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Structural strength: Gasketed vs non-gasketed flange joint under bolt up and operating condition

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Abstract

This paper presents results of an experimental study of the behaviour of the stress variation at the flange of a gasketed and non-gasketed flanged pipe joints during both the bolt up (pre-loading) and operating (internal pressure loading) conditions. Stress variations showing flange yielding, flange rotation, effects of joint tightening sequence, identification of the mode of response to loading (static or dynamic) is discussed. In addition the effects of re-tightening, importance of high quality bolting with proper surface treatment and use of proper tooling are also discussed.

Keywords: Gasketed; Non-gasketed; Flanged pipe joints; Test rig; Stress variation; Re-tightening

1. Introduction

The use of bolted flanged pipe joints is common place in the pressure systems industry. However, with the need for more onerous service duties, typically as found in the oil and gas exploration industry, there is an increasing requirement for higher operational pressures and temperatures as the industry seeks to go deeper and further in the search for resources. Previous work by Abid et al. (2000, 2002, 2003), Webjörn, 1989 has indicated that a 'dynamic mode', as opposed to 'static mode' governs in gasketed pipe joint, during bolt up and operating conditions. 'Dynamic' here represents a situation where the flange faces move and rotate relative to one another resulting in a change in bolt load during operation. Such situations occur when a gasket element is present. Due to this, gasketed joints are prone to leakage, even after careful

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Nomenclature

A	axial
H	hoop
G	gasketed joint
N	non-gasketed joint
HF	hub flange fillet
HP	hub pipe fillet
HC	hub centre
DP	design pressure
PP	proof test pressure
E	Young's Modulus of elasticity
v	Poisson's ratio
σ_1, σ_2	principle stresses

pre-loading. An initial deformation produced in the flange at hub flange intersection, provides alignment problem for gasket crushing and bolt bending, resulting in leakage and eventual joint failure. This problem becomes apparent when subjected to operating conditions and even worst when subjected to a combination of loading conditions. The inclusion of the flexible gasket element between the flanges in a gasketed joint leads to a continually varying 'dynamic' situation. This effect become worst by adopting procedures such as hammering and flogging, and re-tightening which, damage not only the flange joint but also the equipment, to which these are attached. Removal of the gasket element changes the situation to a static loading regime; however this may introduce concerns about minimising leakage. A 'static mode' is defined as no significant movement of the flange faces with a change in the bolt load.

