Optimized Bolt Tightening Strategies for Different Sizes of Gasketed Flanged Pipe Joints

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Abstract

Achieving a proper preload in the bolts of a gasketed bolted flanged pipe joint during joint assembly is considered important for its optimized performance. This paper presents results of detailed non-linear finite element analysis (FEA) of an optimized bolt tightening strategy of different joint sizes for achieving proper preload close to the target stress values. Industrial guidelines are considered for applying recommended target stress values with TCM (torque control method) and SCM (stretch control method) using a customized optimization algorithm. Different joint components performance is observed and discussed in detail.

Keywords: Bolted; flangeed pipe joints; torque control; stretch control; finite element analysis; optimized

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Nomenclature

TCM	Torque control method
SCM	Stretch control method
2D	Two dimensional
3D	Three dimensional
SC	Stretching of bolts at one time during bolt tightening
DIFF	Differential rate in target stress variables
DR	Displacement rate for bolt up
UY	Axial displacement of bolts
CURS	Current stress during each iteration
ASME	American Society for Mechanical Engineers
B	Bolt number
G	Bolt group used during SCM
P	Pass number
M	Semi-automated algorithm
O	Fully automated algorithm
HF	Hub flange fillet location

1. Introduction

Gasketed bolted flange joints are extensively used in process industrial applications. Performance of a gasketed bolted flanged pipe joint is related to the proper joint assembly with all bolts tightened to the recommended target bolt stress values by ASME or industrial guidelines. However based on factors including target stress values available, tools, fitters training, gasket materials, bolts quality and others, sealing and strength of a joint cannot be ensured experimentally. Keeping in view these practical limitations, based on 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. Researchers including Cao and Xu [1], Takkaki and Fukuoka [2-5], Abid [6], Nagata *et al* [7],Tsuji and Nakano [8], Sawa *et al* [9], Fukuoka and Sawa [10], Zhang et al [11], Bouzid and Nechache [12], Shoji [13], Takkaki [14] and Brown and Warren [15] have performed detailed finite element studies keeping in view the limitations of experimental work. They used flange displacement at the bottom of the bolt by hit and trial methods target stresses in the bolts is achieved. Abid *et al* [6] have highlighted yielding at flange and crushing of the gasket during experimental work for one size only as it is impossible to test all sizes which again practically are impossible in field applications. Therefore Abid *et al* [6, 16-32] have performed detailed 2D and 3D numerical studies, but limitations are observed in terms of semi-automatic applications of algorithm and results recording at required locations using manual picking at different joint components concluding hectic and time consuming. Keeping in view the above limitations, a 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 for recommended bolt stress values by Industrial guidelines. In this paper only results of different flange sizes for Class 900# class [33] (1, 4, 5, 6, 8, 10, 20 inch) are only presented.

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

Three dimensional models of different flange sizes are developed. Due to the rotational symmetry, a part of the flange, pipe, bolt and spiral wound gasket is modelled first and is then revolved to form full model. One bolt is modelled first while all others are generated using rotational symmetry. In this study elasto-plastic material model is used for all the flange sizes. Allowable stresses for flange and pipe are as per ASTM A350 LF2 and bolts are as per ASTM SA193 B7 taken from ref. [34] and are used in the industry. Gasket material properties and dimensions are used from Garlock [35]. Flange, pipe and bolts are modelled using Solid45 elements. 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]. Industrial [35] guidelines are used for torque control method (one bolt is tightened at one time) for flange sizes 1, 4, 5, 6, 8 inch. For stretch control method (a group of bolts is tightened at one time) for flange sizes 10 and 20 inch using SKF strategy [37]. Meshed model of gasketed joint with applied boundary conditions is shown in Fig. 1.

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 [38]. 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 the bolt are elongated. In TCM, bolts are tightened in cross pattern (sequence-1) for first four passes and in clockwise pattern (sequence-2) in 5th pass. In SCM, 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 Table 1 and Table 2 respectively.

