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ASSEMBLY PERFORMANCE OF A GASKETED BOLTED FLANGED PIPE JOINT USING DIFFERENT BOLT TIGHTENING STRATEGIES[*]

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Abstract– This paper presents the results of the assembly process of a gasketed bolted flanged pipe joint for two different bolt tightening strategies, i.e. ASME and Industrial using torque control of preload method. The final clamping force is achieved in four and five passes in ASME and Industrial strategy respectively. Axial bolt stress variation, at the end of each pass, individual bolt bending behavior, gasket stress and flange stress variations for both strategies are discussed.

Keywords– Pipe flange joint, assembly process, torque control, clamping force, finite element analysis

1. INTRODUCTION

Bolted flanged joints are used to join pipes to pipes or pipes to equipment. It is necessary to create a proper preload in the bolts of the joint to operate safely and reliably. There are many variables which affect the assembly process so it is difficult to predict and achieve a given amount of preload [1-26]. In industry, the most widely used assembly method is torque control using torque wrenches. In this method, bolt is stretched by turning the bolt or nut against flange surfaces. Bolts are tightened individually in a defined tightening sequence. Due to the elastic interactions of flange, gasket and other components, bolt preload scatter (different bolt preload values in all bolts), bolt bending behavior and gasket crushing is observed. Also, any excessive preload can crush the gasket and its recovery may not be possible. Depending on the size, type and application of the gasket, manufacturer usually provides the upper limit of gasket contact stress. This paper provides the comparison between the ASME [27] and Industrial [28] strategies for 8 inch flange size of Class 900# bolted flanged joint using nonlinear finite element analysis.

2. MODELING AND ANALYSIS

As gasketed bolted flanged pipe joints has both rotational and reflective symmetry, only one pipe, flange and half of the gasket is modeled. Flange, bolt and gasket dimensions are taken as per ANSI B16.5 [29]. SOLID45 are used for bolts and flange, interface (INTER195) elements are used for gasket and contact elements, CONTA171 and CONTA174 are used for contact between different surfaces, i.e. under bolt head and flange top surface, flange bottom surface and gasket top surface in ANSYS software [30]. Figure 1a shows meshed flange, bolt and gasket. For pipe, bolts and flange elasto-plastic material model is used and material properties are given in Table 1 [31]. Bilinear kinematic hardening for the elasto-plastic material properties is used during the analysis. A bilinear material model consists of two sections, each

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having a linear gradient. For the first section, an elastic material is used which is valid until the yield stress and the gradient of this section is the Young's Modulus of Elasticity. The second section functions beyond the yield stress, and gradient (plastic modulus) is 10% of the Young's Modulus of Elasticity [14]. For spiral wound gasket material modeling, simplified approach developed by Takaki *et al* [28] is used. Being a bolted system and not a fixed system, in axial and radial directions, gasket and flange are free to move providing rotation of flange due to relaxation of bolts and joints. This results in stress variations in flange, gasket and bolts. At the gasket lower portion, symmetry conditions are applied. Torque increments and incremental target stress values for each pass for the required preload are given in Table 2. In order to initiate contact and to create the desired preload, an axial displacement is applied at the bottom of the bolts in the downward direction. Structural boundary conditions are shown in Fig. 1b.

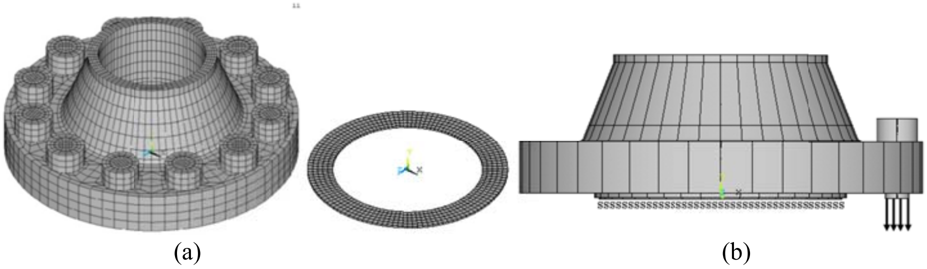

(a) (b)

