Scientia Iranica, Vol. 13, No. 3, pp 303–309

SCIENTIA IRANICA

Research Note

Stress Variation in the Flange of a Gasketed Flanged Pipe Joint During Bolt up and Operating Conditions

M. Abid[1]

This paper presents results of an experimental study of the behavior of the stress variation at the flange and attached pipe section of a gasketed flanged pipe joint during both the bolt up (pre-loading) procedure and under operating (internal pressure loading) conditions. Stress variations showing flange yielding, flange rotation, effects of a joint tightening sequence, identification of the mode of response to loading (static or dynamic) and the effects of retightening are discussed in detail. Additionally, the importance of high quality bolting with proper surface treatment and the use of proper tooling to reduce the magnitude of overall flange stress variation are also discussed.

INTRODUCTION

The use of gasketed flanged pipe joints is commonplace in industry. It is well known that gasketed flanged pipe joints are prone to leakage, even after careful and controlled pre-loading, which becomes apparent during operating loading conditions. In the previous work by Webjorn and Abid et al [1-4], a dynamic mode-of-load is concluded in gasketed pipe joints, during bolt up and operating conditions. Bolt up means that a pre-load is applied in the bolts to make a joint assembly and operating condition means applied loading conditions, such as internal pressure. Due to the flexible gasket element and the gap between flange faces, flange rotation occurs with a pivot point at the outside diameter of the flange-raised face. Under operating conditions, the flange faces move apart. These effects become worse by adopting procedures such as hammering during joint assembly, which damages, not only the joint, but also, the equipment, to which these are attached. Another important factor effecting gasketed joint performance is the re-tightening after the application of the internal pressure loading. Re-tightening means the tightening of the bolts in a joint again after sometime. This is studied and the results are discussed in detail.

1. *Department of Mechanical Engineering, Ghulam Ishaq Khan Institute of Engineering Sciences and Technology, Topi, Pakistan.*

A flanged pipe joint is an assembly of discrete components comprised of a gasket, flanges, bolts and washers. Traditional design codes determine the size and thickness of the flange rings, based on the required pressure, bolt and gasket characteristics. Amongst all the components in the joint, the flange ring, being composed of a thick steel ring and the bolts, due to a bigger diameter, are considered strong. However, in this study, the performance of the flange ring is highlighted as being significantly important for a successful joint, as any yielding at the flange hub leads to joint failure. After initial yielding, specifically at the hub flange fillet, alignment provides a problem for the gasket, even causing the bolts to bend, resulting in failure of the joint and eventual leakage.

In order to reduce stress variation, experimental studies are undertaken and factors, such as the importance of using proper tooling, high quality bolting with proper surface treatment and the correct bolt tightening sequence, are discussed.

EXPERIMENTAL SETUP AND PROCEDURE

To examine stress variation behavior in flanges, experiments were performed with four gasketed flange joint assemblies, using four-spiral wound gaskets, i.e., one gasket in each joint. The four-spiral wound gaskets were of the same make, material and dimensions and were used to examine their effect on the stress variation in the flange during bolt up and under operating conditions. In industrial practice, when it is time for a new joint assembly, i.e., during maintenance or repair, a gasket used once needs to be replaced, in order for proper joint sealing to occur due to compressed or damaged behavior.

Flange Type, Size and Material of Flange Joint Components

A 4-inch nominal bore, class 900# gasketed flange [5] was employed and a suitable test rig was made (Figure 1). For all tests, the same pair of flanges with four different gaskets of the same dimensions, properties and material was used in the assembly to examine variability in the supplied gaskets and their effect on joint behavior. The type of gasket used was of inside bolt circle diameter type B that is used in a raised face flange joint [6]. Material and material properties, as per associated codes/standards with allowable stresses, are given in Table 1.

Strain Gauging and Instrumentation

Flange and Pipe

Initially, 4 pairs of strain gauges of 120 Ω resistance were attached at the hub-pipe fillet, the hub-flange fillet, the hub centre and the pipe (away from stress discontinuity) locations at an angle of 90° to measure axial and hoop strains and to observe the behavior of the flange at the top, bottom and side locations during loading and unloading (Figure 2). Inaccurate strains were initially recorded for gauges of a 3.5 mm length in an axial direction. This was concluded due to the longer length of the strain gauge, i.e., 3.5 mm at the hub flange fillet and hub pipe fillet regions. In order for accurate readings, strain gauges of a small grid length of 1.57 mm were attached at the hub-pipe and hub-flange intersections. Strain measurements recorded experimentally were verified from the finite element analysis results [7].