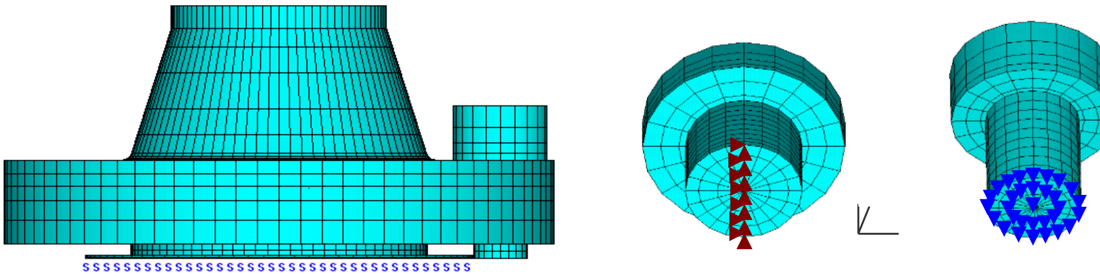

Figure 1: Meshed model of gaskted joint assembly and applied boundary conditions

Table 1: Pre stress values for 1, 4, 5, 6, and 8 inch flange size using TCM (Garlock [35])

NS	Bolt dia(m)	Target Torque(Nm)	Pre-stress value for each pass (MPa)				
			P1	P2	P3	P4	P5
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	-

Table 2: Torque Increments for 10 and 20 inch flange sizes using SCM (SKF [37])

No. of Bolts	Bolt tightening sequence	Group	Bolts Tensioning
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)	G1~4	25 or 33%
16 (10 inch)	1,3,5,7,9,11,13,15 2,4,6,8,10,12,14,16	G1 G2	50%
20 (20 inch)	1,3,5,7,9,11,13,15,17,19 2,4,6,8,10,12,14,16,18,20	G1 G2	
16 (10 inch) 20 (20 inch)	1 to 16 1 to 20	One	100%

3. Optimization Algorithms

A generalized optimization algorithm to achieve bolt up stress values within range defined by ASME and industry is presented in Fig. 2 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 and 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. Maximum iterations are kept up to 600 to achieve required target stress before loop ends. As soon as 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. According to the specified tightening sequences and bolt preload values, during bolt up, 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 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.

