

Contract No:

This document was prepared in conjunction with work accomplished under Contract No. DE-AC09-08SR22470 with the U.S. Department of Energy.

Disclaimer:

This work was prepared under an agreement with and funded by the U.S. Government. Neither the U. S. Government or its employees, nor any of its contractors, subcontractors or their employees, makes any express or implied: 1. warranty or assumes any legal liability for the accuracy, completeness, or for the use or results of such use of any information, product, or process disclosed; or 2. representation that such use or results of such use would not infringe privately owned rights; or 3. endorsement or recommendation of any specifically identified commercial product, process, or service. Any views and opinions of authors expressed in this work do not necessarily state or reflect those of the United States Government, or its contractors, or subcontractors.

Evaluation of Torque vs Closure Bolt Preload for a Typical Containment Vessel Under Service Conditions

Allen C. Smith
Savannah River National Laboratory
Savannah River Nuclear Solutions
Aiken, South Carolina 29808
(803) 725 2943, allen.smith@srnl.doe.gov

ABSTRACT

Radioactive material package containment vessels typically employ bolted closures of various configurations. Closure bolts must retain the lid of a package and must maintain required seal loads, while subjected to internal pressure, impact loads and vibration. The need for insuring that the specified preload is achieved in closure bolts for radioactive materials packagings has been a continual subject of concern for both designers and regulatory reviewers. The extensive literature on threaded fasteners provides sound guidance on design and torque specification for closure bolts. The literature also shows the uncertainty associated with use of torque to establish preload is typically between 10 and 35%. These studies have been performed under controlled, laboratory conditions. The ability to insure required preload in normal service is, consequently, an important question. The study described here investigated the relationship between indicated torque and resulting bolt load for a typical radioactive materials package closure using methods available under normal service conditions.

BACKGROUND

The performance of bolted closures or connections of various kinds is important in many applications, including radioactive materials packaging. Closure bolts for radioactive materials packagings must retain the lid of a package and must maintain required seal loads, while subjected to internal pressure, impact loads and vibration. The package designer determines the size and preload required for the closure bolts as part of the design analysis. The design preload must actually be achieved in assembly and operation for the required level of containment to be maintained. For a particular packaging, the assembly procedures required by the Safety Analysis Report for Packaging specify the means of establishing the required preload in the bolts. For reasons of cost and convenience, the bolt preload is usually achieved by tightening the bolts with a torque wrench. It is recognized that this means of establishing bolt load is subject to significant variation. For this reason, insuring that the specified preload is achieved in closure bolts for radioactive materials packagings has been a continual subject of concern for both designers and regulatory reviewers.

Detailed, complex studies and analyses of bolted connections and closures have been reported extensively in the literature. The literature shows the uncertainty associated with use of torque to establish preload is typically between 10 and 35% [1,2]. Although this body of literature is applicable to radioactive materials packages, much of it

addresses threaded fastener performance under controlled conditions. Under these idealized conditions, and in contrast to operational conditions, studies of bolt-load as a function of torque typically employ mechanical drive so that the bolt is tightened in continuous motion, unlike the intermittent turning associated with tightening an array of closure bolts with a torque wrench, in a “star pattern”. Accordingly, these studies must be employed with consideration of the operational conditions under which closures are assembled and the resulting additional uncertainties.

DESIGN BOLT LOAD

The design bolt load is the load each bolt in the closure must maintain to maintain seal compression against internal pressure, impact and other loads imposed in transportation (e.g., vibration). The relation between bolt load and resulting strain in the bolt is

$$P = \delta AE / L_e$$

Where:

- L_e is the initial effective length of the bolt
- δ is elongation
- P is load
- A is bolt area
- E modulus of elasticity

Standard textbooks on machine design [3] provide expressions for torque as a function of load for threads. These include thread form, friction between nut and screw and friction between head and structure being loaded (e.g., for flanges assembled with bolts, this is the flange or washer upon which the bolt head bears). The load calculated is the axial load imparted to the bolt. The state of stress in the bolt includes both the axial load and torsional loading resulting from the turning of the bolt against the friction in the nut.

This formulation is further idealized, since it assumes the flange surfaces and the matching nut and bolt bearing surfaces are precisely parallel. Typically, these surfaces are not actually, precisely parallel in practice. This occurs as a result of manufacturing, deformation during assembly, pressurization and impact in accident events.