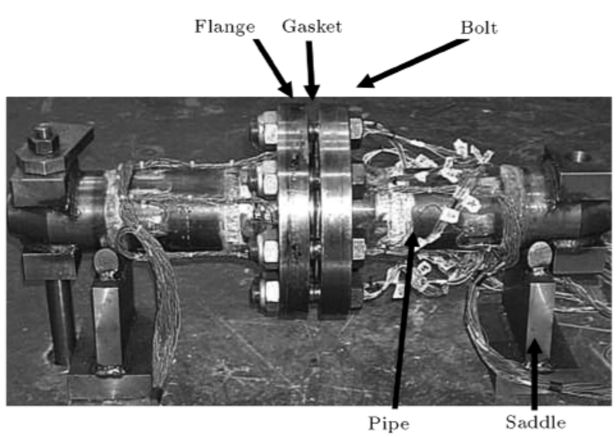

Figure 1. Gasketed flange joint assembly.

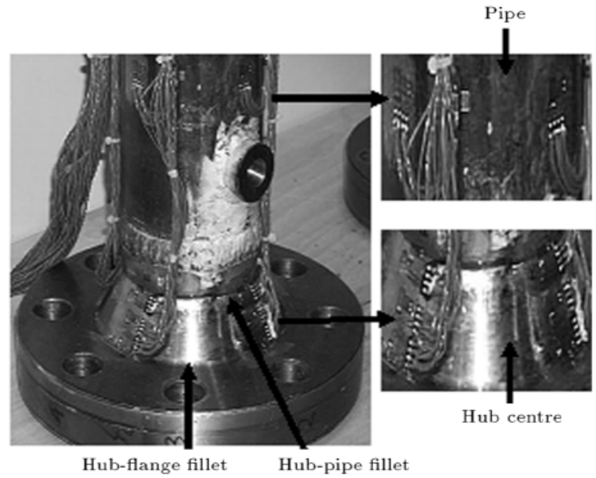

Figure 2. Strain gauging at flange and pipe and location at which strains results are recorded.

Bolts

Four strain gauges of 350 Ω were placed at an angle of 90° on the shaft of each bolt, as shown in Figure 3. To attach the strain gauges, a groove of 2 mm was machined on the bolt shank to avoid damage and all leads were placed on the hexagonal head of the bolt through a very small hole drilled in the head. The hole drilled for the lead wires does not provide any concentration point for stresses, as the strain is measured on the bolt shank and, as the bolt is very strong and of 30 mm in diameter, this does not effect actual strain results.

Strain gauge locations along the top, bottom and side of the hub flange fillet, hub pipe fillet and hub centre, are shown in Figure 4.

Table 1. Material properties.

Parts	E (N/mm^2)	ν	Material as per Code [11]	Allowable Stress (N/mm^2)
Flange/Pipe	173058	0.3	SA-105/ASTM105	248.2 (2/3rdσ_y)
Bolt	168922	0.3	ASTM A193 GRADE B7 and B7M	723.9
Gasket (Taken as Solid Ring)	164095	0.3	SA-182/ASTM-182	206.8 (2/3rdσ_y)

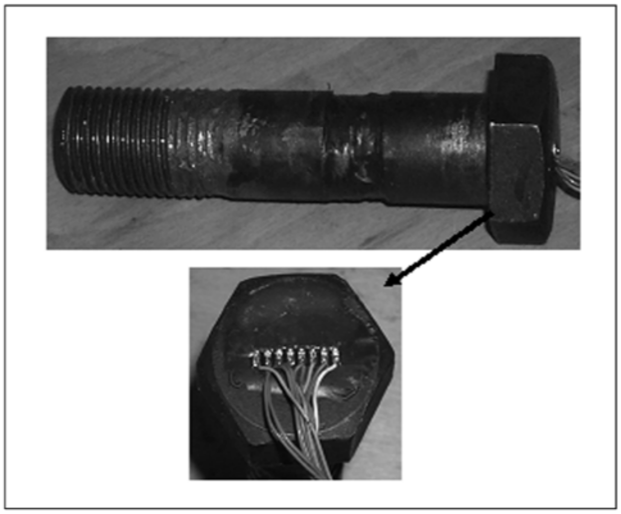

Figure 3. Strain gauging on bolt.