Fig. 1. (a) Meshing of flange, bolt and gasket, (b) Applied boundary conditions

Table 1. Material properties

Part	As per standard	Modulus of Elasticity - E (MPa)	Poisson Ratio (υ)	Allowable Stress (MPa)
Flange/Pipe	ASTM A350 LF2	173058	0.3	248.2
Bolt	ASTM SA193 B7	168922	0.3	723.9

3. BOLT TIGHTENING SEQUENCES

The bolt tightening is performed according to two strategies ASME and Industrial. According to ASME PCC-1 guidelines [27], bolt tightening is performed in four passes, and as per following two sequences;

- Sequence-1: 1, 7, 4, 10, 2, 8, 5, 11, 3, 9, 6, 12 (for first three passes)
- Sequence-2: 1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12 (for the last pass)

As per industrial guidelines [28], bolt tightening is performed in five passes, and as per the following two sequences;

- Sequence-1: 1, 7, 4, 10, 2, 8, 5, 11, 3, 9, 6, 12 (for first four passes)
- Sequence-2: 1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12 (for the last pass)

For preload calculation different variables and factors are considered. Equation (1) defines the relationship between applied torque and preload achieved as:

$$T = F\,(KD) \tag{1}$$

where, T = Input Torque (Nm), F = Achieved Preload (N), D = Nominal Diameter of Bolt (m), K = Nut Factors.

For each pass target torque is converted into bolt preload and then average bolt stress is calculated dividing bolt preload by nominal cross sectional area of the bolt shank. For both strategies incremental torque and target stress values are given in Table 2 up to the maximum target torque of 1355 Nm. The target stress in each of the bolts is achieved by applying a displacement (UY) on the bottom areas of the bolts during finite element analysis. The value of the displacement (UY) is obtained from the average axial stress in the bolt shank.

In order to determine bolt relaxation or bolt bending behaviour during bolt tightening as per sequence-1 and sequence-2, four nodes are selected on the shank of each bolt at an angle of 90 degrees. Inner and outer nodes are represented by B1/1 and B1/2, while side nodes are represented by B1/3 and B1/4, and B1/M represents the mid node at the bolt shank. For all other bolts similar nomenclature is used. The mid node on the shank of bolt is selected for the axial bolt stresses. The magnitude of axial displacement applied at the bottom area of the bolt shank to pre-stress each bolt to the target stress value is given in Table 3 for both ASME and Industrial strategies.

Table 2. Torque Increments and Incremental target stress values for each pass.

Tightening Pass	Loading (ASME Strategy)	Pre stress (MPa)	Loading (Industrial Strategy)	Pre stress (MPa)
Pass 1	Tighten to 20% to 30% of Target Torque.	61	Tighten to 15% to 25% of Target Torque.	40
Pass 2	Tighten to 50% to 70% of Target Torque.	132	Tighten to 40% to 50% of Target Torque.	90
Pass 3	Tighten to 100% of Target Torque	202	Tighten to 70% to 80% of Target Torque.	150
Pass 4	Clockwise pattern the same Target Torque value of 100%.	202	Tighten to 100% of Target Torque	202
Pass 5	--------	--	Clockwise pattern of the same Target Torque value of 100%.	202