Torque and bolt load

A useful formulation of the relationship between torque and bolt load [3] is

$$T = P(r_t(A/B) + r_c\mu_2)$$

Where:

- $A = \cos\theta_n \sin\alpha + \mu_1\cos\alpha$
- $B = \cos\theta_n \cos\alpha - \mu_1\sin\alpha$
- μ_1 = coefficient of friction for threads
- μ_2 = coefficient of friction for collar
- α = helix angle
- $\theta = \frac{1}{2}$ included thread angle (i.e., 30°)

$$\theta_n = \arctan(\tan \theta \cos \alpha)$$

$$r_t = \text{pitch radius of thread}$$

$$r_c = \text{mid radius of collar}$$

$$P = \text{bolt load}$$

$$p = \text{pitch}$$

For 5/8-11 UNC-2A bolts such as those employed for this study, with coefficient of friction of 0.45 (for un-lubricated SS on SS), the 150 ft lb torque is calculated to produce a bolt load of about 3400 lb. For lubricated threads ($\mu_1 = 0.20$), the predicted bolt load is about 8000 lb.

A simplified expression giving torque as a function of bolt load, bolt diameter, and coefficient of friction is frequently employed. For the bolt considered here, such an expression can be developed from the equation given above.

$$T = 1.89\mu dP$$

Where:

T is torque
 μ is coefficient of friction
 D is nominal diameter
 P is load

This is similar to the commonly used “nut factor” equation

$$T = KdP$$

Where, in this case, the nut factor $K = 1.89\mu$

Nut factors reported by Ganeshmurthy & Nassar [4] for un-lubricated, zinc plated M12 threads were typically between 0.2 and 0.25.

BOLT LOAD TESTING

The study described here investigated the relationship between indicated torque and resulting bolt load for a typical RAM package closure under normal operational conditions and compared torque wrench tensioning to the “turn of the nut” method.

The closure assembly used for this test consisted of the lid from a 9970 packaging, mated with a blind flange. The containment vessel for the 9970 packaging consists of a section of Schedule 40 pipe with a standard pipe cap for the bottom closure and standard ASME flange assembly for the closure. A standard cap is welded to the top flange to complete the lid. For the present test, a 9970 containment vessel closure and blind flange, secured with eight 5/8-11 UNC-2A bolts were employed. The bolts were ASTM A-193, Grade B8, Class 2, having a yield strength of 100 ksi and A-194 Grade 8C nuts. Standard washers were employed between the bolt head flange and between the nut and flange. Round head brass tacks were installed in each of the bolt heads to provide a consistent

reference point for measurement of bolt elongation. Likewise, the screw end of each of the bolts was smoothed and filed to provide a consistent measurement point.

The preload was determined using a micrometer able to measure to 1/10,000 in. This is typical of instrumentation of the sort commonly available under operational conditions. The repeatability of measurements made with the micrometer was determined from the repeated measurement of the bolt initial length. This study indicated that the uncertainty in the bolt length measured in this way was approximately 10% of the maximum elongation. This is sufficient for the method to be a valid means of evaluating the preload in the closure bolts and confirming the intended preload is achieved, within the uncertainty interval of the method of establishing the preload. These results are shown in Table 1.

The closure was assembled, following typical practice for securing the lid on a radioactive materials package with a flange type closure. The bolts were tightened in a star pattern in stages of 102, 170, and 204 Nm (75, 125 and 150 ft lb). Following the A final circumferential pass was made at 204 Nm (150 ft lb). The initial bolt length was measured and the bolt length measured after the bolt was torqued (i.e., after the preload was established). For the initial trials, the bolt length was measured at the end of each pass.

Tests were performed for un-lubricated bolts and for lubricated bolts. The lubricant employed was lithium-molybdenum EP grease, Valvoline NLGI #2 GC-LB, manufactured by Ashland Oil Co. For the lubricated cases, the bolts and nuts were re-lubricated before each trial.

For the Turn-of-the-Nut tests, the rotation of the nut which would yield the target bolt load was determined. For a soft joint, tightening compresses the structure (e.g., flange and gasket) rather than only producing elongation of the bolt. For this case, the joint is stiff, relative to the bolt, and there is no gasket between the flanges. The modulus of elasticity is the same for flange and bolt, but the flange cross sectional area is much larger. For this test, it was found that turning the nut 90° from the hand tight position yielded the elongation corresponding to the design preload of about 15000 lb (which corresponds to the average bolt load achieved for the lubricated case). The bolts were lubricated for this test, in the same way as for the lubricated case using the torque wrench.

