuse or replace

Customized Training
Training on your terms

Valve Stem Seals
Control the flow



THIS OLD SCHOOL, OVER-THE-TOP ENGINE BUILT BY MIKE MAVRIGIAN FEATURES A GM LS PLATFORM, BORED AND STROKED TO 408 CID, WITH A 4.030" BORE AND 4.000" STROKE COMBINATION.

The Reuse or Replacement of Critical Fasteners

BY STEVE SCOTT

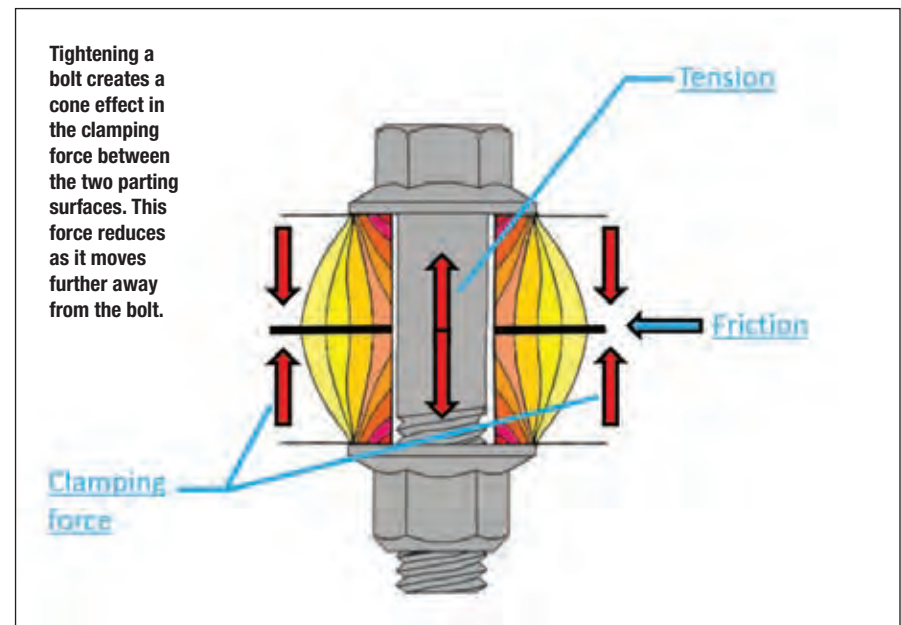
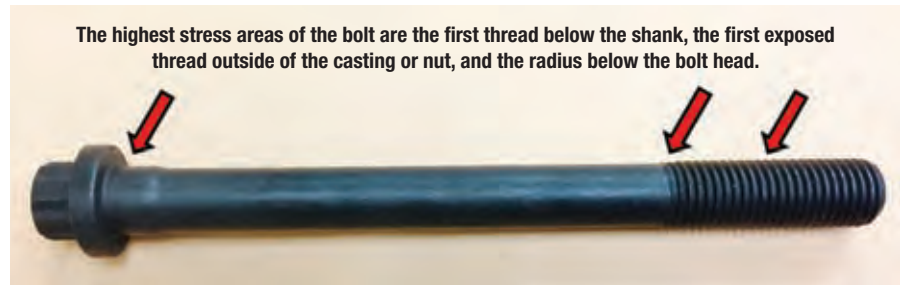
There are various criteria for inspection and replacement of critical fasteners, and when it comes to cylinder head bolts it can be somewhat confusing. OE Manufacturers often publish general guidelines for reusability, and occasionally will reference a bolt length specification to determine if the bolt is within limits. However, those are only guidelines and the actual application may call for head bolts to be replaced after each use, or after a specific number of uses. Even more confusing is that some manufacturers include the head bolts in the appropriate gasket kits, while others do not. So, depending on what is included (or not included) in the kit is not a guarantee that a bolt is reusable. Referring to the latest OE service publications is the best resource. Researching this information in advance will help to avoid any unexpected expense during assembly.

Head bolts are not necessarily an inexpensive item. A Cat 3500 series head bolt lists for around \$17.00 each. Considering there are 8 primary bolts per cylinder head (\$17.00 X 8=\$136.00), and depending on the design, there can be 8 to 24 heads on the engine. That is a considerable expense that will need to be budgeted or priced into an engine build.

The highest stress areas of the bolt are the first thread below the shank, the first exposed thread outside of the casting or nut, and the radius below the bolt head.

Even if you are replacing the head bolts with new ones, the inspection guidelines are a wise precaution to follow. Damage to the shank, threads, and under side of the bolt head can cause a stress riser, and/or compromise the clamping force of the bolt. Damage to the threads or the underside of the fastener flange can affect torque. The increase in friction can result in inaccurate rotation whether it be TTY, Torque to Angle or just straight torque value.