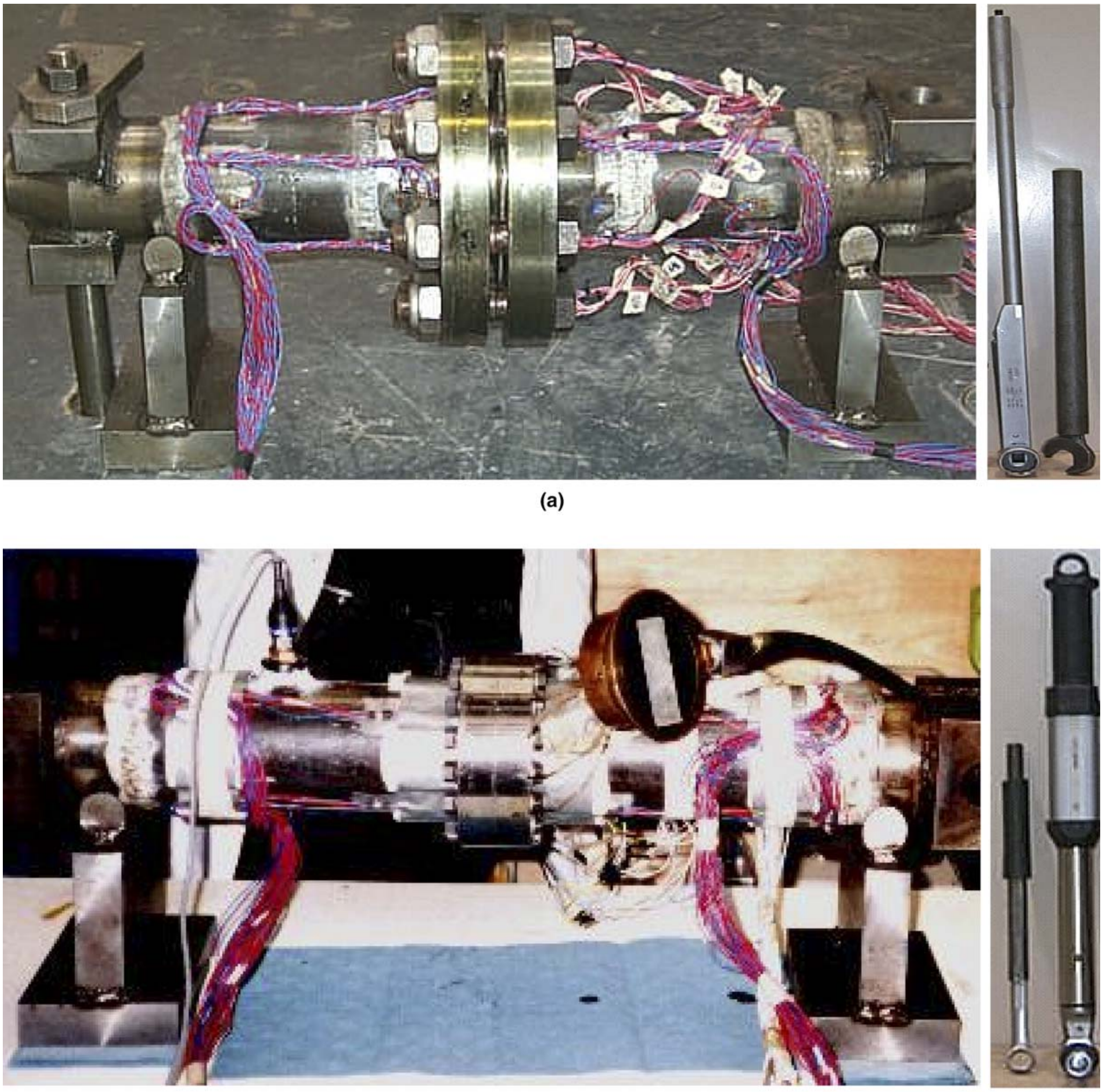
It is recognised that amongst all the constitutive components of a bolted joint assembly, the flange ring is considered to be very strong, as it is generally comprised of thick steel material. The performance behaviour of the flange ring is highlighted as significant for a successful joint as any yielding at flange hub leads to joint failure. In the light of continued problems associated with gasketed joints, highlighting their strength and sealing capabilities, the present comparative study is carried out in detail for a non-gasketed joint (Abid and Nash, 2003) as an alternate, which has better joint strength and sealing performance with a substantial reduction in size and weight. Although a few analytical results are available in literature highlighting the stress variation behaviour in flanged joint during bolt up and operating conditions, no specific experimental results are available. To reduce stress variation for improved joint strength, importance of the use of proper tooling, high quality gasket and bolting with proper surface treatment is also discussed.

2. Experimental programme

A series of experiments using different gasketed and non-gasketed flange joint assemblies are undertaken to examine flange behaviour during joint pre-loading, operating condition and re-tightening.

2.1. Flange type, size and tools

A 4 in. nominal bore, class 900# gasketed and an equivalent non-gasketed (with positive taper angle 0.03° on flange surface and elliptical hub (Abid, 2000; Nash and Abid, 2000) flange joint is selected and



(a)

(b)

Fig. 1. Flange joint assembly with tools: (a) gasketed and (b) non-gasketed.

suitable test rigs are made. For all tests, the same pair of gasketed flanges with three different gaskets of same dimension, same properties and same material is used in assembly to examine variability in supplied gaskets and its effect on joint's behaviour. Similarly, three non-gasketed joint assemblies with and without a secondary seal ring are used. The gasketed and non-gasketed joint assemblies and tools used to make assemblies are shown in Fig. 1a and b respectively.

2.2. Strain gauging and instrumentation

Quarter bridge circuits are made with the data-logging system for strain measurements.

Flanges: Four pairs of strain gauges of 120Ω resistance are attached at hub-flange fillet, hub centre (elliptical portion for non-gasketed flange) at an angle of 90° to measure axial and hoop strains and to observe the behaviour of the flange at the top, bottom and side locations during loading and unloading. On gasketed flange strain gauges are also attached at hub-pipe fillet. The locations of the strain gauges on gasketed and non-gasketed flanges are shown in Figs. 2a, b and 3 respectively.

Bolts: Four strain gauges of 350Ω at an angle of 90° on the shaft of each bolt of gasketed joint and two strain gauges of 350Ω at separation angles of 180° on the shaft of each bolt for a non-gasketed joint are placed to observe any bending.

2.3. Flange joint assembly and tests

Hand-tightening methodology (being the first and the most economical choice of assembly) is adopted to make joint assemblies. An calibrated torque wrench for gasketd joint, whereas, electronic torque wrench and special ring spanners with long handles (to achieve correct specified pre-load with a little additional effort) for non-gasketed joint are used.