RESULTS

In the unlubricated tests, the average preload in the bolts (which were tightened to 150 ft lb) was 9,000 lb, with a typical standard deviation of 2055 lb. The preload varied from bolt to bolt and trial to trial, with min preload of 868 lb and maximum of 23490 lb, over all trials. The average bolt load for each trial varied from 10854 lb to 6350lb. The average for a given bolt across all trials varied from 11594 lb to 6653 lb. Because of questions about the data, results for trials 2 & 3 were not included in these averages. These results are shown in Table 2 and Figures 1. Figure 2 shows the load in each bolt following each pass in the tightening process.

In the lubricated tests, the average preload in the bolts (which were tightened to 150 ft lb) was 15,500 lb, with a typical standard deviation of 1640 lb. The preload varied from bolt to bolt and trial to trial, with min preload of 7646 lb and maximum of 26,407 lb over all trials. The average bolt load for each trial varied from 17,400 lb to 13,100 lb. The average for a given bolt across all trials varied from 18,656 lb to 13,349 lb. These results are shown in Table 3 and Figures 31. Figure 4 shows the load in each bolt following each pass in the tightening process for the lubricated case.

In the lubricated tests, tightened by “Turn of the Nut”, the average preload in the bolts was 14,165 lb, with a typical standard deviation of 835 lb. The preload varied from bolt to bolt and trial to trial, with min preload of 7100 lb and maximum of 26,837 lb, over all trials. The average bolt load for each trial varied from 17637 lb to 11,738 lb. The average for a given bolt across all trials varied from 15171 lb to 12,736 lb. These results are shown in Table 4 and Figures 5.

DISCUSSION

In this test, the preload achieved for the 150 ft lb torque applied was about 10,000 lb for the un-lubricated case and 15000 lb for the lubricated case. The corresponding values of coefficient of friction determined from the design equations are approximately 0.15 for the un-lubricated case and 0.10 for the lubricated case. These are noticeably less than the values expected from references on coefficient of friction. The corresponding values of Nut Factor are 0.19 for the lubricated case and 0.28 for the un-lubricated case. These agree with the range of values, 0.2 to 0.25, noted above.

The load corresponding to the 100 ksi yield strength for the bolt material is 22600 lb. Although the average loads are well within this value, the highest single value from the data set (i.e., peak load) determined for each case, un-lubricated, lubricated and turn of nut) exceeded this value. The lack of any change in initial length for each trial shows conclusively that no yielding occurred. So, either the actual yield strength of the material exceeded the nominal value, or the extension was less than indicated. Using the average initial length for the bolt in question to calculate the load, instead of the value measured for the trial in question, reduced the load by about 10 %. Likewise, evaluating the load at the lower end of the tolerance band for the measured value yielded a similar reduction.

The un-lubricated case showed, as expected, consistently lower load for given torque than lubricated case. For the un-lubricated case there was more variation in load for given torque. The results also show significant variation both from trial to trial, for a given bolt, and from bolt to bolt, for a given trial.

The lubricated case consistently showed significantly greater bolt preload for same torque, than the un-lubricated case. The results for the lubricated case were uniformly more consistent than for un-lubricated case, for both variation from, trial to trial for given bolt and from bolt to bolt for a given trial.

Although the results for lubricated threads display higher bolt load for the specified torque, and are more consistent, it is recognized that, for radiological reasons, there are circumstances when lubricant cannot be used for radioactive materials packagings.

For the Turn-of-the-Nut case the bolt preload was significantly greater and results more consistent than for un-lubricated case. This was true both for variation from trial to trial, for given bolt, and for variation from bolt to bolt, for a given trial. Compared with the lubricated case tightened by torque wrench, the average load obtained by Turn-of-the-Nut was a bit less (though this could be corrected by a small increase in the turn of nut), but the variation from trial to trial was less.

The benefit of the Turn-of-the-Nut technique was that it yielded more consistent results. The test showed the importance of initial condition (starting torque) for this technique. Determining rotation required is not as simple as calculating the rotation to correspond to the required bolt strain, because the joint is elastic. The amount of rotation used here (90°) was determined by several trials, then applied consistently (to all bolts). Determining the correct rotation is achieved requires marking nut prior to turning. The bolt must be carefully held to prevent rotation for proper results to be obtained.

