Pressure Distribution & Calculation of Pressure Cone Angle

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How measuring pressure distribution can be used to determine bolted joint stiffness and avoid compressive yield.

Perhaps the most influential factor in determining how axial loads are reacted in a bolted joint is its stiffness ratio, defined as the stiffness of the bolt relative to the stiffness of both the bolt and the joint. The calculation of this ratio requires that the area, length and elastic modulus of each geometry is known. The only real challenge to calculating bolt stiffness is estimating the contribution of elongation within the head and nut. An experimental look at this question was published in the October 2008 issue of this magazine.

The calculation of joint stiffness faces a more fundamental uncertainty—how is the bolt and nut load transferred within the joint members? The practical problem is defining a shape that can be assumed to encompass these loads. The most common embodiment is the truncated "pressure cone" shown in **Figure 1a**. Studies have suggested the half-angle α is influenced by joint geometry and materials, with estimates ranging from 25° to 45° offered. These estimates have been based on both analytical and experimental methods.

Calculating α for a Single Set of Joint Members

This article describes a test conducted to calculate α for a single set of joint members. As part of the test, the influence of head and nut geometry was explored. To define the pressure cone, not only does an included angle need to be specified, but also the cone's origin. **Figure 1a**, **Figure 1b** and **Figure 1c** show commonly accepted estimates for the fastener bearing diameter D_b, which defines the start of the pressure cone. These estimates will be examined by testing four bolt/nut combinations chosen to provide a range of diameters and stiffness. A summary of each combination is listed in **Table 1**. Fasteners are 1/2" – 12 x 3" *SAE Grade 8* or *ASTM* equivalent,

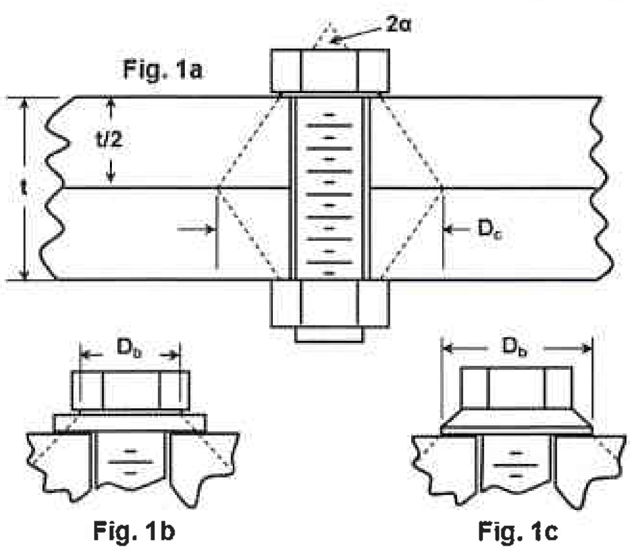

Fig. 1 — Pressure cone.

Table 1. Test Matrix

Test #	Bolt	Washers	Nut
1	Hex Head Cap Screw	None	Hex Nut
2	Hex Head Cap Screw	Flat (2)	Hex Nut
3	Hex Flange Head Cap Screw	None	Hex Flange Nut
4	12 Pt Flange Head Cap Screw	None	12 Pt Flange Nut

Table 2. Relevant Fastener Dimensions

Test #	Description	Width Across Flats, in	OD, in	Bearing Dia, in
1,2	Hex Head Cap Screw	3/4	-	0.685
1,2	Hex Nut	3/4	-	0.735
2	Flat Washer	-	1-1/16	1.065
3	Hex Flange Head Cap Screw	3/4	1 1/16	1.000
3	Hex Flange Nut	3/4	1	0.980
4	12 Pt Flange Head Cap Screw	1/2	3/4	0.740
4	12 Pt Flange Nut	9/16	13/16	0.820

chosen for availability. The fastener dimensions relevant to the test are shown in **Table 2**.

The test joint was constructed from two cylinders of hardened steel with their ends ground flat and parallel to one another. The cylinder was 1.1" thick, representing a rigid member in which a full load profile will form, as opposed to thin clamp plates which are prone to flexure.

The test method used for estimating the cone half-angle α was to determine how large a diameter was loaded at the interface of the two clamp plates, (D_c in **Figure 1a**) and calculate the angle from the bearing diameter D_b. D_c was determined by placing pressure-sensitive film between the plates, tensioning the bolt in each of the four fastener sets to approximately 10,000 lb and then digitizing the film. Bolt tension was measured ultrasonically.

The test setup is presented in **Figure 2** with the joint assembly highlighted. The film scanner that was used is shown in **Figure 3** with a digitized test sample in the laptop's display. Two samples were tested for each combination. Results are based on the test exhibiting the most symmetrical pressure pattern. In addition to indicating the diameter loaded, the scanner software can generate pressure maps and profiles. The pressure profiles show that the peak load, registered at the edge of the clearance hole, was approximately 40% greater than the average pressure over the entire surface. Knowledge of peak pressure at joint interfaces, and particularly under head, is helpful in avoiding compressive yield.

As seen in **Table 2**, the bearing diameter D_b under the head and the nut are somewhat different from one another in each test. As a result of this, the maximum pressure cone diameter does not coincide with the interface of the two clamped cylinders, but instead into the cylinder which abuts the larger D_b.

Therefore, calculation of the half-angle α will be based on the smaller D_b. See **Table 3** which contains the result of those calculations.