Instrumentation

Quarter Bridge circuits were made for all the strain gauges attached to bolts, flanges and pipe sections, with a data-logging system for strain measurements.

Data Recording and Strain Measurements

After making all the connections, joint assembly was first undertaken and then tests were performed, during loading and unloading, with internal pressure. Strains were recorded in real time using a computerized data logging system. Gauge factors for all the strain gauges used were input into a computer and then initializing strains were recorded for different loadings. Results were saved in the computer and were plotted in the required format by converting into stresses. In the data logging system, the plot of each strain gauge reading was observed, which provided verification of accurate results.

Flange Joint Assembly and Tests

A hand-tightening methodology, using an ordinary spanner, was adopted to make the joint assemblies, as it is often the first and most economic method of assembly. To ensure a proper pre-load in a joint and to highlight the importance of the bolt tightening sequence, the following two bolt tightening sequences were used:

- Sequence-1: 1, 5, 3, 7, 2, 6, 4, 8; an industry standard approach [8],
- Sequence-2: 1, 2, 3, 4, 5, 6, 7 and 8; as per experimental testing [7].

Each bolt was tightened by increasing the torque in four increments, i.e., 210, 310, 400 and 505 Nm, as per bolt tightening Sequence-1, with copper slip lubricant applied on the threads of all bolts, as recommended per the industrial standard [8]. After the last torque load (505 Nm) application, as per Sequence-1, all bolts were then tightened, as per Sequence-2, to achieve higher pre-load values in the bolts. These pre-load values are recommended by the gasket suppliers to achieve proper gasket seating stress and control of gasket crushing. The maximum achieved average strain in each bolt at the applied torques is plotted in Figure 5. From initial strain results, it was observed that the maximum recommended torque applied could only achieve 30 $\sim$ 35% stress of the yield stress of the bolt material. This is concluded very low, resulting in bolt relaxation during bolt up and leakage during conditions of operation [7]. Although applied pre-loads avoid gasket crushing, at these pre-load values, maximum stress, close to the yield stress of the flange material at certain locations, is observed around the flange hub fillet. This is concluded to be due to the flange rotation [7].

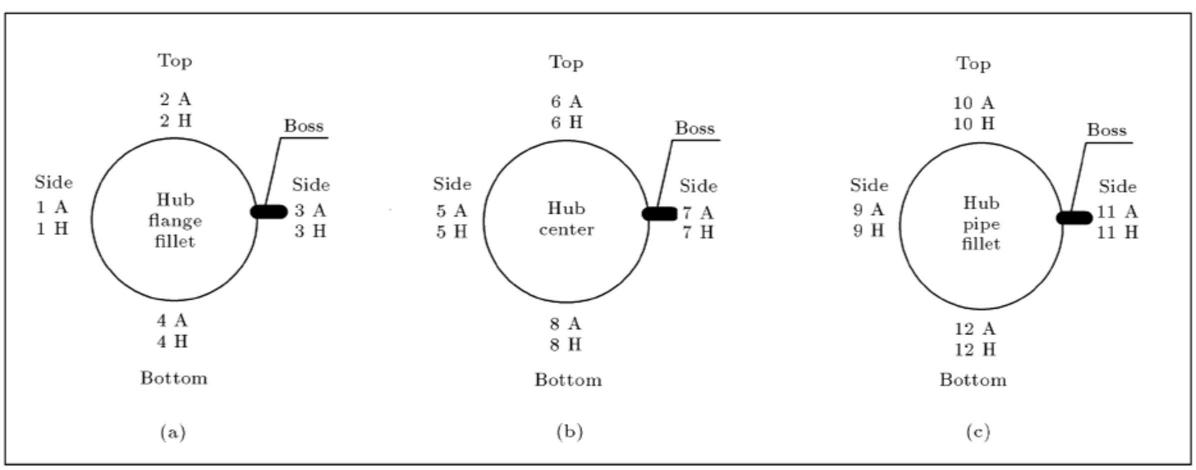

Figure 4. Strain gauging location at: (a) Hub-flange fillet, (b) Hub center, (c) Hub-pipe fillet.

