



SURFACE VEHICLE INFORMATION REPORT

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(R) Torque-Tension Tightening for Fasteners (Inch and Metric)

RATIONALE

Because the concepts of torque and tension are the same for both inch and metric fasteners and only the units of measurement differ, this document represents the combination of two existing SAE documents, SAE J1701 and SAE J1701M. In addition to combining the documents, this edition has been completely revised, incorporating many of the same concepts but adding much more explanation and detail to each section.

FOREWORD

Fundamentally, threaded fasteners are required to create a clamping force or load on the assembled joint to prevent loosening. In order to accomplish this, tensile load is often applied onto a bolt, screw, or stud by either tightening into a threaded component or by a nut tightened onto the bolt, screw, or stud. In general terms, the tensile load in the bolt, screw, or stud will create an equal and opposite load in the components it is clamping - known as a clamp force. Although clamping force or fastener tension can be measured by load cells, strain gauges, ultrasonics, or pressure-sensitive films, these methods are often impractical for high-production assembly applications. What is relatively simple and cost effective to measure is the torque applied during assembly. Fortunately, there is a direct relationship between the applied torque and the clamp force created. This allows torque-tension tightening to be a practical and effective method for assembly, but only when the relationship between torque and tension and the inherent variability are understood.

1. SCOPE

This SAE Information Report is provided as an advisory guide and is not intended to be made a procurement requirement. Individual application discretion is recommended. The content has been presented as accurately as possible, but responsibility for its application lies with the user. The document covers a number of the variables in the torque-tension relationship: friction, materials, temperature, humidity, fastener and mating part finishes, surfaces, and the kind of tightening tools or equipment used. With an understanding of the variables to be considered, several methods to determine and tighten fasteners using the torque-tension relationship are identified.

This guide is limited in application to fasteners with ISO-metric or UN series threads. Other thread types, such as self-tapping or thread forming, may apply to some aspects of this standard but are not specifically covered.

The procedures described in this document are based on general factors for the determination of the torque-tension relationship for the use of end users. For critical joint design, the assembly torque values should be determined by experimentation using the exact assembly components.

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2. REFERENCES

2.1 Applicable Documents

The following publications form a part of this specification to the extent specified herein. Unless otherwise indicated, the latest issue of SAE publications shall apply.

2.1.1 SAE Publications

Available from SAE International, 400 Commonwealth Drive, Warrendale, PA 15096-0001, Tel: 877-606-7323 (inside USA and Canada) or +1 724-776-4970 (outside USA), www.sae.org.

SAE J429 Mechanical and Material Requirements for Externally Threaded Fasteners

2.1.2 ISO Publications

Available from International Organization for Standardization, ISO Central Secretariat, 1, ch. de la Voie-Creuse, CP 56, CH-1211 Geneva 20, Switzerland, Tel: +41 22 749 01 11, www.iso.org.

ISO 898-1 Mechanical properties of fasteners made of carbon steel and alloy steel - Part 1: Bolts, screws and studs with specified property classes

ISO 16047 Fasteners - Torque/clamp force testing

2.1.3 NASA Publications

Available from NASA Technical Services, NASA STI Program STI Support Services, Mail Stop 148, NASA Langley Research Center, Hampton, VA 23681-2199, Tel: 757-864-9658, Fax: 757-864-6500, <http://ntrs.nasa.gov/>.

NASA-STD-5020A Requirements for Threaded Fastening Systems in Spaceflight Hardware

2.1.4 U.S. Government Publications

Copies of these documents are available online at <https://quicksearch.dla.mil>.

MIL-HDBK-60 Threaded Fasteners - Tightening to Proper Tension

2.1.5 VDI Publications

Available from DIN Media at <http://dinmedia.de/en>.

VDI 2230-1 Systematic calculation of highly stressed bolted joints - Joints with one cylindrical bolt

2.1.6 Other Publications

Bickford, J.H. (1995). *An introduction to the design and behavior of bolted joints* (3rd ed.). Taylor & Francis Group.

