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Simulation of Optimized Bolt Tightening Strategies for Gasketed Flanged Pipe Joints

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Abstract

This paper presents results of detailed non-linear finite element analysis (FEA) of gasketed bolted flange pipe joints (GBFJ) of different sizes (1, 4, 5, 6, 8, 10, 20 inch) of 900# pressure class for achieving proper preload close to the target stress values with and without considering yielding at bolt and flange and gasket crushing recommended by ASME and industrial guidelines for optimized performance using customized optimization algorithm. In addition, two strategies TCM (torque control method) and SCM (stretch control method) are used which is a normal practice in the industry.

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Keywords: Bolted; Pipe; flange; joints; torque control; stretch control; finite element analysis; yielding; optimized

1. Introduction

Gasketed bolted flange joints are extensively used in process industrial applications. Due to the availability of the computational power in the last decade, modeling and simulation has made it possible to visualize the behavior of individual components and as assembly as a whole for its safe operation. Performance of the gasketed joint is realized based on the assembly process of the bolts in the joint. Initially studies are observed for 2D or 3D

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axisymmetric models, however studies using full 3D models are also observed to study bolts and overall joint relaxation behaviors. In most of the studies displacement at the bottom of the bolt is applied to achieve required bolt pre stress and due to manual inputs and hit and trial runs these does not provide clear methodology and does not ensure confidence, accuracy and time saving [1-32]. Based on these limitations, generalized algorithm is developed for accurate results for the required target stress, times saving for solution and result recording as required compared to the manual inputs and can be implemented to all different flange sizes and classes using both the TCM and SCM. In this paper only results of different flange sizes for 900# class are only presented.

2. Finite Element Modeling, Meshing, Material Selection, Boundary conditions and Solution

Three dimensional models of different flange sizes of Class 900# [33] (1, 4, 5, 6, 8, 10, 20 inch) are developed. Due to the rotational symmetry, a part of the flange, pipe, bolt and spiral wound gasket is modeled first and is then revolved to form full model. One bolt is modeled first while all others are generated using rotational symmetry. Flange sizes of 4 and 5 inch while 6 and 8 inch follow the same modeling and meshing strategies because of the same number of bolts. In this study elasto-plastic material model is used for all the flange sizes. Allowable stresses for flange and bolts are given in the Table 1 [34], whereas gasket material properties and dimensions are used from Garlock [35]. Flange, pipe and bolts are modeled using Solid45elements. Interface elements (INTER195) and TARGET170 and CONTA174 elements are used for gasket and contacts generation. In order to have flange rotation gasket and flange are free to move in the radial and axial direction, with symmetry conditions applied at the lower portion of the gasket. To observe bolt bending, bolts are constrained at mid-section in the tangential and radial direction. To apply preload, an axial displacement is applied in the downward direction at bolt bottom areas. ANSYS software is used for analysis [36]. ASME [37] and two industrial [38,39] guidelines with two bolt up strategies i.e. torque control method (one bolt is tightened at one time) for flange sizes 1, 4, 5, 6, 8 inch and stretch control method (a group of bolts is tightened at one time) for flange sizes 10 and 20 inch are analyzed in this study. Meshed model of gasketed joint with applied boundary conditions is shown in Fig. 1.

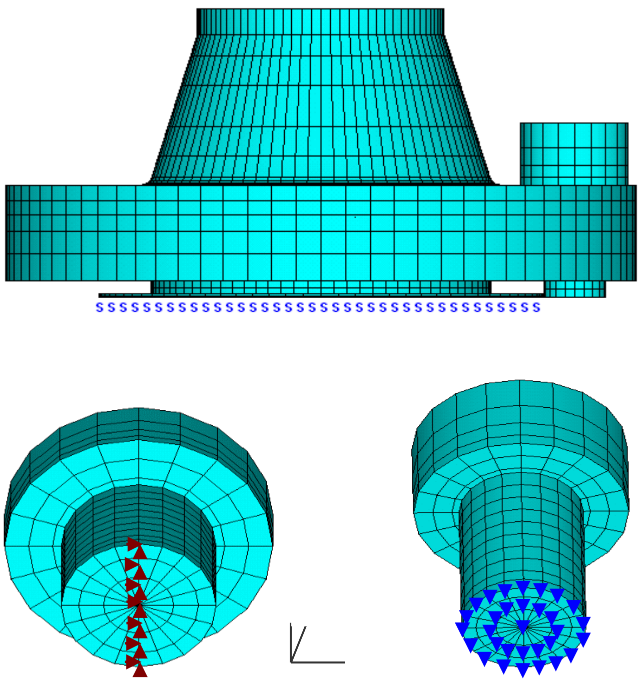


Fig. 1. Meshed model of gaskted joint assembly and applied boundary conditions.