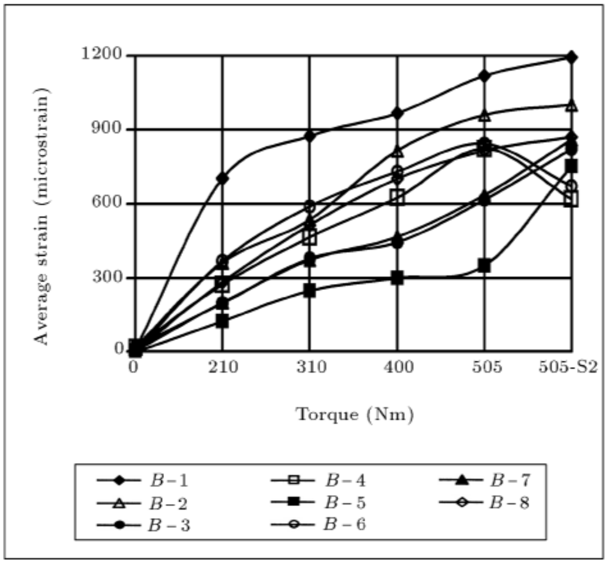

Figure 5. ANSI joint-torque vs avg. bolt strain. (S = Sequence, G = Gasket, S1 = 15372648, S2 = 12345678.)

During joint assembly, even when using the same bolts, set-up, technicians, lubricant and calibrated torque wrench, the stress behavior of each joint was marginally different. In addition, the joint assembly was undertaken in a very controlled environment and such a controlled loading cannot be ensured in the actual field. To tighten the joint, two technicians were engaged in joint assembly, which was clamped in the ground to avoid rotation (Figure 6).

After joint assembly, a test rig was filled with hydraulic fluid and the loading and unloading with internal pressure was applied in increments of 5 MPa using a hydraulic pump. Strains and internal pressure were recorded in the data logging system. Internal pressure loading was applied equivalent to the design pressure of 153 bar (15.3 N/mm^2) and proof test pressure of 230 bar (23 N/mm^2).

Figure 6. Arrangement for gasketed flange joint assembly.

RESULTS AND DISCUSSION

Stress Variations During Bolt up and Operating Conditions

Stress variation results, calculated from strains recorded at various strain-gauged locations during bolt up and under operating conditions, are discussed below. Strains were converted to principle stresses, both in the axial and hoop directions, using the following expressions, as two strain gauges were attached at one location, i.e., in an axial and hoop direction:

$$\sigma_1 = \frac{E}{1 \quad \nu^2} \left(\sigma_1 + \nu\sigma_2\right),$$

$$\sigma_2 = \frac{E}{1 \quad \nu^2} \left(\sigma_2 + \nu\sigma_1\right).$$

Hub-Flange Fillet

During bolt up, a rapid stress increase was observed at 505 Nm torque, as per bolt tightening Sequence-2, with a maximum stress variation of 40 N/mm^2 in the axial direction and 35 N/mm^2 in the hoop direction, along the top, bottom and side locations, whereas almost no stress variation was observed under operating conditions up-to the allowable working pressure of 15.3 N/mm^2 (153 bar) (Figure 7). The maximum stress variation (140-215 N/mm^2), calculated under operating conditions, was close to, but less than, the allowable yield stress (248 N/mm^2) of the flange material.

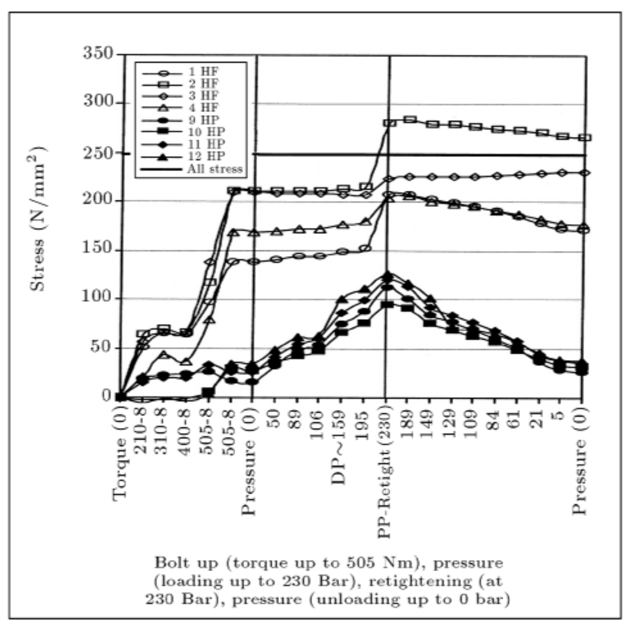

Figure 7. Maximum principle stress (axial) variation at hub-flange fillet and hub-pipe fillet during tightening, loading and re-tightening of gasketed flange joint (HF = Hub Flange, HP = Hub Pipe).

