Influence of a Washer on the Stress Distribution on a Joint Using Rational and Pre-Tension FEA Methods

by Michael Oliver, Ph. D

When torque is applied to a screw, a great many physical things start to occur: pressure is created on the joint's bearing surface from the screw, pressure is created on the flanks of the internal threads by the rotating external threads, and tension or clamp-load is developed. The pressure, however, just does not remain on the surface; it gets distributed throughout the volumes of the screw, nut, and jointed members. This is exactly the point of this article: to discuss exactly what the stress distribution of a threaded screwed joint is, as torque is applied and clamp-load is developed. The discussion will revolve around the stress distribution on the surface and volume of the jointed member. This discussion will also examine results of adding a washer to the underside of the rotating screw. Finally, a comparison of analytical results from rotating the socket head screw to using what is called a pre-tension (non-rotation) approach will also be discussed.

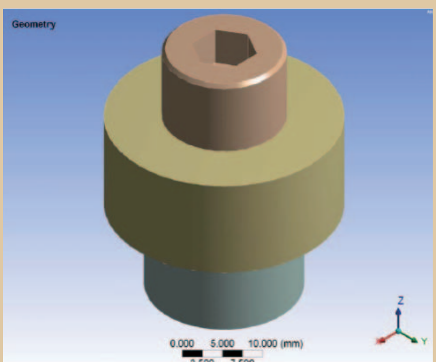

Figure 1 Image of Socket Head Screw JointFi.

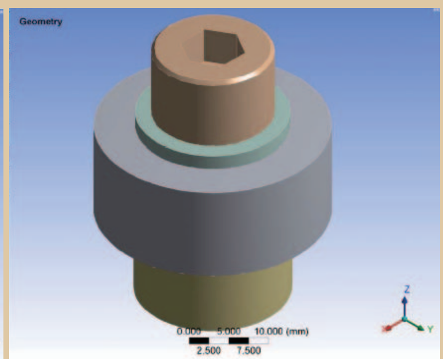

Figure 2 Image of Socket Head Screw Joint with Washer.

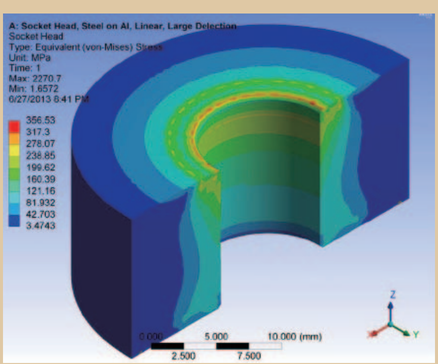

Figure 3 FEA plot of Von Mises Stress on a sectioned view of the jointed member from a screw with no washer.

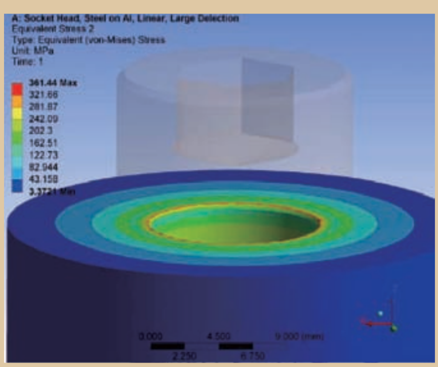

Figure 4 FEA plot of the jointed member with an overlay of the screw.

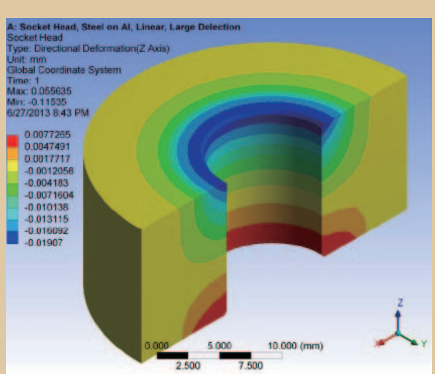

Figure 5 FEA plot of Deflection in the Z direction on a sectioned view of the jointed member from a screw with no washer.

There are numerous discussions and articles written with respect to the fact that when a washer is added to the underside of a rotating or stationary bolt or screw, the pressure and load is more uniformly distributed: as compared to running the bolt or screw alone. This has been proven with the aid of photo-elastic material that actually shows the barrel shaped stress in the joint, under clamp-load. With the invention of finite element analysis (FEA), a more detailed examination of this distribution in the joint can now take place.