Table 1. Material Properties.

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

TCM uses torque wrenches to apply torque on bolts due to which nut or bolt is turned against the surface of the flange and bolt is stretched and bolt preload is calculated using Bickford and Nassar [40]. In SCM stud is stretched by applying hydraulic pressure to the tensioner; nut is coiled against joint face and then pressure is released after which tool is removed. As the pressure is released bolts act as spring and tension is created in the bolt and bolt elongation is calculated as per ref [37]. In TCM, using ASME guidelines, bolts are tightened in cross pattern (sequence-1) for first three passes and in clockwise pattern (sequence-2) in the 4th pass. Using industrial guidelines Garlock [35] and ES090 [39] bolts are tightened in cross pattern (sequence-1) for first four passes and in clockwise pattern (sequence-2) in 5th pass.

In stretch control method, bolts are tightened by stretching 100% (SC100), 50% (SC50) and 33% (SC33) of the bolts at a time. Details of tightening sequence, number of passes and percentage increment of target torque for TCM and tensioning for SCM are summarized in Pre stress values and torque/stretch increments are given in Table 2 and Table 3 respectively.

Table 2. Pre stress values for 1, 4, 5, 6, and 8 inch flange size.

NS	Bolt dia(m)	Target Torque(Nm)	Pre-stress value for each pass (MPa)				
			P1	P2	P3	P4	P5
ASME-PCC1 ASME [37]							
4	0.0280	902	78	170	261	261	-
5	0.0320	1268	74	160	246	246	-
6	0.0285	1037	85	185	285	285	-
8	0.0350	1593	71	155	238	238	-
INDUSTRIAL Garlock [35]							
1	0.0222	198	35	75	115	115	-
4	0.0280	780	68	147	226	226	-
5	0.0320	1091	64	138	212	212	-
6	0.0285	896	74	160	246	246	-
8	0.0350	1359	61	132	203	203	-
INDUSTRIAL ES090 [39]							
4	0.0280	505	52	86	115	146	146