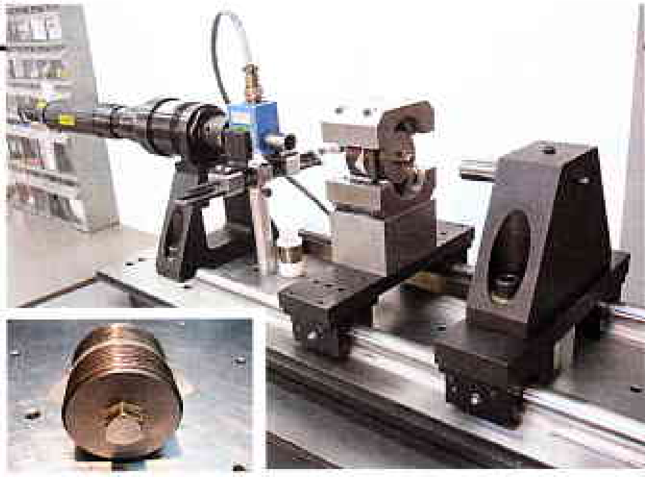

Fig. 2 — Bolt tensioning setup with a joint close-up seen at lower left.

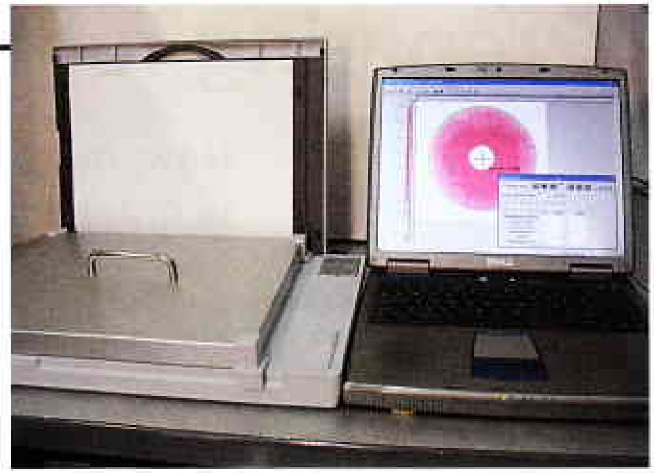

Fig. 3 — Film scanning system with test sample displayed on laptop computer.

Influence of Under Head Pressure

In two cases the actual under head bearing diameter over which pressure was applied did not match the convention shown in **Figure 1**. This was determined by creating the same joints with pressure sensitive film under the nut and bolt rather than between the clamped members. To eliminate bolt or nut rotation on the film, the bolt was tensioned to 10,000 in a load frame. The setup is shown in **Figure 4**.

Seen in **Figure 5**, the pressure profiles under the fasteners in test two show that there is a distinct high pressure area at the bolt's bearing area, an indication that the washer is not stiff enough to distribute load very effectively.

The actual pressure diameter under the head was smaller than the **Figure 1b** definition with only a 30° half-angle assumed. The flange head fasteners saw an even more distinct variation from the D_b defined in **Figure 1c**. The nuts contacted on their outer diameter (OD) while the bolt head contact was near the edge of the clearance hole.

It is my understanding that by specification this convex bias is not permitted, and my experience is that the contact profile of flange fasteners varies noticeably. As an informal survey, I gathered all M10 and 1/2" flange nuts and bolts in our lab and produced pressure profiles as shown in **Figure 5**. The result was that of four nuts, two contacted on the OD as in **Figure 5**, one contacted from the OD inward to about half the flange width and the last contacted across the entire flange face. One of the two bolts available also contacted as did the **Figure 5** nut, while the other contacted across the flange face. The impact of this variation on the torque-tension relationship is measureable, as the resisting moment created by under head friction will act at different radii.

Test Summary

The half-angles calculated from the four fastener configurations tested ranged from 36.0° to 38.2° based on actual contact under head diameters, and 35.0° to 38.2° based on theoretical contact diameters. Use of standard thickness flat washers under the hex fasteners does not eliminate the high-pressure band coinciding with the fastener bearing area.

The contact surface of flange head bolts ranged from bands at the OD to a band at the clearance hole. Flange contact on nuts ranged from the OD to across the entire face.

The peak pressure of the joint interface was approximately 40% greater than the average pressure across the face.

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Table 3. Pressure Cone Half-Angle

Test #	Description	Min Bearing Dia, in	Press Dia @ Joint, in	α, deg
1	Hex	0.683	2.32	36.7
2	Hex w/ Washer	0.810[a]	2.35	35.0
2	Hex w/ Washer	0.750[b]	2.35	36.0
3	Flange	0.980[a]	2.45	33.8
3	Flange	0.716[b]	2.45	38.2
4	12 Point	0.740	2.38	36.7

[a]As defined in Figure 1. [b]As measured from pressure sensitive film

Fig. 4 — Setup for measuring under-head pressure prior to its installation in a load frame.

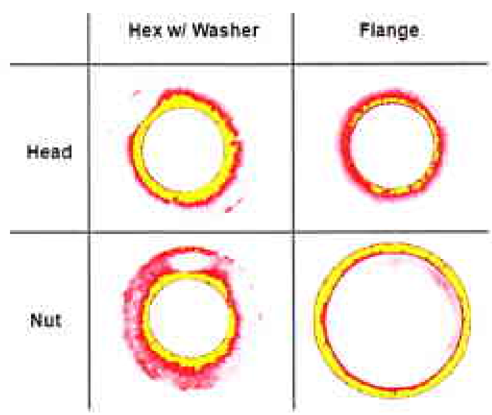

Fig. 5 — Pressure profile at fastener bearing surface.

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