Motosh, N. (1976). Development of design charts for bolts preloaded up to the plastic range. *J. Eng. Ind.*, 98(3), 849-851.

3. DEFINITIONS

BOLTED JOINT: Assembly consisting of an externally threaded fastener (bolt, screw, or stud), an internally threaded component (nut, tapped hole, or hole tapped by the fastener), and any number of components being clamped. The fasteners in a bolted joint are used to create clamp load that resists external working loads acting on the joint.

CLAMP LOAD: Sometimes called preload or initial load, it is a compressive force that exists at the interface between two members generated through the cumulative effect of tightening one or more fasteners. Clamp load is typically an equal and opposite force to the cumulative bolt tension. It is typically expressed in pounds-force (lbf) or newtons (N).

TENSION: The axial load that has a tendency to stretch an externally threaded fastener. It is applied by tightening a fastener into a bolted joint in order to create an equal and opposite clamping load. It is typically expressed in pounds-force (lbf) or newtons (N).

TORQUE: The tendency of a force to rotate an object about its axis. The magnitude of the torque is the applied force multiplied by the distance from the rotational axis (often referred to as the lever arm). It is typically expressed in foot-pounds (ft-lb), inch-pounds (in-lb), or newton-meters (N·m).

4. JOINT DESIGN

In order to perform the intended function, a bolted joint must be designed such that the clamp load achieved is sufficient to withstand the external forces acting on the joint. These forces may be axial, transverse, bending, or any combination of these. In addition, all bolted joints have a tendency to relax or lose clamp load after tightening, and this relaxation must be taken into account when determining the initial required preload in the joint. Joint design, including the determination of bolt size and strength, spacing, and target preload during assembly, is a complex process and beyond the scope of this guide. For additional guidance, refer to documents such as VDI 2230-1, MIL-HDBK-60, and NASA-STD-5020. Once a target preload has been determined through appropriate joint design practices, the information that follows can be used as a reference when determining the proper installation torque.

5. TORQUE-TENSION RELATIONSHIP

As noted in Section 4, the goal when assembling a bolted joint is to achieve the target preload that will result in a clamp load that is sufficient to resist the applied forces throughout the service life of the joint. However, measuring clamp load directly in a production environment is typically not practical. What is relatively simple and cost effective to monitor and control is the torque applied during assembly. Fortunately, there is a direct relationship between the torque applied and the resulting clamp load. If that relationship can be determined for the joint in question, the clamp load generated can be estimated from the applied torque.

The following equation representing the relationship between torque and clamp load was presented by Motosh (1976) and simplified by Bickford (1995) to yield:

$$T = F_P \left(\frac{P}{2\pi} + \frac{\mu_t r_t}{\cos \beta} + \mu_n r_n \right) \quad (\text{Eq. 1})$$

where:

T = applied torque (inch-pounds, newton-millimeters)

F_P = preload created by the fastener (pounds-force, newtons)

P = pitch of the threads (inches, millimeters)

μ_t = coefficient of friction between the bolt and nut threads

r_t = effective contact radius of the threads - assumed as half the pitch diameter (inches, millimeters)

β = half-angle of the threads (30 degrees for UN or ISO profile threads)

μ_n = coefficient of friction between the bearing surface of the turned fastener and the joint member

r_n = effective radius of contact between the turned fastener and the joint member (inches, millimeters)

Equation 1 can be considered the sum of three terms, where the first term ($F_P \frac{P}{2\pi}$) is the torque required to stretch the bolt and create tension or clamp load, the second term ($F_P \frac{\mu_t r_t}{\cos \beta}$) is the torque created due to friction between the threads, and the third term ($F_P \mu_n r_n$) is the torque created due to friction between the bearing surfaces.

This equation, corroborated with empirical testing, demonstrates that ~90% of the energy expended during the fastener tightening process is lost to friction (lost as heat) and that ~10% is converted to potential energy stored as fastener tension. The friction resistance between the external and internal threads accounts for ~40% of frictional losses, and the frictional resistance between the fastener element that is turned (bolt or screw head or nut) and the mating surface (bearing surface) against which the fastener element is turned accounts for ~50% of frictional losses.