Figures 1 and 2 shows the images of the two joints under examination for this article, the joints are using a M10 x 1.5 threaded screw. The first figure shows the socket head screw, joint, and internal threaded member. The second figure shows the socket head screw, washer, joint, and internal threaded member. These two images represent 3D solid models that were meshed in a FEA software package for the sole purpose of examination of the stress distribution. In these examinations, experimental data from heavy hex head bolt were used (thread and under-head coefficient of friction (CoF)) as inputs. Experimental data for the socket head screw, with and without a washer were not available. This examination is operating under the premise of all things being equal.

The first joint examined is the socket head screw, without the washer. A torque of 45 N-m was applied to the inner hexagonal surfaces of the hex feature. A clamp-load of 26,898 N was developed. Figures 3 and 4 show the Von Mises Stress of a sectioned view of the jointed member with an overlay of the screw (for reference purposes) in the latter figure respectively. Figure 5 shows the Deflection (in the Z direction) for a sectioned view of the jointed member. Note the positive and negative values of the deflection plot. This is due to the fact that the upper surface was being compressed by the rotating screw and the bottom was also being compressed, but in the opposite direction, due to the chamfered region of the internal threaded member.

The second joint discussed is the same as the preceding paragraph, but with the exception that a torque was not applied to the screw and therefore no rotation occurred. Instead, a series of Pre-Tension elements were applied to the shank of the screw to develop the clamp-load. The clamp-load developed in the preceding model was used as an input to this model. This is an approach which is used much more than applying a moment or torque for one lone reason, speed. The model using the pre-tension approach will probably solve in 10%-30% of the time required for that of the rotation approach. Figures 6 and 7 show the Von Mises Stress as well as the deformation of the sectioned jointed member respectively. Note the difference between that of the Pre-Tension and that of the rotation model (356 MPa vs. 281 MPa). This represents approximately a 24% difference between the two methods. From previous articles I have written on the subject, the most accurate method for simulating tightening a bolt or screw is through the application of a moment or torque. Therefore, this percent difference represents a trade off in accuracy and speed of the solve. More discussion on this will take place at the end of the article.

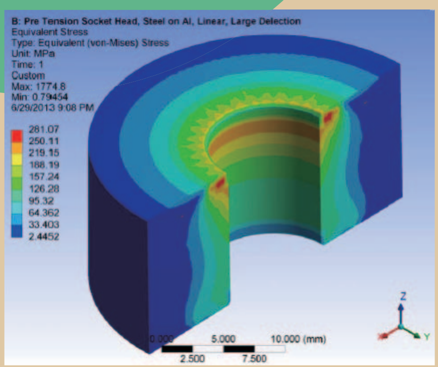

Figure 6 FEA plot of Von Mises Stress on a sectioned view of the jointed member from the Pre-Tension model of a screw with no washer.

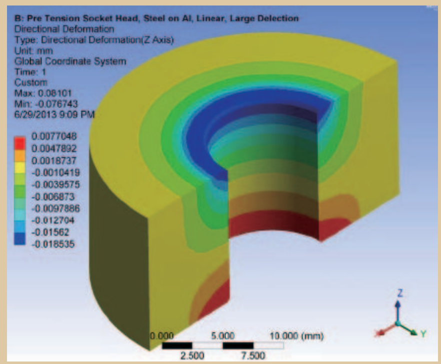

Figure 7 FEA plot of the Deformation in the Z direction on a sectioned view of the jointed member from the Pre-Tension model of a screw with no washer.

The third model adds a 2 mm thick washer to the joint from the first model, Figure 2. In this model, the CoF was set equal between the screw and the washer as well as the washer and the jointed member. As with the first model, a 45 N-m torque was applied to the internal hexagonal features of the drive of the screw. This produced a rotation as well as a clamp-load of 27,948 N. This clamp-load is about a 4% increase over the model with no washer (first model). This is probably due to the extra 2 mm in length of the shank of the screw to account for the thickness of the washer. Figure 8 and 9 shows the Von Mises Stress on the sectioned, jointed member as well as an overlay of the washer and screw respectively. Figure 10 shows the deflection on the jointed member.

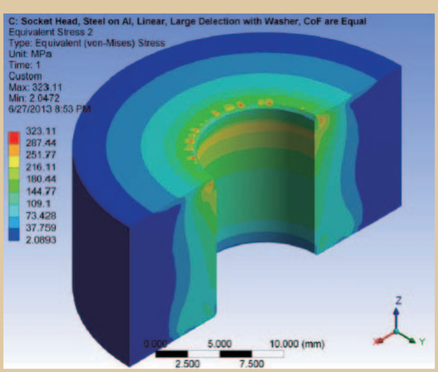

Figure 8 FEA plot of Von Mises Stress on a sectioned view of the jointed member with the addition of a washer (CoF values are equal).