A more detailed flange stress variation during bolt up for each bolt, as per bolt tightening Sequences-1 and 2, is plotted in Figure 8. During tightening Sequence-1, almost the same stress variation pattern is observed along all the locations for all torque values (210-505 Nm). The maximum stress variation, of 70-140 N/mm^2 between the top and side locations and 45-55 N/mm^2 between the top and bottom locations, was observed during bolt-5 tightening for each torque value. After tightening the first four bolts (1, 5, 3, 7), a sudden stress decrease, converging to one point, was observed when bolt-2 was tightened, which, again, increased during bolt-4 tightening. Minimum stress was achieved while tightening the last bolt at all the locations. During Sequence-2, a stress variation of 120 N/mm^2 was observed until bolt-6 tightening, which decreased to 40 N/mm^2 between the top and bottom locations and 73 N/mm^2 between the sides. At the highest bolt torque value of 505 Nm, stress variation (highest for Sequence-2) is obvious for both applied bolt tightening sequences.

During tightening of the bolts at the top and bottom locations, stresses at the side locations were relaxed and vice versa. Although at all locations the stresses were tensile, stress variations showed stress relaxation. Each time, for the next higher torque values, stress was at maximum close to the bolt and at a minimum at an angle of 90°. However this varied continuously as each bolt was tightened. From stress results at the same locations for the two flanges assembled, stress variation, due to unavoidable flange rotation, was observed.

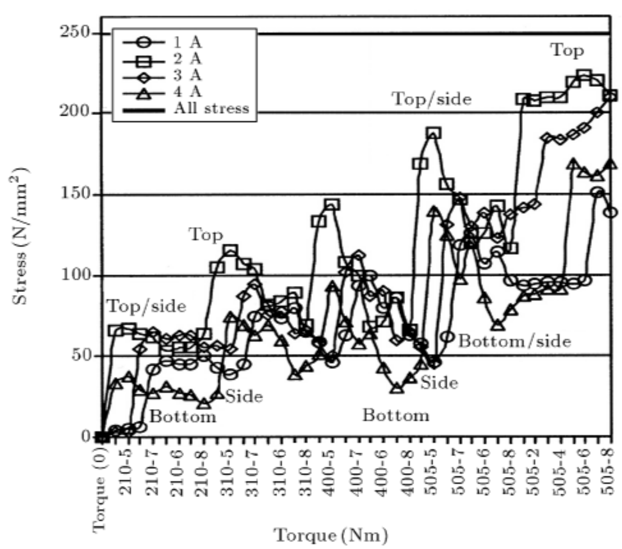

Figure 8. Maximum principle stress (axial) variation at hub-flange fillet during bolt up of gasketed (first as per Sequence-1 and then as per Sequence-2 for last tightening pass). Top, bottom and side are referenced to the locations shown in Figure 4.

Hub-Pipe Fillet

During bolt up, a rapid stress increase is observed at 505 Nm torque, as per bolt tightening Sequence-2, with a maximum stress variation of 15 N/mm^2 (much less than the allowable stress) along the top, bottom and side locations. An almost linear stress increase (50-85 N/mm^2) resulted under operating conditions up to the allowable working pressure of 15.3 N/mm^2 (153 bar, Figure 7).

A more detailed flange stress variation during bolt up for each bolt, as per bolt tightening Sequences-1 and 2, is plotted in Figure 9. During bolt tightening Sequence-1, almost the same stress variation behavior was observed along all the locations. Both tensile and compressive (sinusoidal stresses variation) patterns showed relaxation. This is concluded to be due to flange rotation and any possible movement of the flange assembly in an axial direction, as the assembly was placed on a frictionless saddle with one end fixed to the ground during joint tightening. Each time, for the first three bolts (1, 5, 3), tightening as per Sequence-1, positive axial stress variations of 60, 55, 130, 155 N/mm^2 between the top, bottom and side locations were recorded, which reversed at bolt-7 for the next four bolts (2, 6, 4, 8) and decreased to 40-50 N/mm^2. During Sequence-2, a stress variation of 15 N/mm^2 was observed, without stress reversing.