Bolt tightening sequence: To ensure that a proper pre-load is achieved in a bolted joint, the importance of proper bolt tightening sequence is also highlighted in this study.

Gasketed joint: For the gasketed joint, following two bolt tightening sequences are adopted;

- Sequence-1 1, 5, 3, 7, 2, 6, 4, 8; an industry standard approach (ES/090, 1998)
- Sequence-2 1, 2, 3, 4, 5, 6, 7 and 8; as per experimental testing (Abid, 2000)

Each bolt is tightened by increasing 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 the bolts recommended as per industrial standard (Abid, 2000). After last torque load application (505 Nm), as per Sequence-1, all the bolts are also 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 control gasket crushing and achieve required gasket

Fig. 2. Strain gauging at: (a) gasketed flange and (b) non-gasketed flange.

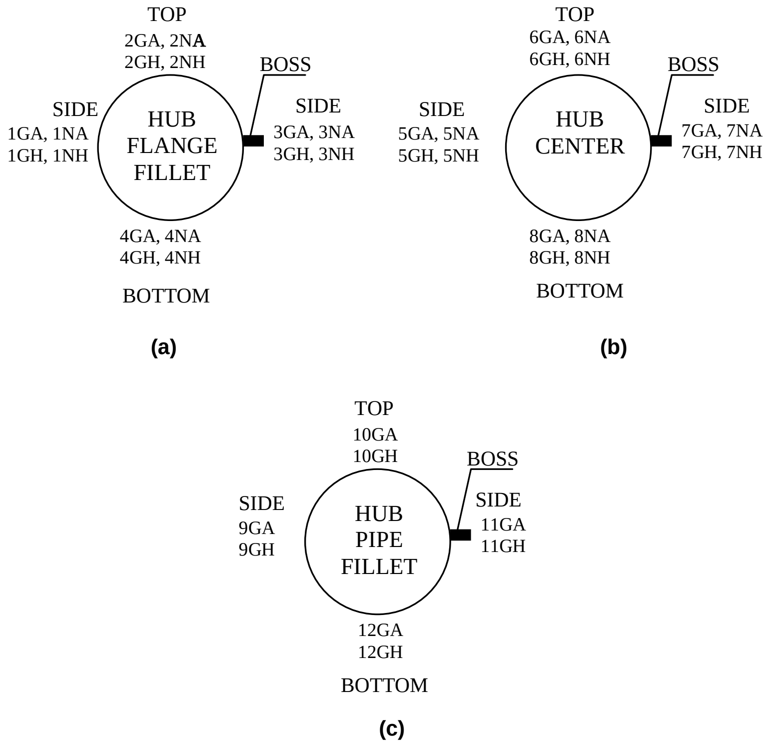


Fig. 3. Strain gauging location at: (a) hub-flange fillet, (b) hub center and (c) hub-pipe fillet.

seating stress. From the initial strain results, it is observed that maximum recommended torque applied could only achieve 30–35% pre-stress of the yield stress of the bolt material. This is concluded very low, resulting in bolt relaxation during bolt up and leakage during operating conditions. Although these preloads avoid gasket crushing, but still provide stresses close to the yield stress of flange material at certain locations around the flange hub fillet due to flange rotation (Abid, 2000).

During joint assembly, using the same bolts, the same set-up, same technicians, same lubricant, and calibrated torque wrench even then, stress behaviour of each joint is marginally different. In addition, joint assembly is made in a very controlled environment, and such controlled loading cannot be ensured in actual field. As it is difficult to tighten the joint, therefore, two technicians are engaged whilst the assembly is held and fixed in the ground to avoid it from rotation, Fig. 4.

Non-gasketed joint: For a non-gasketed joint, with a large number of bolts, certain sequences can be adopted, e.g. firstly four bolts each at 180° to each other and then every second bolt or all bolts pairs located at 180° positions. It is noted for the non-gasketed joint that the 'clockwise' sequence could also be used as there is no gasket. However it is preferred and recommended that the proper sequence for sixteen bolts of a typical joint as 1, 9, 5, 13, 3, 11, 7, 15, 2, 10, 6, 14, 4, 12, 8, and 16 be used. Both bolt tightening sequences are adopted during joint assembly tests and no recorded difference in strain measured is observed.