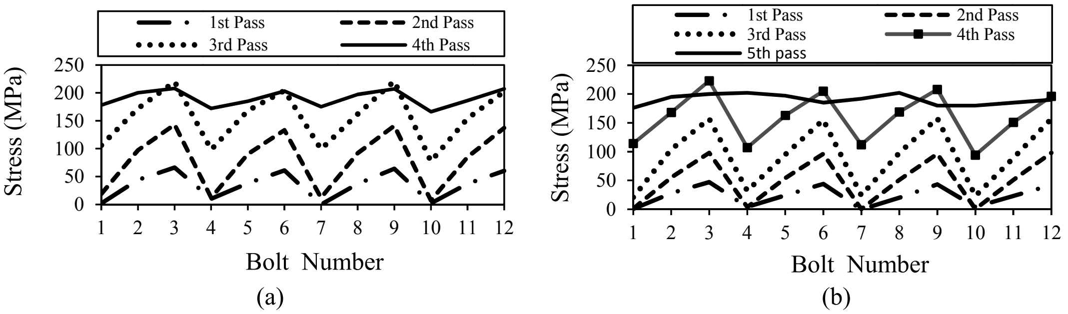
Table 3. Magnitude of UY for each pass

Bolt #	ASME Strategy				Industrial Strategy				
	UY (mm)				UY (mm)				
	Pass 1	Pass 2	Pass 3	Pass 4	Pass 1	Pass 2	Pass 3	Pass 4	Pass 5
Bolt 1	0.153	0.191	0.257	0.129	0.090	0.119	0.172	0.201	0.091
Bolt 7	0.112	0.205	0.263	0.141	0.060	0.129	0.180	0.194	0.106
Bolt 4	0.176	0.167	0.245	0.128	0.106	0.117	0.170	0.177	0.129
Bolt 10	0.118	0.178	0.252	0.150	0.079	0.110	0.174	0.180	0.105
Bolt 2	0.207	0.225	0.242	0.082	0.123	0.137	0.189	0.181	0.066
Bolt 8	0.154	0.236	0.255	0.097	0.087	0.145	0.190	0.184	0.066
Bolt 5	0.190	0.214	0.231	0.089	0.114	0.141	0.179	0.167	0.084
Bolt 11	0.160	0.213	0.244	0.100	0.098	0.139	0.179	0.171	0.070
Bolt 3	0.234	0.242	0.240	0.044	0.141	0.155	0.195	0.175	0.038
Bolt 9	0.178	0.252	0.264	0.051	0.105	0.160	0.199	0.167	0.020
Bolt 6	0.196	0.240	0.239	0.068	0.119	0.158	0.194	0.159	0.038
Bolt 12	0.199	0.242	0.222	0.066	0.118	0.160	0.196	0.157	0.038

4. RESULTS AND DISCUSSIONS

a) Bolt preload scatter

Figure 2 shows bolt stress variations after the completion of individual pass. Stress variations at target stresses are observed considerable in the first three and four passes tightened as per sequence 1for ASME and Industrial strategy respectively. Gradual stress increase in the first three and four passes for ASME and industrial strategies, respectively, is observed. The difference between the maximum and minimum stress values during the first three passes for ASME strategy is 65 to 145 MPa while for the Industrial strategy the difference of stresses between the first four passes varies between 45 to 115 MPa. In the last pass as per sequence 2, bolt stress variations reduce 42 MPa for ASME and 22 MPa for Industrial strategy. Hence the 4th pass with 100% of target torque tightened as per sequence 2 is concluded to be very important to have a pronounced effect in reducing the stress variations.

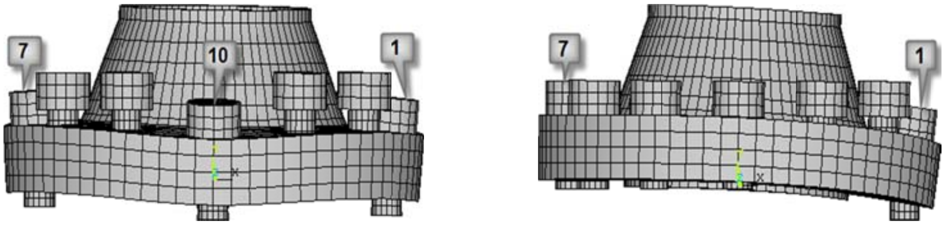


(a) (b)

Fig. 2. Axial bolt stress variation after each pass for; (a) ASME strategy, (b) Industrial strategy