The initial condition is critical to obtaining consistent results using the Turn-of-the-Nut method. If the initial tightening does not take up all the slack motion in the assembly, Turn-of-the-Nut will not produce adequate preload. Conversely, if the initial position is too tight, the load will be greater than intended. For this test, the nuts were tightened as much as possible by hand, without tools. The bolt assembly was moved slightly while tightening to insure that all potential gaps were closed. The nuts were then tightened to 5 ft lb. This was found to correspond to the point in the initial tightening, where noticeable resistance is detected. When all bolts were at this initial condition the 90° rotation was applied.

Manual assembly results in intermittent turning of nut. Since the static friction must be overcome each time the nut is turned, this contributes to the variation in torque/preload results, from case to case and bolt to bolt. Continuous, mechanical drive of nuts would yield more consistent results [4], but has been observed to result in galling and seizing of nuts to bolts in some cases, where manual assembly avoided these problems. Both the lower relative velocity of the surfaces (lower rate of work) and the intermittent motion allow dissipation of heat generated and effectively reduce the likelihood of galling.

CONCLUSIONS

Direct measurement of bolt length can confirm preload, This is the case even when using ordinary micrometer for measurement.

Tightening bolts with a torque wrench is adequate method to establish bolt preload, but variation is large, as is noted in the literature. For this reason, calibration of torque wrenches to tight tolerance is not justified. Good technique is important to obtaining consistent results. Lubrication provides great benefits in both consistency of results and preload obtained for a given torque.

The Turn-of-the-Nut is somewhat more consistent than using a torque wrench, but starting condition is important to obtaining good results. The 5 ft lb initial condition employed in these tests was satisfactory. In concept, turn of nut is independent of lubrication, but in practice achievement of a consistent rotation is greatly facilitated by lubricant.

CONTRACT NUMBER

This document was prepared in conjunction with work accomplished under Contract No. DE-AC09-08SR22470 with the U.S. Department of Energy.

DISCLAIMER

The United States Government retains, and by accepting the article for publication, the publisher acknowledges that the United States Government retains, a non-exclusive, paid-up, irrevocable, worldwide license to publish or reproduce the published form of this work, or allow others to do so, for United States Government purposes.

This work was prepared under an agreement with and funded by the U.S. Government. Neither the U. S. Government or its employees, nor any of its contractors, subcontractors or their employees, makes any express or implied: 1. warranty or assumes any legal liability for the accuracy, completeness, or for the use or results of such use of any information, product, or process disclosed; or 2. representation that such use or results of such use would not infringe privately owned rights; or 3. endorsement or recommendation of any specifically identified commercial product, process, or service. Any views and opinions of authors expressed in this work do not necessarily state or reflect those of the United States Government, or its contractors, or subcontractors.

REFERENCES

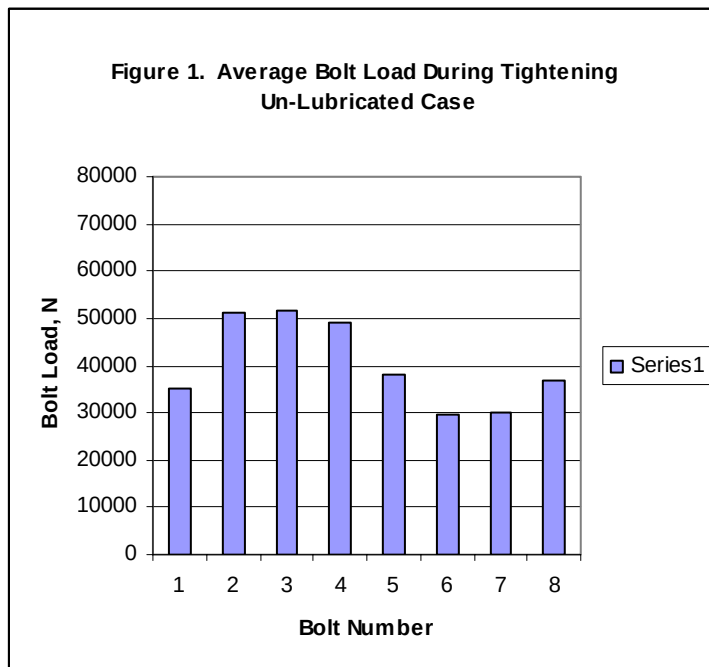
1. Brown, W., Marchand, L., Evard, A., and Reeves, D., "The Effect of Bolt Size on the Assembly Nut Factor," PVP2007-26644, Proceedings of the 2007 ASME Pressure Vessels and Piping Conference, San Antonio, Texas, July, 2007.
2. Blake, J. C., and Kurtz, H. J., "The Uncertainties of Measuring Fastener Preload", Machine Design, September, 1965.
3. Spotts, M. F., Design of Machine Elements, 6th Edition, Prentice Hall, 1985.
4. Ganeshmurthy, S., and Nassar, S. A., "Study of Two-Stage Tightening of Threaded Fasteners Using Various Tool Speed Combinations", PVP2007-26719, Proceedings of the 2007 ASME Pressure Vessels and Piping Conference, San Antonio Texas, July, 2007.
5. Title 10, United States Code of Federal Regulations, Part 71, Packaging and Transportation of Radioactive Material (10 CFR 71), 2005.
6. Wu, T.-t., "Stress Analysis of Closure Bolts in Shipping Packages of Radioactive Materials," PVP2009, 77428, Proceedings of the 2009 ASME Pressure Vessels and Piping Conference, Prague, Czech Republic, July, 2009.