NOTE: The listed percentages are general approximations and will fluctuate based on specific joint conditions.

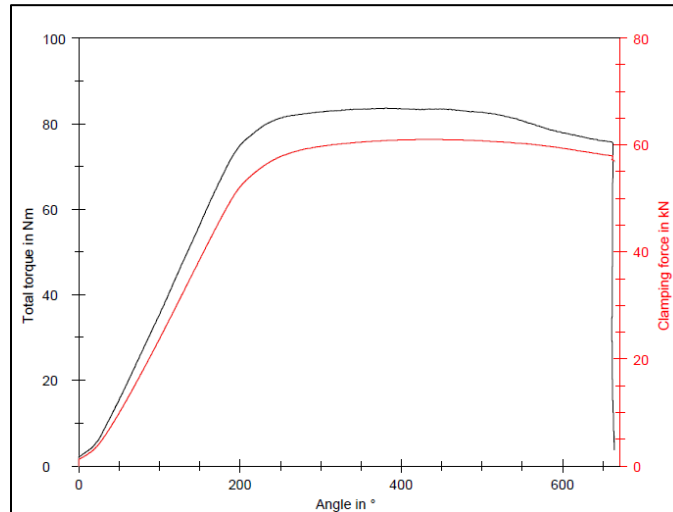


Figure 1A - Plot of torque and clamp load versus tightening angle

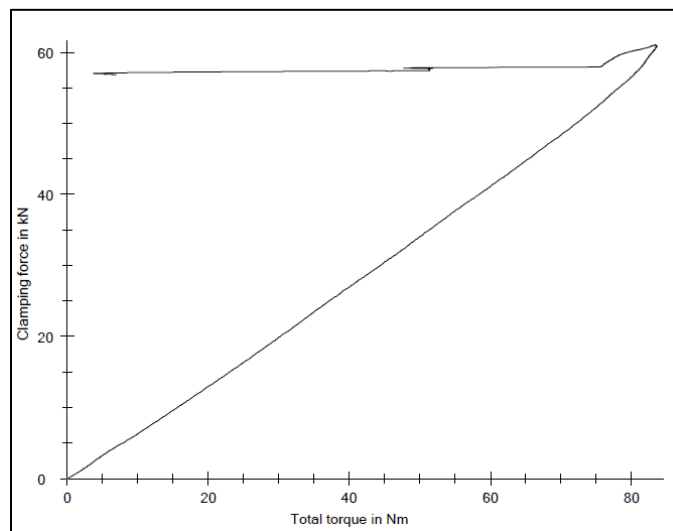


Figure 1B - Data from Figure 1A re-plotted as clamp load versus torque

6. VARIABLES IN THE TORQUE-TENSION RELATIONSHIP

Other than clamp load (F_p) and torque (T), the remaining variables in Equation 1 consist of geometric factors and friction coefficients. While the equation is quite straightforward to use if fastener and clearance hole geometry are known, it is idealized, and some additional guidance on how fasteners, joint components, and assembly conditions may affect the torque-tension relationship are illustrated in Figure 2 and described further in the following paragraphs.