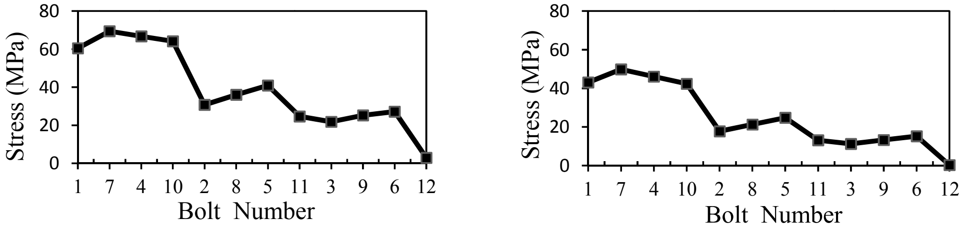
b) Bolt relaxation behavior

Figure 3 shows bolt relaxation behavior and effect of elastic interaction on the neighboring bolts during tightening of first four bolts 1, 4, 7 and 10. Bolts also experience an increase in the bolt stress as bolts on the opposite side are tightened such as bolt 7 during tightening of bolt-1. Each time a bolt is tightened, stresses in all of the other bolts are observed varying. Stresses in bolts may increase or decrease when other bolts are tightened depending upon the relative position of the bolt tightened. Figure 4 shows stress variations in the first bolt during tightening of all other bolts in first pass as per specified sequence. During tightening of first bolt itself a, a required stress of 60 and 43MPa is achieved for ASME and Industrial strategies respectively. Its value becomes maximum, i.e. 70 and 50MPa while tightening the 7th bolt which is at 180 degrees, while it reduces to almost zero when 12th bolt is tightened. Almost similar bolt stress variation behavior is observed for both the ASME and Industrial strategies, whereas difference in values for both strategies is observed due to the difference in the specified target torque value.



(a) (b)

Fig. 3. Exaggerated deformations of flange and bolt relaxation as individual bolts are tightened



(a) (b)

Fig. 4. Stress variation in bolt-1 while tightening other bolts during pass-1 for
(a) ASME strategy, (b) Industrial strategy

c) Bolt bending behavior

Due to the bending of the bolts, joint relaxation and the bolt scatter results, concluding dynamic-mode-of-load is determined as the main reason for joint relaxation and behavior [14, 22-26]. In order to study bending behavior of the bolts, four nodes are selected on each bolt at 90 degrees. The bending behavior of the bolts is shown in Fig. 5 whereas stress variation is plotted in Fig. 6a and Fig. 6b for both strategies.

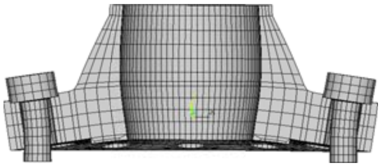
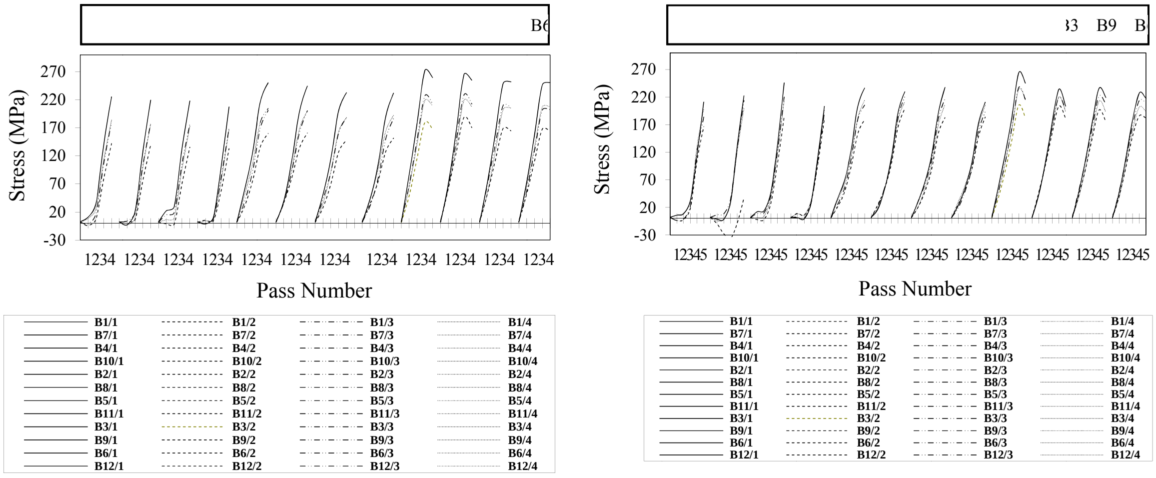