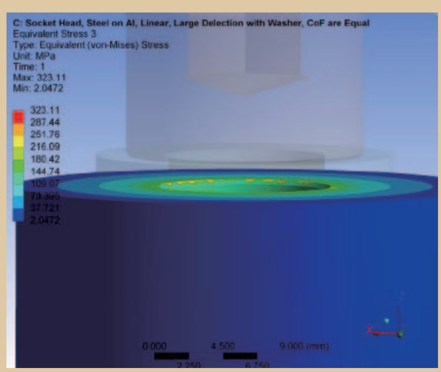

Figure 9 FEA plot of the jointed member with an overlay of the screw and washer.

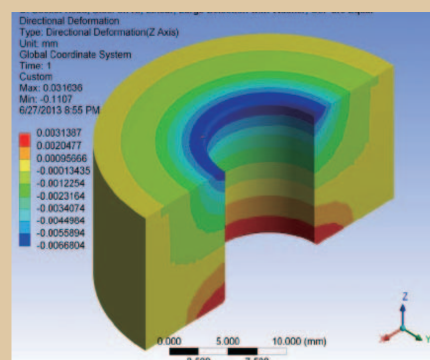

Figure 10 FEA plot of deflection in the Z direction on a sectioned view of the jointed member with the addition of a washer (CoF values are equal).

Note the maximum resulting stress in Figure 8 and compare it to that of the first model (with no washer), 356 MPa vs. 323 MPa. The addition of a washer brings about a 9% drop in stress, even though the clamp-load increased by 4%. Looking at the deflection between the two models shows that the magnitude on the upper surface (under the head of the screw) decreased by 65% with the addition of the washer (.0191 mm to .0066 mm). This is a remarkable decrease.

Investigating the stress distribution between the two models further, with and without washers, can be found in Figure 11. This figure represents linearized Von Mises Stress on the jointed member's surface, as well at a location 2 mm below the surface (plotting from the outer diameter (OD) of the jointed member to the joint's inner diameter (ID)) of the through hole. The location of the linearized stress location can be found in Figure 12. The location for the sub-surface plot is exactly 2 mm below the surface following the same path as the one on the surface. The plot shows two interesting trends. First, the linearized Von Mises Stress on the surface of both with and without

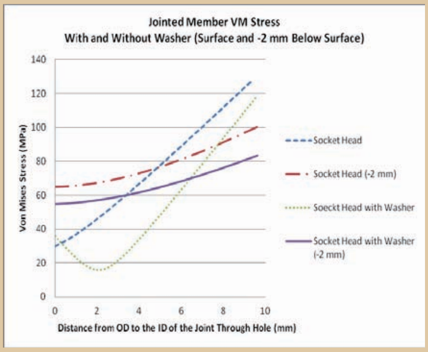

Figure 11 Plot of linearized Von Mises Stress on the surface as well as 2 mm below the surface for the, with and without washer models.

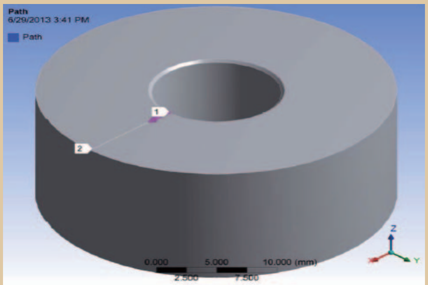

Figure 12 Image of the location of the linearized Von Mises Stress on the surface of the jointed member with and without washer.

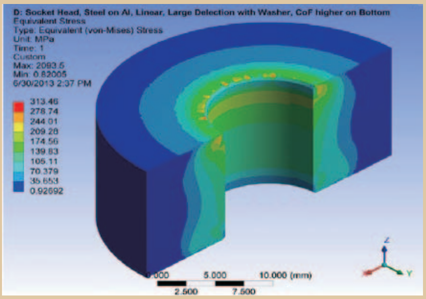

Figure 13 FEA plot of Von Mises Stress on a sectioned view of the jointed member with the addition of a washer (CoF is higher on the bottom of the washer compared to the top of the washer).