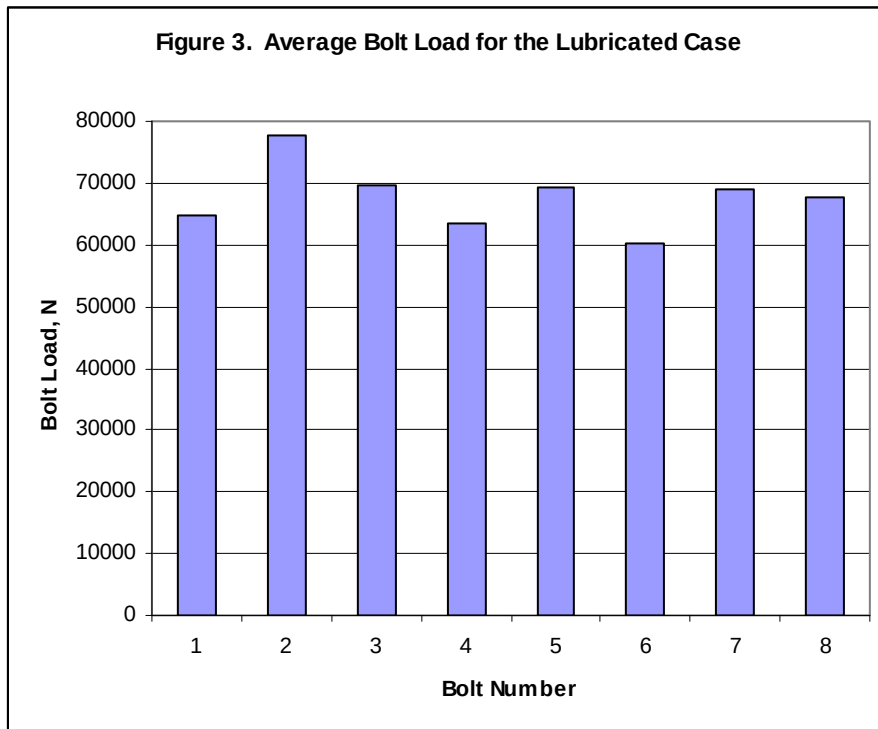
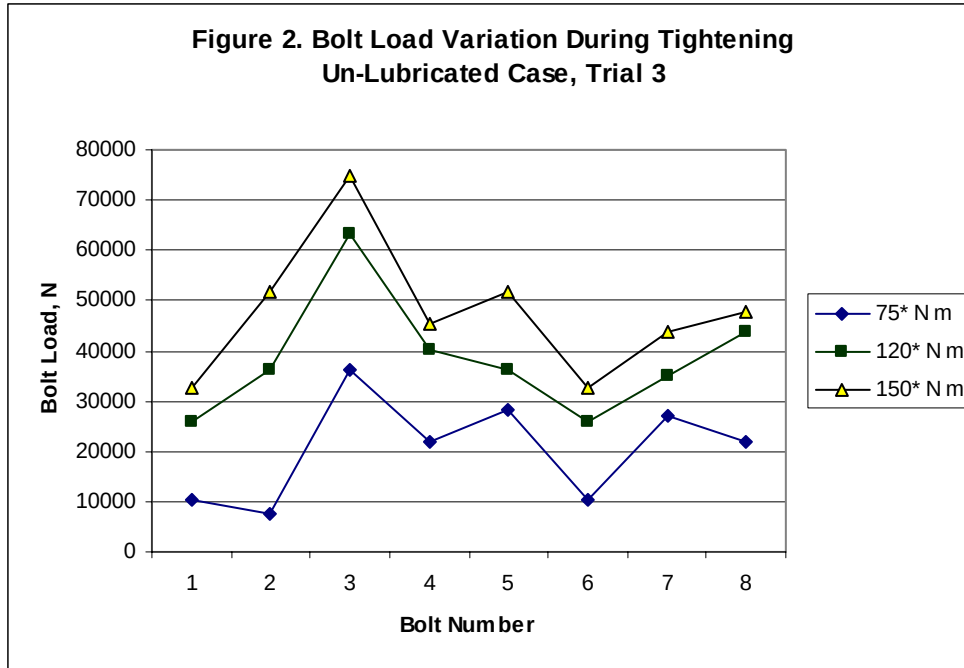
7. Nagata, S., Sawa, T., and Hamamoto, S., “Effects of Scatter in Bolt Preload on the Sealing Performance in Bolted Flange Connections with Cover of Pressure Vessel Under Internal Pressure,” PVP2006-ICPVT-11-93538, Proceedings of the 2006 ASME Pressure Vessels and Piping Conference, Vancouver, Canada, July, 2006.