Fig. 5. Bolt bending behavior during assembly process

The bending behavior of each bolt is observed different in the joint. It is observed that bolt 1, 7, 4, 10, 2, 8, 5 and 11 show an increasing trend in all the passes for both strategies. Compressive stresses in bolt 1, 7, 4, and 10 are observed diminishing after the 2nd and 3rd pass. Bolts 3, 9, 6 and 12 show an increasing trend up to the 3rd and 4th pass for both strategies which decreases for the last pass. On every bolt maximum (tensile) and minimum (compressive) stress is observed at the inner and outer nodes, respectively, concluding bolt bending. From analysis with ASME strategy (4 passes), node B3/1 shows maximum stress of 271MPa which reduces to 259MPa in the last pass (Fig. 6a). Whereas for analysis with Industrial strategy, the same bolt shows maximum stress of 263MPa which reduces to 245MPa in the last pass (Fig. 6b). Bending of bolts concludes their effect in joint assembly process, as a major portion of the preload is consumed in bending the bolts and hence the effective preload is observed less than the anticipated preload.

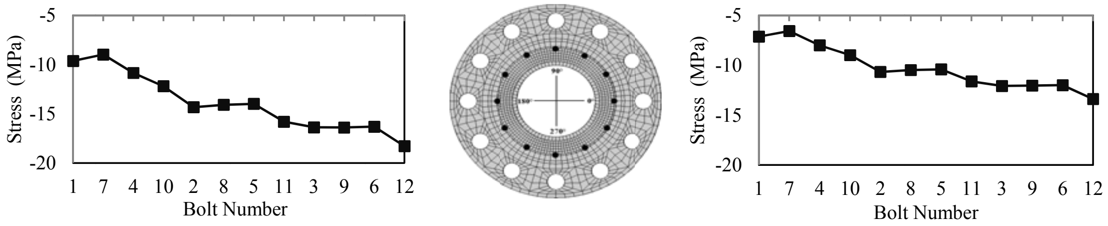
d) Gasket stress variation

Gasket sealing is investigated by the distribution of the stress on the gasket sealing area. On outer diameter of the gasket, nodes are selected along the locations of the bolts and the contact stresses are observed after each pass. Figure 7a-c shows the gasket contact stress variation along first bolt during first pass. It is observed that every time a bolt is tightened, contact stresses become almost double at the end of the first pass for both strategies. Contact stresses along all bolts after the completion of each pass are plotted in Fig. 8. Gasket contact stresses are almost uniform in the first two passes with average values of 20MPa and 50MPa for ASME guidelines, while for Industrial guideline the gasket contact stresses are almost uniform in the first three passes with average values of 10MPa, 30MPa and 50MPa. The last two passes, however, show considerable variations in stress in both strategies. During the second last pass a maximum of 118MPa and 107MPa is observed in ASME and Industrial Strategies with average stress of 109MPa and 102MPa respectively, while in the last pass a maximum stress of134MPa and 135MPa with average stress of 131MPa, and 132MPa are observed for ASME and Industrial strategies, respectively. Instead of these variations the average gasket stress for both the strategies lies well within the specified stress limit of 202MPa [32]. A more uniform gasket stress distribution is observed in Industrial strategy as compared to the ASME strategy.