After the joint is successfully assembled, a series of tests are undertaken with; (i) a design pressure of 153 bar (15.3 N/mm^2), (ii) a proof test pressure of 230 bar (23 N/mm^2) and (iii) higher pressure which is 2.6 times the design pressure i.e. 400 bar (40 N/mm^2). Strains are recorded and subsequent stress calculations are performed.

Fig. 4. Arrangement for gasketed flange joint assembly.

3. Stress variations results during bolt up and operating conditions

Stress variation results calculated from measured strains recorded at various strain-gauged locations (top, bottom and side) during bolt up and operating conditions are reported. Strains are converted to principle stresses both in the axial and hoop directions using following expressions as two strain gauges are attached at a single location, in axial and hoop direction

$$\sigma_1 = \frac{E}{1 - v^2}(\sigma_1 + v\sigma_2)$$
$$\sigma_2 = \frac{E}{1 - v^2}(\sigma_2 + v\sigma_1)$$

3.1. Hub-flange fillet

Gasketed joint: Stress variation in the axial and hoop directions during bolt up (for bolt-8 of each tightening pass) and operating condition is plotted in Fig. 5. A rapid stress increase is observed at 505 Nm torque as per bolt tightening Sequence-2, with maximum stress variation of 40 N/mm^2 in the axial and 35 N/mm^2 in the hoop directions along top, bottom and side locations. Almost no stress change observed under operating condition up to the design pressure. Maximum stress calculated (140–215 N/mm^2) at operating condition is close to but less than the allowable yield stress (248 N/mm^2) of the flange material.

At the proof test pressure, a maximum axial and hoop stress increase of 8 N/mm^2 and 15 N/mm^2 observed with maximum stress variation of 62 N/mm^2 along axial direction between top and bottom locations. After re-tightening the joint at proof test pressure, a maximum axial stress increase of 76 N/mm^2 observed between top and bottom locations. Similarly at the highest test pressure of 400 bars, a maximum axial and hoop stress increase of 142 N/mm^2 and 61 N/mm^2 observed with maximum stress variation of 23 N/mm^2 along hoop direction between top and side flange locations. The overall maximum stress (224–281 N/mm^2) after re-tightening along side locations is close to and more than where as along top and bottom (205–208 N/mm^2) locations is less than the allowable stress. The overall maximum stress

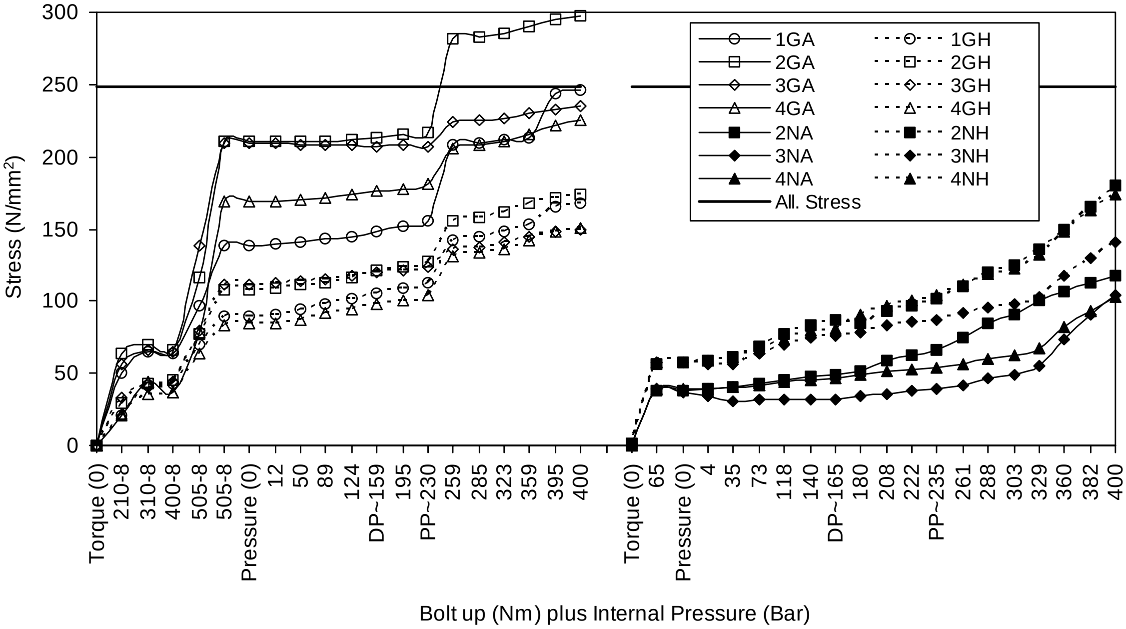


Fig. 5. Principle stress variation in axial and hoop directions at hub-flange fillet during bolt up and operating condition for gasketed and non-gasketed flange joint assemblies.