3. Optimization Algorithms

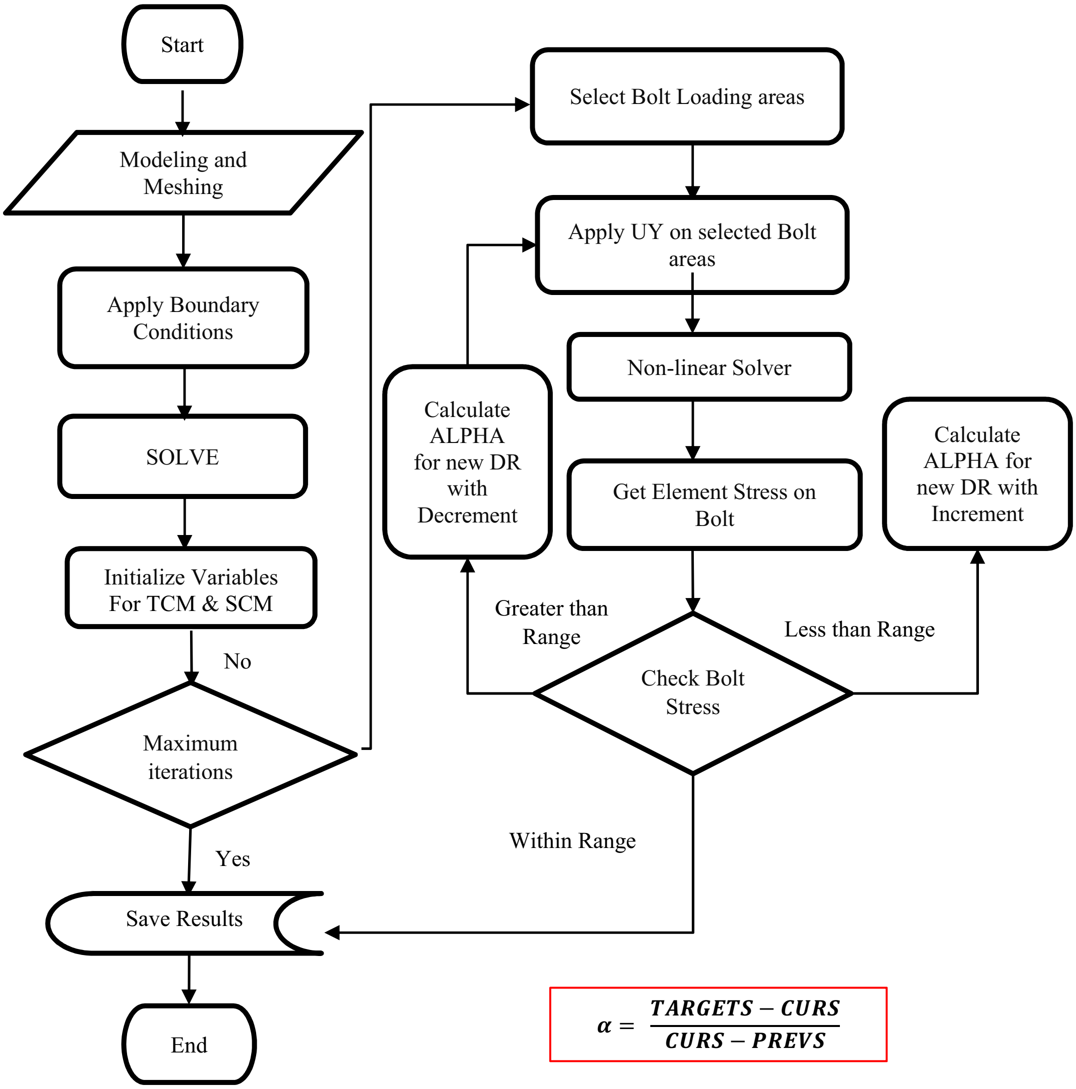
Keeping in view the limitations of previous work based on axisymmetric modeling and manual axial displacement inputs, in this present study, a detailed 3D nonlinear finite element analysis is performed for Class 900# only but for different flange sizes using a generalized optimization algorithm to achieve bolt up values within range defined by ASME and industry and works for all bolts and for all passes for TCM and SCM. In addition analysis is performed by updating optimization code avoiding any yield in the flange, resulting in bolt scatter, bolts and joint relaxation and compromised sealing. Algorithm is also explained in Fig. 2. Optimization algorithm with yielding is explained for both the TCM and SCM.

- Target stress variables are defined including differential rate (DIFF) and displacement rate (DR) for bolt up and directional variable for increment/decrement in the displacement (UY) value to keep the target stress in range. Current stress (CURS) is defined to indicate the value of stress for each iteration.
- Differential rate is for the initial jump to reach the required target stress value quickly.
- Displacement rate is the initial value for the bolt up which is to be incremented after iteration until the target stress is achieved.
- Number of iterations is kept up to 600 to achieve required target stress before loop ends.
- Initial bolt up includes both the differential and displacement rates.
- Differential rate (DIFF) is returned to zero after the first iteration while the subsequent iterations are continued with the increment in DR only. Minimum and maximum stress values are saved in LOW_TARGETS and MAX_TARGETS variables respectively, for each pass.
- Just as the target stress value is achieved in the bolt, results of all required stresses i.e. of flange, bolts and gasket are saved in the output file and are called using macros to avoid repetitions

Table 3. Torque Increments.