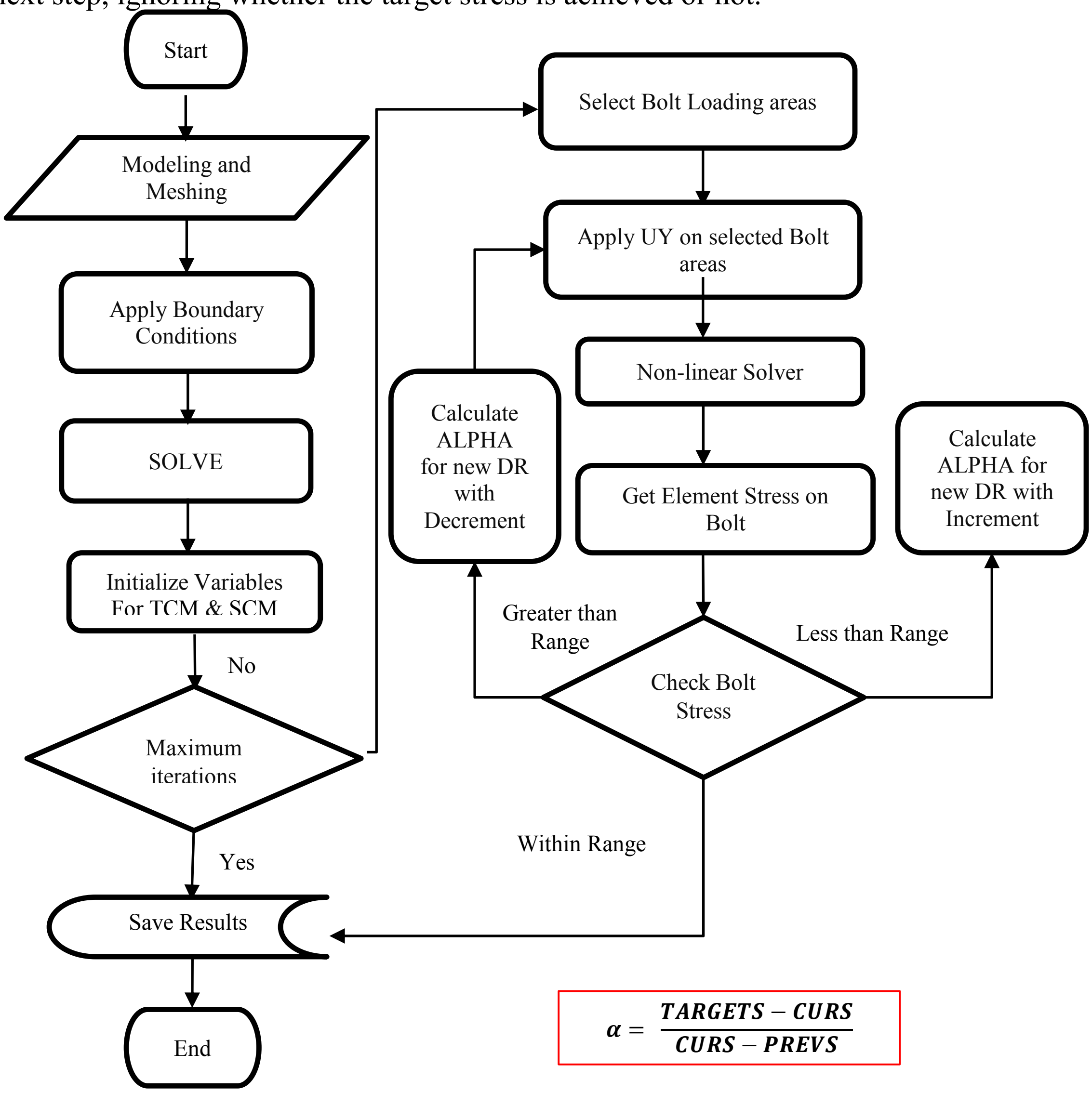

Figure 2: Optimization Algorithm for TCM and SCM

4. Results and discussions

In this section results are discussed in detail only for 8 inch flange size of Class 900#, whereas results are summarized for flange sizes of 1, 4, 5, 6, 8, 10, 20 inch.

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

In this section results of flange of size 8 inch are presented. Target bolt stress variation and stress variation at hub flange fillet were observed. Comparison of optimized target bolt stress results of manual input and automated algorithm are summarized in Table 3. 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 3: Comparison of manual and optimized target stress values

SQ	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 1,4,5,6,8 inch sizes using TCM and for 10 and 20 inch sizes using SCM.

4.3.Axial bolt stress variation:

Fig. 3 shows 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:

Fig. 4 shows 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.

4.5.Hub flange stress variation:

Fig. 5 shows 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.