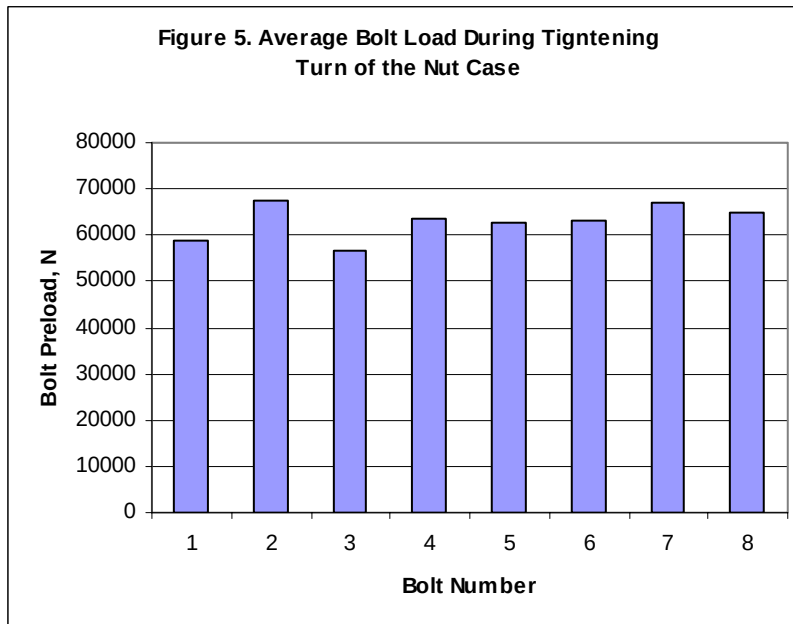
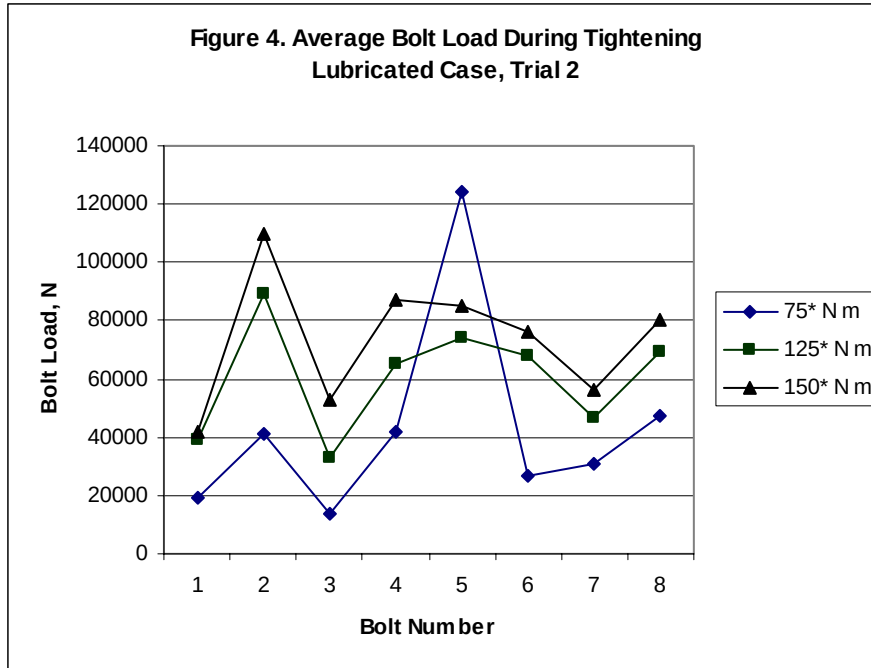
8. Brown, W., “Flange Assembly Bolt Load Selection Based on Leak Before Break Analysis”, PVP2006-ICPVT-11-93075, Proceedings of the 2006 ASME Pressure Vessels and Piping Conference, Vancouver, Canada, July, 2006.