No. of Bolts	**Bolt tightening sequence**	**Pass**	**Bolts Tightening**
TCM - ASME PCC-1 ASME [37]			
 4 8 12	Sequence-1 1-3-2-4 1-5-3-7-2-6-4-8 1-7-4-10-2-8-5-11-3-9-6-12	P1 P2 P3	20% to 30% of Target Torque 50% to 70% of Target Torque 100% of Target Torque
 4 8 12	Sequence-2 1-2-3-4 1-2-3-4-5-6-7-8 1-2-3-4-5-6-7-8-9-10-11-12	P4	100% Target Torque
TCM – Industrial ES090 [39]			
 4 8 12	Sequence-1 1-3-2-4 1-5-3-7-2-6-4-8 1-7-4-10-2-8-5-11-3-9-6-12	P1 P2 P3 P4	25% to 35% of Target Torque 50% to 60% of Target Torque 70% to 80% of Target Torque 100% of Target Torque
 4 8 12	Sequence-2 1-2-3-4 1-2-3-4-5-6-7-8 1-2-3-4-5-6-7-8-9-10-11-12	P5	100% of Target Torque
SCM SKF [38]			
No. of Bolts	Bolt tightening sequence	Group	Bolts Tightening
16 (10 inch) 20 (20 inch)	(1,5,9,13), (2,6,10,14), (3,7,11,15), (4,8,12,16) (1,5,9,13,17), (2,6,10,14,18), (3,7,11,15,19), (4,8,12,16,20)	G1,G2,G3,G4	25 or 33% Tensioning
16 (10 inch) 20 (20 inch)	1,3,5,7,9,11,13,15 2,4,6,8,10,12,14,16 1,3,5,7,9,11,13,15,17,19 2,4,6,8,10,12,14,16,18,20	G1 G2 G1 G2	50% Tensioning
16 (10 inch) 20 (20 inch)	1 to 16 1 to 20	One	100% Tensioning

According to the specified tightening sequences and bolt preload values, during bolt up, there yielding is observed at hub flange fillet in different flange sizes, resulting in their failure. Area of hub flange fillet is selected around 360° after which elements attached to this area are selected. Finally nodes attached to these elements are selected and grouped in a component to check stress value in them. To avoid yield, variable for flange yield's stress is defined. When the yield value defined in this variable is achieved the iteration is stopped and moved to the next step, ignoring whether the target stress is achieved or not.



$$\alpha = \frac{TARGETS - CURS}{CURS - PREVS}$$

Fig. 2. Optimization Algorithm for TCM and SCM.

4. Results and discussions

4.1. Comparison of optimized results of manually input and automated algorithm

For 8 inch flange size, target bolt stress variation and stress variation at hub flange fillet were observed as per industrial Garlock guidelines. Comparison of optimized target bolt stress results of manual input and automated algorithm are summarized in Table 4. Maximum stress at hub flange fillet using manual input and automated algorithm observed is 218MPa and 264MPa respectively for no yielding case whereas allowable stress value is 248 MPa. As target bolt stress values are 65MPa, 135MPa and 205MPa for 1st, 2nd, 3rd and 4th pass respectively, however, using manual inputs and automated algorithms, stress variation from target bolt stress of 40 MPa and 3 MPa respectively are observed.

Table 4. Comparison of manual and optimized target stress values (M = Manual input, O = Automated algorithm) – 8 Inch flange size.

SQ1	P1(M)	P1(O)	P2(M)	P2(O)	P3(M)	P3(O)	SQ1	P4(M)	P4(O)
B1	54	63	125	134	197	202	B1	191	202
B7	61	62	127	132	197	202	B2	191	202
B4	47	61	108	134	190	204	B3	242	203
B10	54	61	116	133	194	201	B4	176	202
B2	57	61	120	132	179	202	B5	192	203
B8	60	60	124	131	179	205	B6	239	203
B5	55	60	119	134	184	204	B7	185	203
B11	56	60	121	135	184	201	B8	202	202
B3	55	60	115	131	195	205	B9	217	204
B9	46	63	97	134	162	204	B10	189	203
B6	53	61	114	133	196	204	B11	204	204
B12	53	63	117	131	194	201	B12	202	202

4.2. Comparison of TCM and SCM for all flange sizes

Different flange sizes are compared on the basis of axial bolt stress variation, gasket stress distribution and flange hub stress variation for 4,5,6,8 inch sizes using TCM and for 10 and 20 inch sizes using SCM.

4.3. Axial bolt stress variation:

Figure 3 show the comparison of stress variation at the end of last pass for 1, 4, 6, 8, 10 and 20 inch flange sizes. In case of 8 bolts flange size stress variation is higher for 4 inch size as compared to 5 inch size. In case of 12 bolt flange size stress variation is higher in 6 inch size compared to 8 inch. In case of 10 and 20 inch sizes tightened according to SCM stress variation is less as compared to all other sizes. In case of 1, 4 and 5 inch flange sizes maximum variation is 32MPa, 42MPa, 48MPa respectively while in case of 10 and 20 inch sizes maximum difference is 6MPa and 7MPa respectively.