Tightening a bolt creates a cone effect in the clamping force between the two parting surfaces. This force reduces as it moves further away from the bolt. If the



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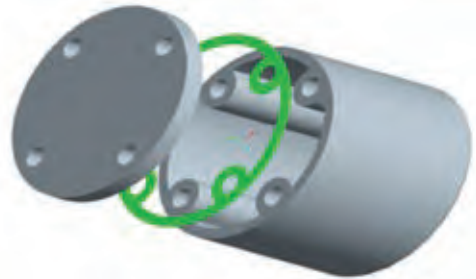
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bolt is tightened properly it should only experience a tension (stretching) load. Friction between the two parting surfaces created by the clamping force would carry the shear or bending loads.

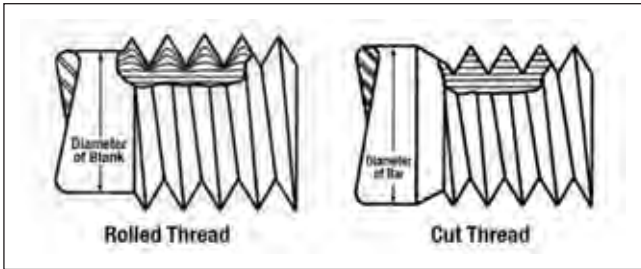
The amount of force that a properly tightened bolt(s) apply is amazing. Something as small and simple as a steel flange using four 8mm bolts and tightened to 18.5 lb-ft (24.97 Nm) can create over 3500 pounds of tension (clamping) force per bolt.

Given:
8mm(.315") bolt
18.5 ft-lbf of tightening torque
0.2 coefficient of friction (steel on steel)
Where $F=T/(c*D)$:
 $F=(18.5 \text{ ft-lbf}/(.2*.315\text{in}))* (12\text{in}/\text{ft})= 3524 \text{ lbf}$

Most cylinder head bolts are made from forged blanks with rolled threads in order for cylinder head bolts to withstand the hundreds of pounds of torque and the operating forces of the engine. Forging the head on the bolt optimizes the grain structure and increases fatigue life to three times that of a machined bolt. Rolling the threads, as opposed to cutting them, creates continuous unbroken grain lines following the contours and increases the tensile and shear strength.



Flange force.



This illustration shows a rolled thread in comparison to a cut thread. Rolling the threads, as opposed to cutting them, creates continuous unbroken grain lines following the contours and increases the tensile and shear strength.

(continued)

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The method for tightening (torqueing) cylinder head bolts varies by OE manufacturers, and by application. For example (for reference only):

A Detroit 60 Series engine prior to September 2002 used a torque of 250-285 Nm (185-210 lb-ft), and after September 2002 uses 198 Nm (220 lb-ft). Torque in sequence and then repeat to verify.

For a Cummins ISX, the first torque is 150 lb-ft, then a second torque of 300 lb-ft, followed by 90° turn or angle.

For a 3500 series Cat engine, one example for the main head bolts instructs:

- a. Tighten Bolt (1) through Bolt (8) in numerical sequence to a torque of 30 ± 5 N·m (22 ± 4 lb-ft).
- b. Tighten Bolt (1) through Bolt (8) in numerical sequence to a torque of 100 ± 15 N·m (75 ± 11 lb-ft).
- c. Tighten Bolt (1) through Bolt (8) in numerical sequence to a torque of 450 ± 15 N·m (330 ± 11 lb-ft).
- d. Loosen Bolt (1) through Bolt (8) in numerical sequence an angle of $90^\circ \pm 5^\circ$.
- e. Tighten Bolt (1) through Bolt (8) in numerical sequence to a torque of 490 ± 15 N·m (360 ± 11 lb-ft).
- f. Tighten Bolts (9) and (10) to a torque of 55 ± 10 N·m (40 ± 7 lb ft)

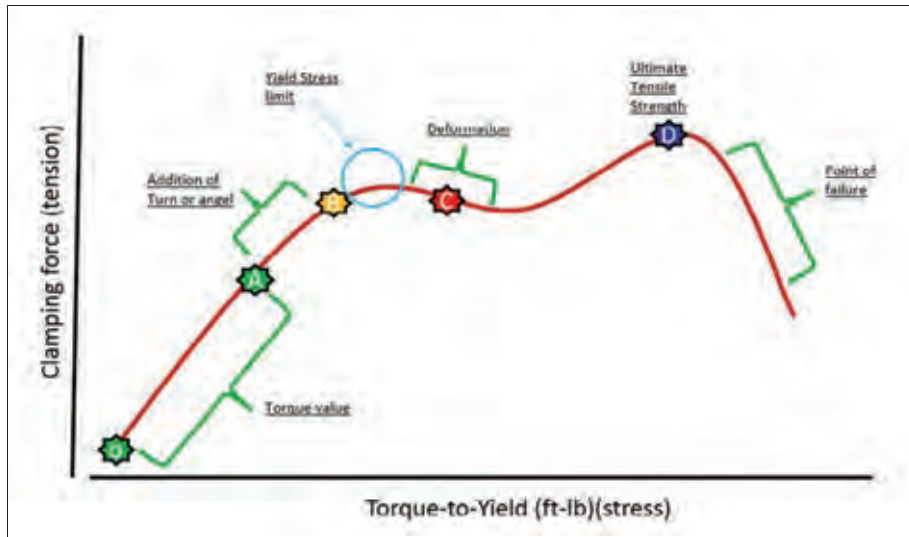
And for a C10 or C12 Cat engine, the process is even more involved:

- a. Tighten bolt (1) through bolt (26) in a numerical sequence to a torque of 160 ± 15 N·m (120 ± 11 lb-ft).
- b. Tighten bolt (1) through bolt (26) again in a numerical sequence to a torque of 160 ± 15 N·m (120 ± 11 lb-ft).
- c. Place an index mark on each bolt head. Turn bolt (1) through bolt (26) in a numerical sequence for an additional 90 degrees (1/4 turn).
- d. Loosen bolt (1) through bolt (26) until the washers are loose under the bolt heads.
- e. Tighten bolt (1) through bolt (26) in a numerical sequence to a torque of 160 ± 15 N·m (120 ± 11 lb-ft).
- f. Tighten bolt (1) through bolt (26) again in a numerical sequence to a torque of 160 ± 15 N·m (120 ± 11 lb-ft).
- g. Place an index mark on each bolt head. Turn bolt (1) through bolt (26) in a numerical sequence for an additional 90 degrees (1/4 turn).
- h. Tighten bolt (27) through bolt (33) in a numerical sequence to a torque of 28 ± 7 N·m (20 ± 5 lb-ft).

The science and mathematical calculations that go into determining the torque sequence, value, material, load, and processes are very involved. And, altering these values and procedures can have negative effects. Cylinder head bolts function like a spring (tension) that maintains a constant clamping (crush) force between two surfaces. How the bolt is designed, and consideration for the forces it has to withstand, determines if it can be used multiple times, or if it is only for a single use. The torque-to-turn, or torque-to

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angle procedures help dial in the stress force that is applied to the bolt.

The graph above helps illustrate the proportional limit and yield point for a bolt:

- Points O-A represent the tension zone that torque applies to the bolt
- Points A-B would be tension added by additional turn or angle (i.e. ¼ turn or 90° degrees). Up to this point there is no permanent change, and the bolt can be reused.
- C is the point at which the bolt material begins to permanently deform. It will not return to its previous state. The bolt cannot be reused.
- D is the bolts' point of ultimate tensile strength. Any addition load beyond this point can result in fracture or failure.

Once a bolt passes its yield stress limit, it is permanently compromised and will not return to its previous state. Some technicians are tempted to add additional foot pounds of torque, or degrees of turn (angle), beyond the manufacturers specifications. Although added torque may not stretch the bolt past the yield stress limit or ultimate yield point, it may have negative effects on other components (head gasket, liner flanges, castings, etc.). If the bolt is stretched past these critical points, any resulting failure is not the fault of the bolt. Unfortunately, unless the bolt shows signs of failure, there is no visible way to tell that the bolt has been compromised.

The engineering and manufacturing specifications for a specific bolt can identify the elastic limit and yield points, but to illustrate the physical effects of altering the torque values, we measured the length of six new head bolts for 60

Series Detroit engine, and then applied various torque and torque turn values. The graph shown below displays our findings. After the 1st torque at 220 lb-ft, the bolts measure identical to their beginning length. We repeated this exercise again on these bolts at the same 220 lb-ft, and found only a slight change in the length for 2 of the 6 bolts. For the third test, we increased the torque to 250 lb-ft and added a 45° turn to one bolt, and a 90° turn to 2 of the other bolts. This is the point where we began to find permanent deformation. The bolt with the 250 lb-ft plus the 45° turn was able to return to its original length, but both of the bolts with the 250 lb-ft plus the 90° turn have reached their point of no return. At some point the addition of the 45° to 90° turn compromised the integrity of these two bolts, resulting in permanent deformation. And, it is important to note that there was no visible difference in the bolts.

Although the added foot pounds of torque and additional degrees of turn used in our experiment may be higher than most technicians would add to a given assembly specification, an important point to consider is that the exact limitations for any bolt are seldom known at the time of assembly, and visible inspection may not reveal its structural condition. If the OE manufacturer requires the bolt be replaced after each use, following their recommendation is the best practice. The seemingly high cost of the head bolts can be substantially less than the cost of the potential damage and second repair due to bolt failure. Also, for engines where their history is not known, and for applications that require bolts to be replaced after a set number of uses, a decision will have to be made as to risking reusing the bolts versus the expense of replacing them. ■

Bolt	Beginning length	1ST torque @220fP	2ND torque @ 220FP	3RD torque @250FP	
A	7.487"	7.487"	7.487"	plus 90° turn	7.495"
B	7.489"	7.489"	7.489"	7.489"	
C	7.485"	7.485"	7.487"	plus 45° turn	7.487"
D	7.482"	7.482"	7.482"	7.482"	
E	7.492"	7.492"	7.495"	plus 90° turn	7.500"
F	7.485"	7.485"	7.485"	7.485"	



Steve Scott joined the service department at IPD in 1982, working with parts, service and sales for a variety of equipment, diesel, and natural gas engines. Since 2004, he has been the director of product development and technical support for IPD. For more information, email sscott@ipdparts.com.