Hub-Centre

Maximum axial (60-75 N/mm^2) and hoop (80-98 N/mm^2) stress variations were observed during conditions of operation (Figure 10).

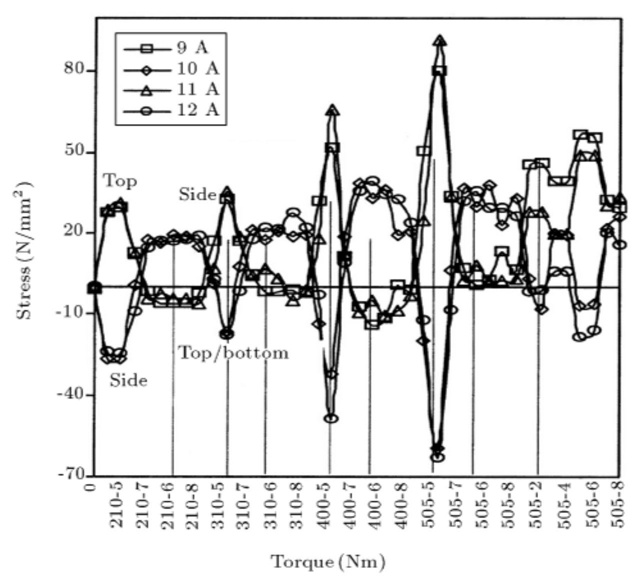

Figure 9. Principle stress variation in axial and hoop directions at hub-pipe fillet of gasketed flange during bolt up and operating condition.

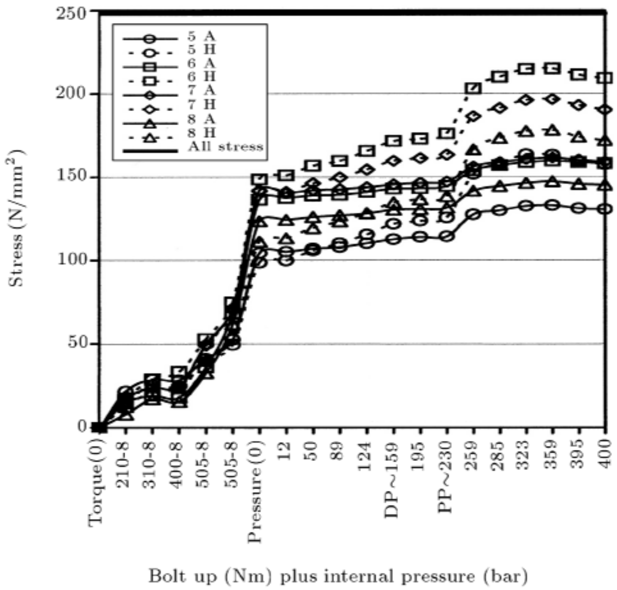

Figure 10. Principle stress variation in axial and hoop directions at hub-center of flange during bolt up and operating condition.

Pipe Centre

Maximum axial (0-28 N/mm^2) and hoop (40-50 N/mm^2) stress variations were observed during conditions of operation.

Based on the above results, it is concluded that in an ideal joint, with symmetrical dimensions, stress variation at all locations around the flange should be the same. However, it was not even identical in the pipe section during bolt up, which is concluded to be due to flange rotation and possible movement of the assembly as it was placed on the frictionless saddle. Stress variation patterns at the hub-flange fillet were significant, as the magnitude of some stresses is close to the allowable stress of the flange material. The conclusion is that, no matter how much care is taken, yielding at the hub flange fillet cannot be avoided when using raised face gasketed joints.

Retightening

Re-tightening is a common practice in actual applications for the gasketed flanged pipe joints to control leakages. Re-tightening of the joint was carried out, as per Sequence-1, when the joint was pressurized up to the proof test pressure of 230 bar (23 N/mm^2). The resulting increase in axial stress at the hub-flange fillet was surprisingly high (15-60 N/mm^2), even though the torque in the bolts was applied very smoothly and carefully without any sudden jerks. After unloading, a residual stress of about 12-45 N/mm^2 and 0-10 N/mm^2 was observed at the hub flange and hub-pipe fillet locations (Figure 7). Maximum stress (280 N/mm^2) was calculated at the top location, which is more than the allowable stress of the flange material. This shows that the re-tightening of the joint during operation adds to the effect of flange straining or yielding. After unloading, the joint was checked for any bolt relaxation and bolt-5 was found to be relaxed, while the remaining were found to be reasonably tight. Re-tightening was undertaken again after unloading, as per Sequence-2, before loading again. A small relaxation was observed in bolts 5, 6, 7 and 8, with bolt-5 being the most relaxed.