4.4. Gasket Stress distribution:

Figure 4 show the comparison of gasket stress distribution for different flange sizes at the end of last pass. 1 inch flange size shows highest variation but gasket stress is uniform as the flange size increases such as 8 inch flange size using TCM. In case of 10 and 20 inch sizes it is observed uniform using SCM. Maximum difference in case of 1, 4 and 5 inch flange sizes is 50MPa, 40MPa and 22MPa respectively and in case of 10 and 20 inch flange sizes maximum difference is 0.1MPa and 0.4MPa respectively which is almost negligible difference.

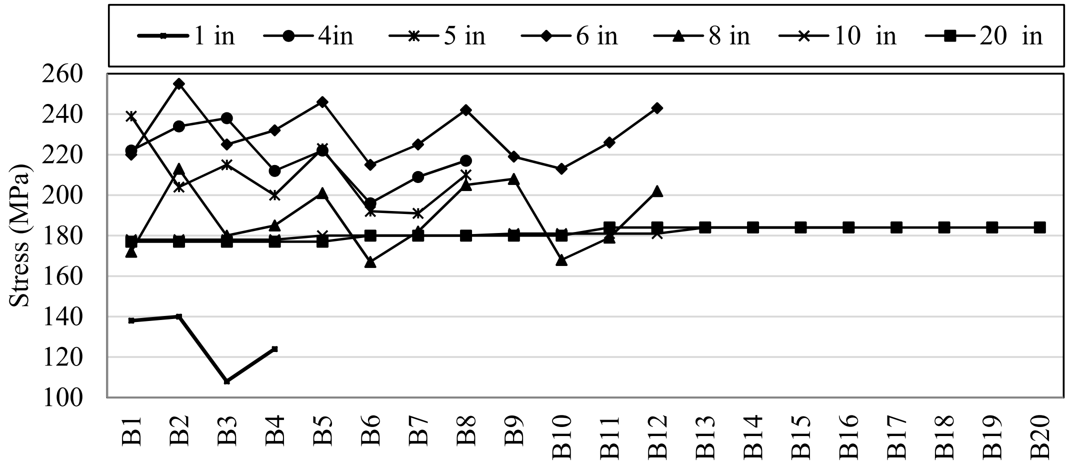


Fig. 3. Axial bolt stress variation of different flange sizes.

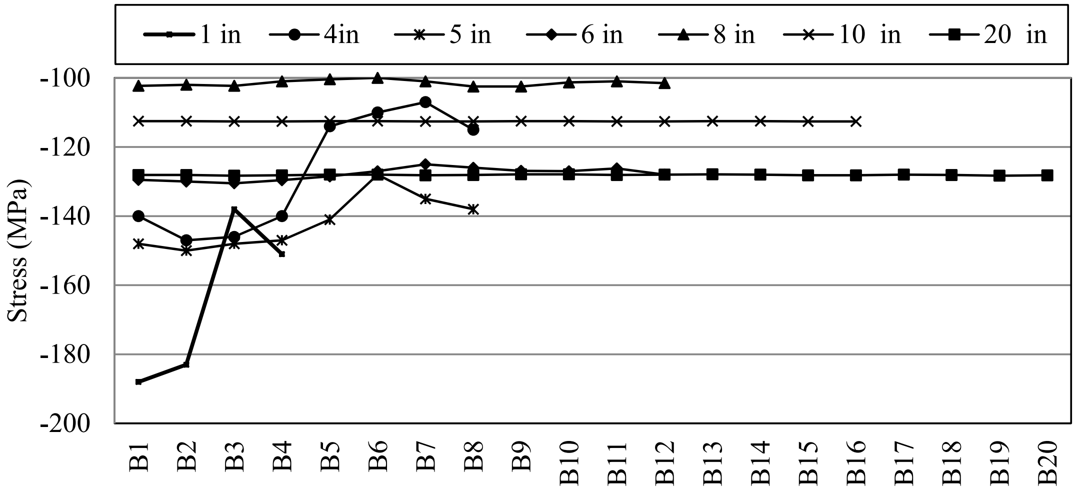


Fig. 4. Gasket stress distribution for different flange sizes.