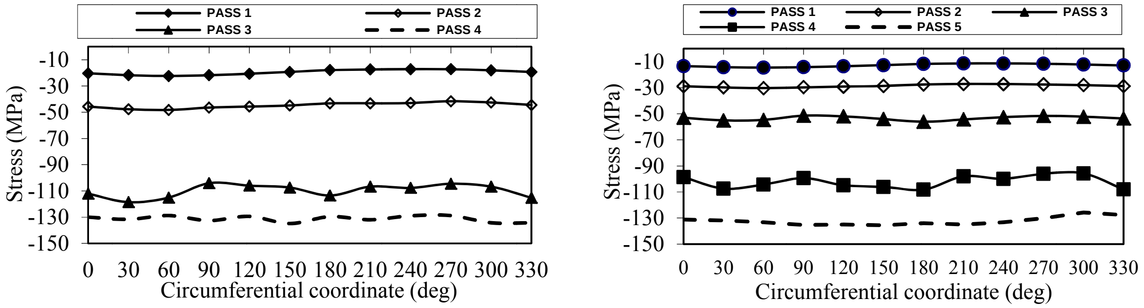
(a) (b)

Fig. 6. Individual bolt bending behavior for (a) ASME strategy, (b) Industrial strategy



(a) (b) (c)

Fig. 7. Gasket stress variations along first bolt location during first pass: (a) ASME strategy, (b) Location of nodes selected for gasket contact stress, (c) Industrial

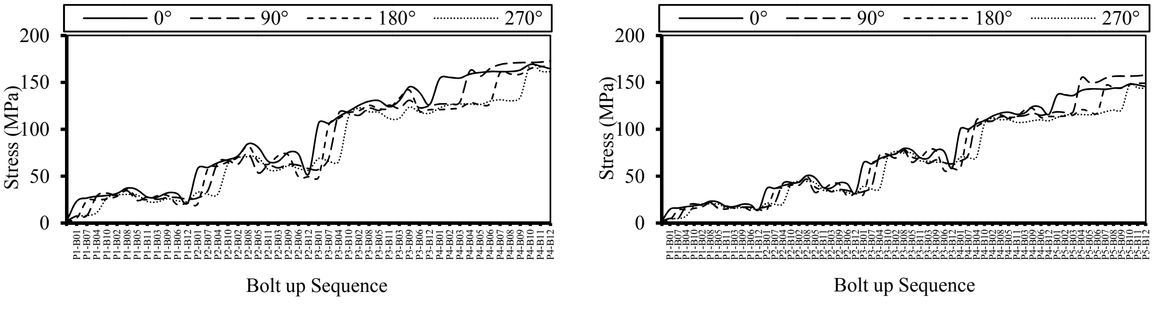


(a) (b)

Fig. 8. Gasket contact stress distribution along the outer diameter of spiral wound gasket for: (a) ASME strategy, (b) Industrial strategy

e) Flange stress variation

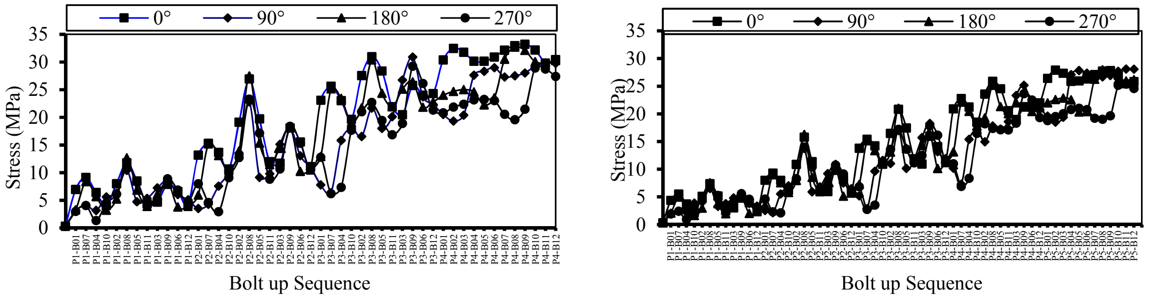
Figure 9a-b shows maximum principal stress variations at hub flange fillet (HF). Stress is maximum at a location close to the bolt being tightened and is minimum at a location at 90 degrees. Stress along 0 degree and 180 degrees is observed more than along 90 degrees and 270 degrees. For sequence-1, almost the same stress variation pattern but with higher magnitude of variation with each pass is observed. Higher stresses are observed at the last pass but with more uniform stress than the first three and four passes for ASME and Industrial strategies respectively. However stress is more uniform with less variation in industrial strategy compared to the ASME strategy. Overall, maximum stress observed is less than the yield strength of the material.