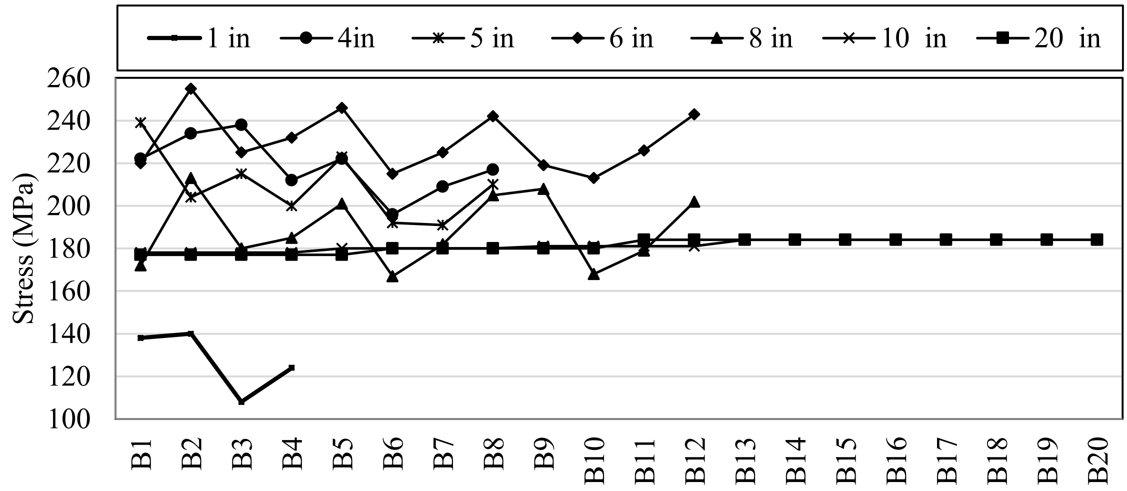

Figure 3: Axial bolt stress variation of different flange sizes

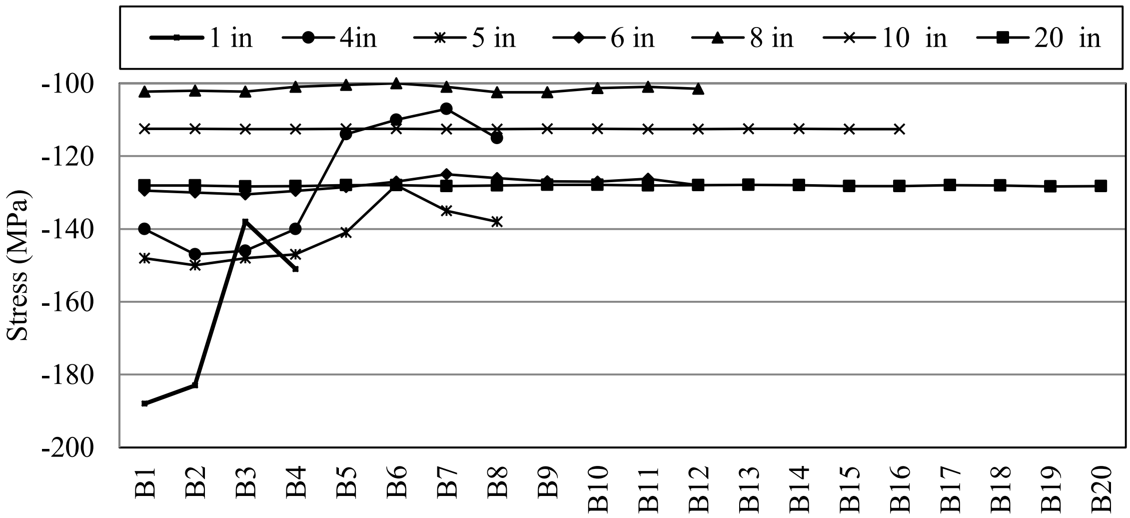

Figure 4: Gasket stress distribution for different flange sizes

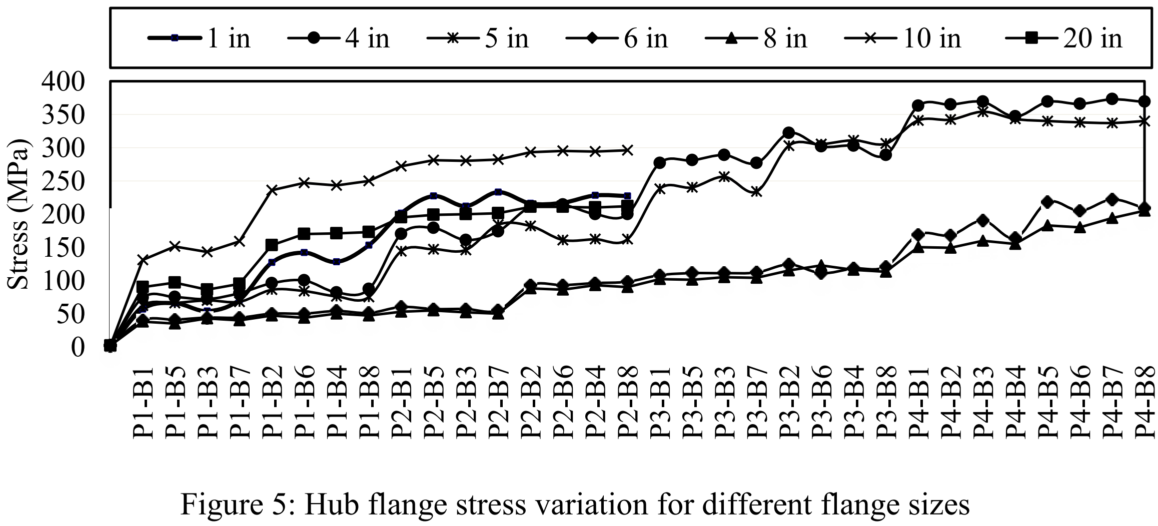

Figure 5: Hub flange stress variation for different flange sizes

5. Conclusions

It is concluded that using customized algorithms developed in this study, bolt target stress values are observed almost close to the defined target stress as errors are minimized to 0-5MPa compared to manual hit and trial values 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. In addition simultaneous simulations can be run at a time as desired without any manual interaction. 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. 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. Comparative behavior of all the sizes concludes SCM better than the TCM for axial bolt stress variation, bolt bending, gasket stress distribution and maximum stress at hub flange fillet.

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