Based on the results discussed above, it is concluded that in actual practice, the effect of retightening is not understood, as the main concern is to minimize any leakage by further tightening. This may result in some temporary control, but, thereafter, more severe leakage results and, ultimately, the joint requires replacement. After unloading, bolt relaxation was confirmed and, even after retightening, as per Sequence-2, relaxation remained. This is concluded to be due to the yielding of the flange, providing an additional effect on relaxation of the joint during bolt up and any re-tightening (especially using hammering) makes the situation worse. The effects of re-tightening the gasketed joint should be properly understood, for better or worse, as it is not mentioned in any code and is under conflict, as mentioned in the European Sealing Association document [9].

Quality of Bolts, Proper Tooling and Tightening Sequence

During the experimental work, the bolts used were of good quality with a proper lubricant, as no thread or bolt failure was observed after using them many times. Bolt failures were observed to be due to poor quality with improper surface treatment during the industrial survey and an extra effort to tighten the bolts to the required pre-load level was claimed to be made [10]. During experimental study, a calibrated torque wrench was used with proper sockets etc. and a calculated torque was applied to each bolt to avoid stress variations. Such cannot be guaranteed in industrial practices where hammering is commonly applied to tighten the bolts. Even by using bolt tightening Sequences 1 and 2 and by applying torque in a number of passes, the stress variations observed in the joint components were obvious. In addition, stress variations in the bolts show bolt bending, which is concluded to be due to flange rotation and gasket flexibility.

CONCLUSIONS

From stress variation results in the bolts and flanges, the dynamic mode of load in the gasketed joint during bolt up, operating conditions and retightening are concluded to result in joint failure. To control stress variation in the flange joint components, it is concluded that bolt quality, with proper surface treatment, proper tooling and a correct tightening sequence play a major role. Re-tightening is concluded to have the worst effect in straining the flange during operation and should be avoided.

NOMENCLATURE

G	gasketed
A	axial
H	hoop
E	Young's modulus of elasticity
ν	Poisson's ratio
σ_1, σ_2	principle stresses
HF	hub flange
HP	hub pipe

REFERENCES

1. Webjörn, J. "Flange design in Sweden", *Proceedings of the Petrochemical Mechanical Engineering Conference, American Society of Mechanical Engineers*, Philadelphia, PA, September 17-20, Paper 67-PET-20 (1967).
2. Webjörn, J. "An alternative bolted joint for pipework", *Proc. Instn. Mechanical Engineers*, **203**, pp 135-138 (1989).
3. Abid, M. and Nash, D.H. "An experimental and analytical study of conventional (ANSI) and compact flange (VERAX) subjected to different singly and combined loading", Report Submitted to Shell Expro UK Ltd 1998 and European Pressure Equipment Research Council (EPERC) (1999).
4. Abid, M., Nash, D.H. and Webjorn J. "The stamina of non-gasketed, flanged pipe connections", *Fatigue 2000, Conference on Fatigue and Durability Assessment of Materials, Components and Structures*, Robinson College, Cambridge, UK (April 2000).
5. *British Standard BS1560: Section 3.1* "Circular flanges for pipes, valves and fittings", BSI, London, UK (1989).
6. *British Standard BS7076: Part 1* "Dimensions of gaskets for flanges to BS1560; specifications for non-metallic flat gaskets", BSI, London, UK (1989).
7. Abid, M. "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 (2000).
8. *Design and Engineering Practices; Bolt Torquing and Tensioning*, ES/090 Rev:1, Shell UK Exploration and Production (1993).
9. European Sealing Association, "Guidelines for safe seal usage-flanges and gaskets", Report No. ESA/FSA 009/98, pp 1-40 (Sept. 1998).
10. Abid, M. and Nash, D.H. "FMEA of gasketed and non-gasketed bolted flanged pipe joints", *Journal of Engineering Integrity Society*, **17**, pp 14-21 (March 2005).
11. *ASME Boiler and Pressure Vessel Code, Material Properties*, Section II, Part D., American Society of Mech. Eng., New York, USA (1998).