(a) (b)

Fig. 9. Maximum principle axial stress variation at hub flange fillet during bolt up for (a) ASME strategy (b) Industrial strategy

Figure 10 represents the principle axial stress variations at location of hub pipe. During tightening of the bolts considerable variations in stresses are observed around the flange but the magnitude of stress does not exceed 35 MPa in both strategies. After completion of last pass stress distribution is uniform around the flange and maximum stress observed is less than the yield strength of flange material which is 250 MPa.



(a) (b)

Fig. 10. Maximum principle axial stress variation at hub pipe fillet during bolt up for
(a) ASME strategy (b) Industrial strategy

5. CONCLUSION

- In torque control method, bolts are tightened individually, which is concluded to be the main reason for non-uniform stresses in the bolts. The maximum preload reduction is observed in the first bolt tightened and the least in the last bolt. It is concluded that the bolt scatter cannot be eliminated using torque control method, but can be reduced within acceptable level by proper bolt up sequence and multiple pass tightening.
- The variations in the stresses in the gasket are directly related to bolt pre load scatter in the torque control method. However, maximum stress in the gasket is less than the crushing limit of 206MPa.
- The bending behavior of each bolt is observed different in the torque control method.
- The strength and the sealing performance of a joint depend upon the preload applied, the sequence of tightening and the number of passes selected.
- The variations of stresses within a pass and in between the passes are observed greater for ASME Strategy than the Industrial Strategy.
- Smoother stress variations could be achieved by increasing the number of passes and decreasing the torque increments in between the passes
- Industrial strategy is concluded to be better than the ASME for less stress variations, better gasket sealing, and for good overall behavior of the joint. However, this method requires more time to increase number of passes.

REFERENCES

1. Thompson, G. (1998). *An engineers guide to pipe joints*. Professional Engineering Publishing, ISBN 1-86058-081-5.
2. Bickford, J. H. & Nassar, S. (1998). *Handbook of bolts and bolted joints*. CRC Press, ISBN 0-8247-9977-1.
3. Bickford, J. H. (2008). *Introduction to the design and behavior of bolted joints*. CRC Press, ISBN 0-8493-81762.
4. Sawa, T., Higurashi & Akagawa, H. (1991). A stress analysis of pipe flange connections. *Journal of Pressure Vessel Technology*, Vol. 113. pp.497-503.
5. Nagata, S., Shoji, Y. and Sawa, T. (2002). A simplified modeling of gasket stress-strain curve for FEM analysisin bolted flange joint design. *ASME Internationa PVP Conference*. Vol. 433, pp. 53-58.
6. Sawa, T., Matsumoto, M. and Nagat, S. (2003). Effects of scatter in bolt preload of pipe flange connetions with gasket on sealing performance. *ASME Internation PVP Conference*. pp. 65-75.
7. Cao, D. & Xu, H. (1999). 3-D finite element analysis of bolted flange joint considerring gasket non-linearity. *ASME International PVP Conference*, Vol. 382. pp. 121-126.