9. Nassar, S. A., and Yang, X., “Novel Formulation of the Tightening and Breakaway Torque Components in Threaded Fasteners”, PVP2006-ICPVT-11-93175, Proceedings of the 2006 ASME Pressure Vessels and Piping Conference, Vancouver, Canada, July, 2006.

10. Nassar, S. A., and Veeram, A. B., “Ultrasonic Control of Fastener Tightening Using Varying Wave Speed”, PVP2005-71578, Proceedings of the 2005 ASME Pressure Vessels and Piping Conference, Denver, Colorado, July, 2005.







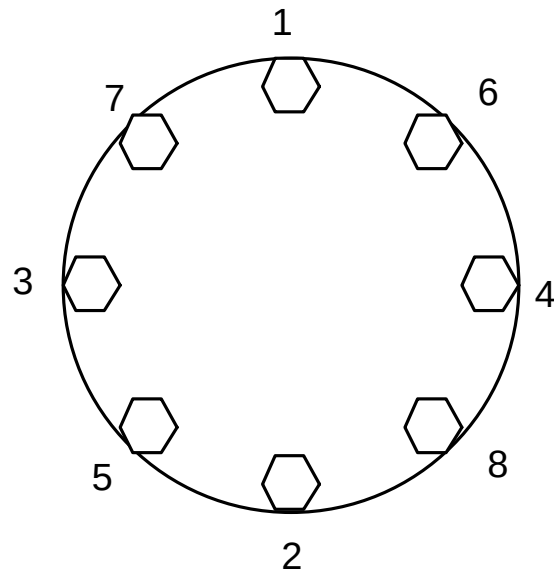


Figure 6. Bolt Numbers and order of tightening for Test

Table 1. Initial Lengths of Bolts

Bolt	Trial 1	2	3	4	5	6	7	8	9	10	Sum	Avg	Std Dev
1	3.1773	3.1772	3.1777	3.1775	3.1774	3.1776	3.1781	3.1780	3.1778	3.1786	31.7772	3.1777	0.0004
2	3.1891	3.1884	3.1903	3.1862	3.1896	3.1902	3.1886	3.1902	3.1905	3.1909	31.8940	3.1894	0.0014
3	3.1925	3.1928	3.1895	3.1937	3.1938	3.1929	3.1936	3.1935	3.1943	3.1938	31.9304	3.1930	0.0014
4	3.1893	3.1888	3.1898	3.1893	3.1901	3.1896	3.1902	3.1902	3.1898	3.1903	31.8973	3.1897	0.0005
5	3.1941	3.1944	3.1955	3.1947	3.1953	3.1944	3.1951	3.1950	3.1953	3.1955	31.9493	3.1949	0.0005
6	3.2008	3.2026	3.2030	3.2013	3.2018	3.2023	3.2026	3.2021	3.2026	3.2025	32.0216	3.2022	0.0007
7	3.1940	3.1936	3.1936	3.1930	3.1937	3.1942	3.1947	3.1948	3.1952	3.1952	31.9420	3.1942	0.0008
8	3.1887	3.1892	3.1901	3.1901	3.1908	3.1899	3.1908	3.1901	3.1906	3.1907	31.9010	3.1901	0.0007
Sum	25.5258	25.5270	25.5295	25.5258	25.5325	25.5311	25.5337	25.5339	25.5361	25.5375	255.3127	25.5313	0.0042
Avg	3.1907	3.1909	3.1912	3.1907	3.1916	3.1914	3.1917	3.1917	3.1920	3.1922	31.9141	3.1914	0.0005
Std Dev	0.0067	0.0072	0.0071	0.0070	0.0069	0.0069	0.0070	0.0068	0.0071	0.0068	0.0691	0.0069	

Table 2. Bolt Load for Un-Lubricated Case.
Data for trials 2 & 3 omitted because of questionable high values.