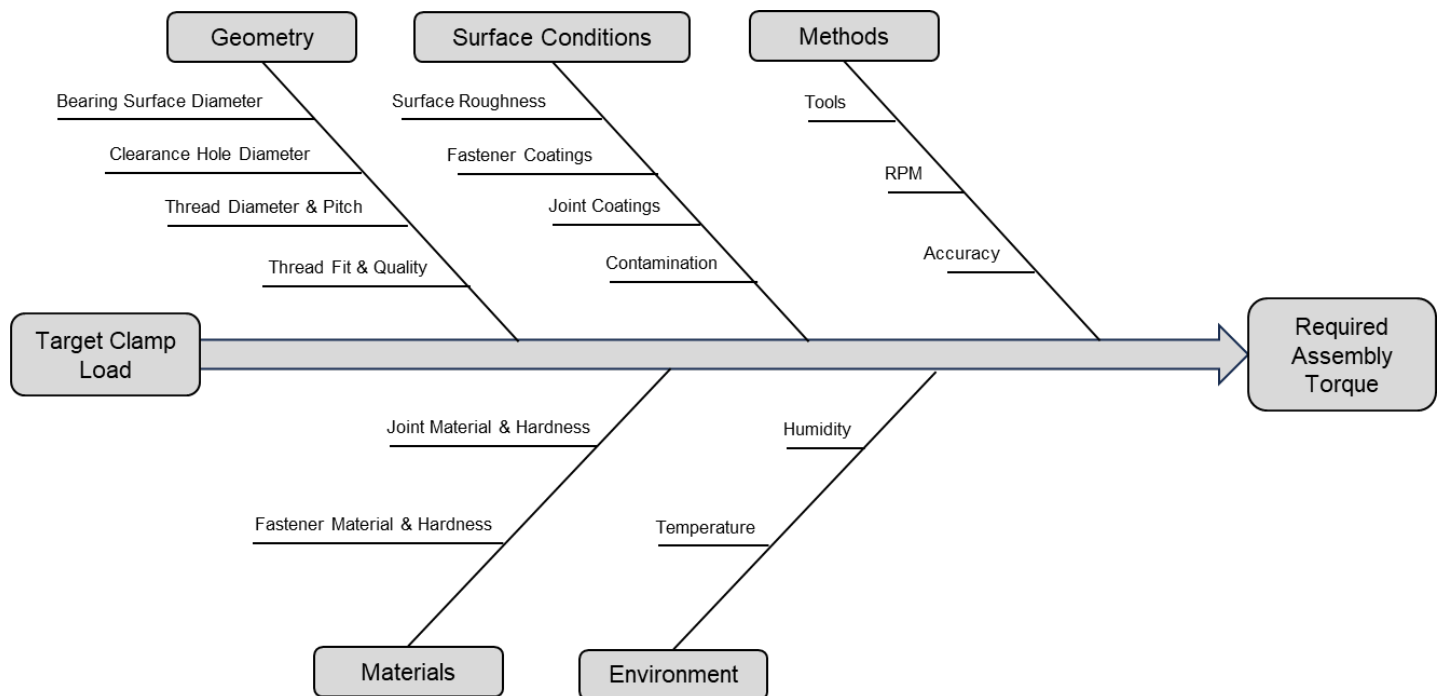


Figure 2 - Variables in the torque-tension relationship

6.1 Geometry

The fastener geometry has a direct effect on the torque-tension relationship in terms of bearing surface areas, thread pitch, and thread fit. As the bearing surface of the turned fastener increases in diameter, the effective radius that the friction resistance is applied at increases, increasing the torque at a given clamp load. For example, all other factors being equal, a bolt with a flanged head would require more torque than a bolt with a standard hex head to achieve the same clamp load. Additionally, changes in bearing surface form (e.g., convex versus concave flange) can also significantly affect the torque-tension performance due to the effective contact area.

In theory, clearance hole geometry has a counterintuitive effect. As the clearance hole gets larger, the torque at a given clamp load increases as calculated using Equation 1. In practice, the effect of both fastener and clearance hole geometry is more complex due to increased surface pressures and potential misalignment due to the increased clearance.

When washers are placed under the component being tightened, they can have a variable effect on the torque-tension relationship due to changing the effective bearing area. If the washer remains stationary during tightening, the bearing area is that of the bolt or nut against the washer. However, if the washer rotates with the bolt or nut, the bearing area becomes that of the washer against the clamped component. The tendency for either situation to occur is based on the friction conditions of all components and is difficult to predict. A further complication arises when this changes during tightening; i.e., the washer starts or stops rotating. The near instant change in effective geometry and related friction conditions can create a significant change in torque at a given clamp load. An example of this effect can be seen in Figure 3.