8. Hurrel, Paul R. (2000). Good practice in modeling of pressure vessel bolted joints for stress and fatigue analysis. *ASME International PVP Conference*. Vol. 405, pp. 123-134.
9. Takaki, T. & Fukuoka, T. (2000). Bolt-up strategy for pipe flange connections using finite element analysis. *ASME International PVP Conference*. Vol. 405, pp. 143-149.
10. Takaki, T. & Fukuaka, T. (2001). Finite element analysis of bolt-up operations for pipe flange connections. *ASME International PVP Conference*. Vol. 416, pp. 141-147.
11. Takaki, T. & Fukuaka, T. (2002). *Systematical FE analysis of bolt assembly process of pipe flange connections*. *ASME International PVP Conference*. pp. 147-152.
12. Takaki, T., et al. (2004). Effects of flange rotation on the sealing performance of pipe flange connections. *ASME Internation PVP Conference*. Vol. 478, pp. 121-128.
13. Takaki, T. & Fukuoka, T. (2004). Effective bolting up procedure using FEA and elastic interaction coefficient method. *ASME Internation PVP Conference*. Vol. 478, pp. 155-162.
14. Abid, M. (2000). Experimental and analytical studies of conventional (gasketed) and unconventional (non-gasketed) flanged pipe joints (with special emphasis on the engineering of 'joint strength' and 'sealing'). PhD. Thesis.
15. Jiang, Y., et al. (2000). An experimental investigation on frictional properties of bolted joints. *ASME Internation PVP Conference*, Vol. 433, pp. 59-66.
16. Tsuji, H. & Nakano, M. (2002). Bolt preload control for bolted flange joint. *ASME Internation PVP Conference*. pp. 163-170.
17. Fukuoka, T. & Sawa, T. (2004). Evaluation of the tightening process of bolted joint with elastic angle control method. *ASME Internation PVP Conference*. pp. 11-18.
18. Zhang, M., Jiang, Y. & Lee, C. H. (2004). Finite element modeling of self-loosening of bolted joint. *ASME Internation PVP Conference*, pp. 19-27.
19. Bouzid, A. H. & Nechache, A. (2004). *Creep modeling in bolted flange joints*. *ASME Internation PVP Conference*, pp. 49-56.
20. Shoji, Y. (2004). An effect of gasket stress distribution on tightness estimation in pipe flange connectios by finite element analysis. *ASME Internation PVP Conference*. Vol. 478, pp. 113-120.
21. Brown, W. (2004). Efficient assembly of pressure vessel bolted joints. *ASME Internation PVP Conference*. Vol. 478, pp. 163-168.
22. Abid, M. & Nash, D. H. (2005). Structural strength: Gasketed vs. non-gasketed flange joint under bolt up and operating conditions. *International Journal of Solids and Structures*, Vol. 43, pp. 4616-4629.
23. Abid, M. & Nash, D. H. (2006). Joint relaxation behavior of gasketed bolted flange joint during assembly. *WSEAS International Conference on Applied and Theoratical Mechanics*. Vol. 123, pp. 319-325.
24. Abid, M. & Nash, D. H. (2006). Bolt bending behavior in bolted flange pipe joint: A comparative study. *ASME International PVP Conference*, pp. 1-9.
25. Abid, M. & Hussain, S. (2008). Gasketed flange joint's relaxation behavior during assembly using spiral wound gasket. *Journal of Process Mechanical Engineering, Proceedings of the IMechE*, Vol. 222/2, pp. 123-134.
26. Hussain, S. (2007). Bolted joint's relaxation behavior using different gaskets. A Comparative Study. MS Thesis.
27. ASME PCC-1, (2000). Guidelines for preussure boundary bolted flange joint assembly.
28. ES/090 Rev:1, Desing and engineering practices (DEPs), DEP 31.38.01.15.Gen (Piping Class Exploration and Production).
29. ASME/ANSI B16.5, (1998). Pipe flanges and flanged fittings. Sec VIII, Div. I.
30. ANYSY Inc. (2004). *ANSYS elements manual*. Seventh Edition.
31. ASME, (2006). Boiler and pressure vessel code. Section II, Part D, American Society of Mechanical Engineering, New York USA.
32. Garlock Gasket Manufacturer, http://www.garlock.com