4.5. Hub flange stress variation:

Figure 5 show hub flange stress variation for different flange sizes. In case of 4, 5, 6, and 8 inch sizes hub flange variation is taken along 0° for all passes. In case of 10 and 20 inch sizes stress variation is taken at HF-1. Maximum hub flange stress variation is observed in 4 inch flange size and minimum variation is observed in 20 inch flange sizes. However, stress variation is uniform at the end of each pass but in case of 1 inch flange size there is variation in the last pass as well. Maximum flange stress in case of 4 inch size is 385MPa and in case of 20 inch size it is 253MPa.

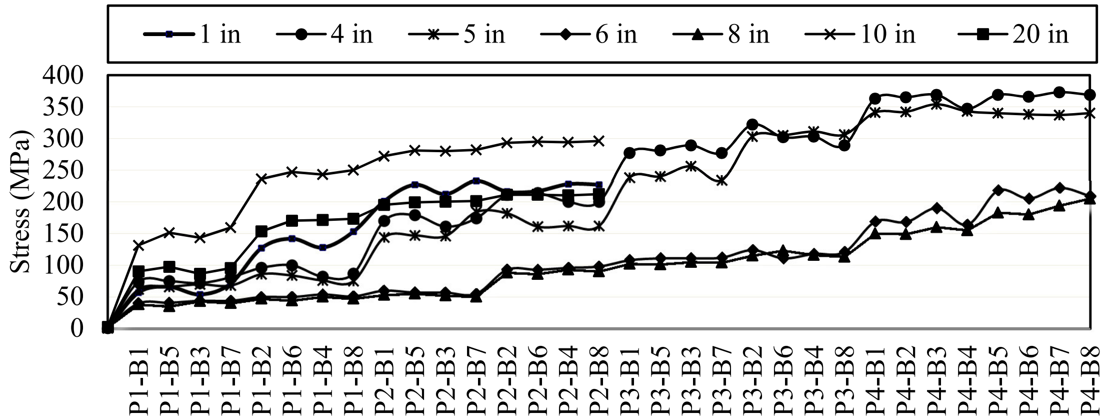


Fig. 5. Hub flange stress variation.

5. Conclusions

From detailed FEA analysis of bolted flange joints following conclusions are made:

- With an increase in the flange size there is less variation in bolts, gasket and flange stresses. However, in small flange sizes these variations are comparatively large.
- Optimization of bolted joints bring bolt preload almost close to the defined target stress as errors are minimized in automatic bolt up by 0-5MPa compared to manual hit and trial of preload of 0-50MPa.
- Creating macros for writing required values in output file and setting bolt preload values automatically, saves significant computational time as compared to manual method.
- Simultaneous simulations can be run at a time as desired without any manual interaction depending on parallel processing.
- Comparison of different target torque values showed that industrial Garlock[35] torque values are better than ASME. ASME target torques is greater than industrial and in all cases joints are overstressed and it initiates yielding earlier than industrial target torques. In case of 20 inch flange size no yielding is achieved if industrial target torques is used, however, yielding is achieved if ASME torques is used.
- Hub flange fillet stress value remains less than yield value (248MPa) of flange using customized optimization code for no yielding along 360° location and should be adopted.
- Gasket stress in all cases for all target torques does not exceed the maximum allowed stress of 206MPa. Therefore, gasket crushing does not occur. However, if yielding is avoided probable leakage is achieved in 4, 5 and 6 inch sizes because of gasket stress, which is less than minimum defined stress.
- Bolt tightening procedure affects the bolt bending behavior. Under TCM each bolt show different bolt bending behavior while in case of SCM bolts tightened simultaneously show similar bending behavior.
- Magnitude of performance parameters are affected by dimensional variations. Small flange sizes show large bolt scatter, as compared to large flange sizes having relatively uniform scatter. To control yielding, dimensional changes are required in flange sizes and ASME target torques should be re-considered to avoid over stressing of flanges.
- Comparative behavior of all the sizes concludes SCM better than the TCM for axial bolt stress variation, for uniform gasket stress distribution and maximum stress at hub flange fillet.
- Further optimization may be performed under operating conditions and the code may further be updated for safe operating conditions.

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