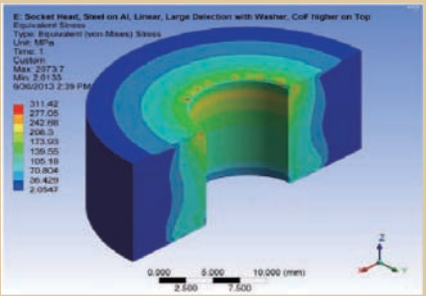

Figure 14 FEA plot of Von Mises Stress on a sectioned view of the jointed member with the addition of a washer (CoF is higher on the top of the washer compared to the bottom of the washer).

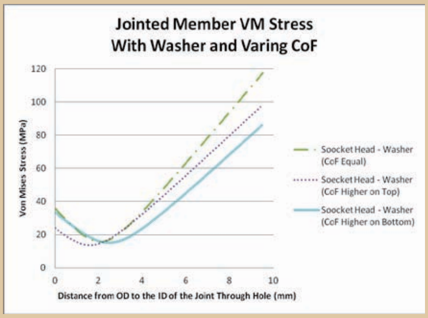

Figure 15 Plot of linearized Von Mises Stress on the surface of the washer models.

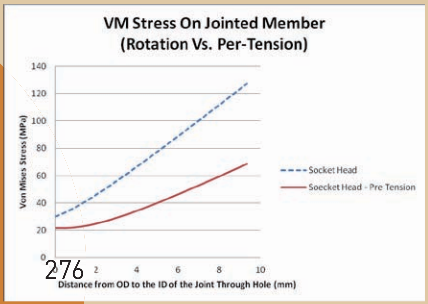

Figure 16 Plot of linearized Von Mises Stress on the surface of the jointed member due to rotation and pre-tension.

washer models is radically different. The stress on the jointed member surface, without a washer, increases almost linearly from the OD of the joint to the ID. The plot of the model with the washer drops by 20 MPa before increasing. The OD of the washer instills a sort of compressive stress on the surface of the jointed member. The Von Mises Stress 2 mm below the surface show similar characteristics between the two models. They both have similar shapes and slopes. They are only different by a magnitude of approximately 10 MPa to 17 MPa throughout the plot.

The forth model uses the same configuration as the third model did, but changes the CoF on the lower surface of the washer to be higher than that of the upper. This forced the washer to remain stationary with respect to the jointed member. The same torque of 45 N-m was applied in this model which developed a clamp-load value of 27,894 N. This value is approximately the same as the third model (CoF equal), (lower by 54 N). The plot of a sectioned view of the Von Mises Stress is shown in Figure 13 for this particular model.

The fifth model uses the same configuration as the third and fourth model, but changes the CoF on the upper surface of the washer to be higher than that of the lower. This forced the washer to rotate with the screw. The same torque of 45 N-m was applied in this model which developed a clamp-load value of 28,098 N. This value is approximately the same as the third model (CoF equal), (higher by 150 N). The plot of a sectioned view of the Von Mises Stress is shown in Figure 14 for this particular model.

The differences between the last two models, compared to the third model that had the same CoF on the upper and lower surfaces of the washer, on the sectioned plots are within 12 MPa of each other. However, looking at the surface linearized Von Mises Stress on the jointed member shows something a little different, Figure 15. The plot shows subtle differences including the model with equal CoF values had the higher stress. Also, the magnitude of the compressive (lower) stress is relatively the same between the three but they all bottom out at different distances from the ID of the hole. The Washer with the CoF higher on the lower surface had its minimum stress closer to the ID of the hole while the washer with the CoF higher on the upper surface had its minimum stress farthest from the ID of the hole. The washer with the CoF values equal had its minimum stress between the other two.

Finally, going back to the differences between the Pre-tension modeling method compared to the rotation method can be summarized in Figure 16. The linearized Von Mises Stress from the surface of the jointed member of each method is fairly distinct. For critical joints for which safety margins are close and maximum stresses are near the yield points, the use of a rotational model will give an indication of either a good or marginal design. The rotational method is also more accurate and more realistic compared to those using Pre-Tension methods.

In conclusion, there is a difference in the stress distribution on the surface and subsurface regions of a jointed member with respect to a washer. The presence of a washer does increase the stress distribution in the jointed member's volume as well as create a sort of compressive stress at the OD of the washer at the surface. This validates the photo-elastic images as well as conventional wisdom. Also, there are small differences that can arise when the CoF values change on the upper and lower surfaces of the washer. Your mileage may vary when decisions on whether to use a washer as well as establishing CoF values between mating surfaces. ■