	Bolt Load										
Bolt	Trial 1	4	5	6	7	8	9	10	Sum	Avg	Std Dev
1	15749	7291	8166	5833	3500	9041	6125	7291	62997	7875	3596
2	10440	11600	23490	8410	11310	8410	8990	9570	92221	11528	4986
3	9565	16812	10725	4058	6377	16232	17101	11884	92754	11594	4899
4	15709	10181	21527	7854	9018	6400	9891	7272	87852	10981	5124
5	12450	11581	869	4922	7817	12739	8976	9265	68619	8577	4059
6	4083	7292	8459	5542	5834	3792	13563	4667	53231	6654	3208
7	8117	9856	5798	7247	2319	6957	7537	6088	53920	6740	2185
8	9823	10689	7800	6934	7512	7223	9245	7223	66448	8306	1413
Sum	85936	85304	86833	50800	53686	70794	81427	63260	578041	72255	
Avg	10742	10663	10854	6350	6711	8849	10178	7908	72255	9032	
Std Dev	3895	3003	7757	1505	2899	3922	3517	2252	16444	2056	

Table 3. Bolt Load for Lubricated Case													
	Bolt Load												
Bolt	Trial 1	2	3	4	5	6	7	8	9	10	Sum	Avg	Std Dev
1	10978	9331	14271	10154	21406	15369	18387	20171	12350	13173	145590	14559	4218
2	18621	24646	17526	23277	13692	14514	24646	12323	11775	13555	174576	17458	5113
3	9077	11828	26407	14304	11553	18017	17055	18705	16504	12928	156379	15638	4912
4	10724	19524	9349	18699	18424	12924	5637	14024	15399	17874	142579	14258	4618
5	14224	19148	16412	14361	10942	16960	15865	18874	15318	13267	155372	15537	2506
6	9130	17153	5810	13280	14387	11897	16600	18537	16600	11897	135290	13529	3974
7	13458	12634	12084	22109	13046	17577	14831	19500	14556	15380	155174	15517	3256
8	18296	18023	17750	16658	9694	15565	15292	17204	15565	7646	151695	15170	3618
Sum	104509	132287	119610	132841	113144	122823	128314	139338	118069	105720	1216655	121665	11709
Avg	13064	16536	14951	16605	14143	15353	16039	17417	14759	13215	152082	15208	1464
Std Dev	3798	4977	6227	4507	3951	2179	5226	2789	1795	2912	11707	1171	

Table 4. Bolt Load for Turn of the Nut Case													
	Bolt Load										Sum	Avg	Std Dev
Bolt	Trial 1	2	3	4	5	6	7	8	9	10	Sum	Avg	Std Dev
1	17564	10566	13173	13722	11664	15917	14682	14545	7684	12761	132279	13228	2801
2	12597	13966	13966	14514	19169	9858	26837	12871	14240	13692	151710	15171	4698
3	15129	16504	12103	8527	14029	9628	14579	14579	8802	13479	127359	12736	2832
4	15674	22549	19249	7975	8937	10449	20074	15399	12374	10449	143129	14313	5067
5	12856	7112	18601	21336	17233	11215	13951	8480	11489	18874	141147	14115	4742
6	12312	13557	15217	11343	10513	11620	16600	19090	13972	17292	141515	14152	2842
7	10162	11398	17028	16479	15105	15929	17577	18538	17165	10986	150368	15037	3046
8	15019	18296	13244	13927	16385	9285	16794	14610	11333	16794	145688	14569	2746
Sum	111313	113948	122581	107823	113035	93902	141094	118112	97059	114328	1133195	113320	13154
Avg	13914	14243	15323	13478	14129	11738	17637	14764	12132	14291	141649	14165	1644
Std Dev	2349	4834	2679	4333	3529	2699	4209	3307	3051	3055	8350	835	