(248–298 N/mm^2) at 400 bars along top and side location is more than where as along side and bottom locations (225–235 N/mm^2) is close to the allowable stress.

Non-gasketed joint: Principle stress variation results are plotted in Fig. 5. A maximum principle axial (38 N/mm^2) and hoop stress of (57 N/mm^2) is observed in the flange during bolt-up. At the design pressure, a maximum axial and hoop stress increase of 10 N/mm^2 and 24 N/mm^2 is observed with the maximum stress variation of 17 N/mm^2 along axial direction between top and side flange locations.

At the proof test pressure, a maximum axial and hoop stress increase of 28 N/mm^2 and 47 N/mm^2 observed with maximum stress variation of 27 N/mm^2 along axial direction between the top and side flange locations. Similarly at the highest test pressure of 400 bars, a maximum axial and hoop stress increase of 80 N/mm^2 and 123 N/mm^2 observed with maximum stress variation of 39 N/mm^2 along hoop direction between top and side flange locations. The overall maximum stress measured (180 N/mm^2) is less than the allowable stress (248 N/mm^2) of the flange material.

A more detailed flange stress variation phenomena for both the joints during bolt up for different tightening sequences is plotted in Fig. 6. In the non-gasketed joint, during any tightening sequence same pre-stress pattern with no stress variation in all the bolts of the joint observed. In gasketed joint, during tightening Sequence-1, almost, same stress variation pattern observed along all the locations for all torque values (210–505 N/mm^2). Maximum stress variation of 70–140 N/mm^2 between top and side locations and 45–55 N/mm^2 between top and bottom locations observed during bolt-5 tightening for each torque value. After first four bolts tightening (1, 5, 3, 7), a sudden stress decrease converging to one point observed when bolt-2 is tightened, which again increased at bolt-4 tightening. The minimum stress is achieved while tightening the last bolt at all the locations. During Sequence-2, stress variation of 120 N/mm^2 observed until bolt-6 tightening, which decreased to 40 N/mm^2 between top and bottom locations and 73 N/mm^2 between 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, the

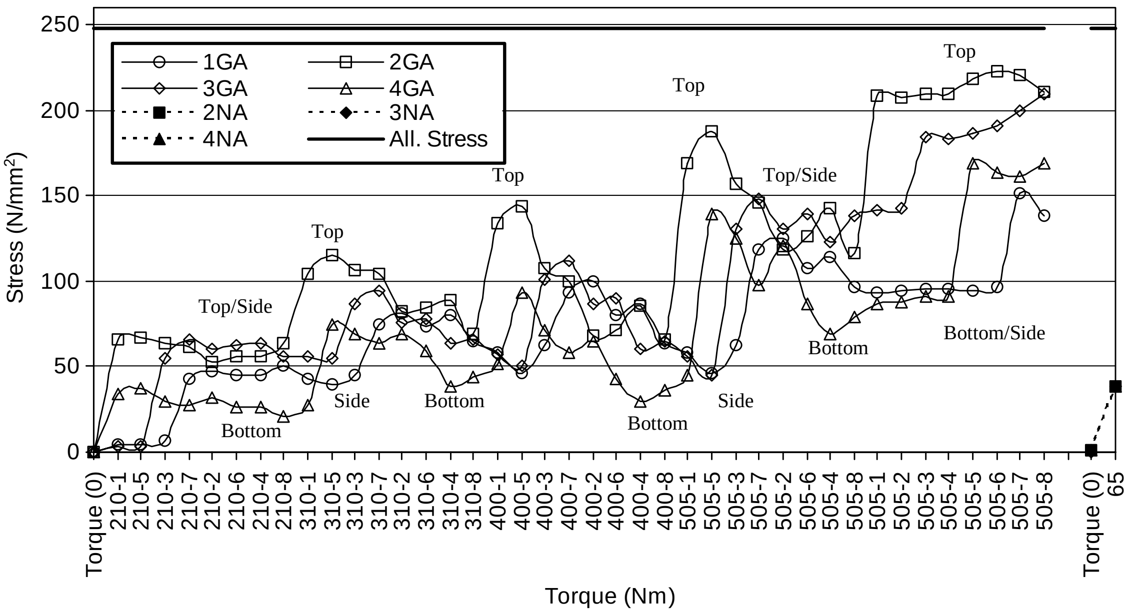


Fig. 6. Maximum principle stress (axial) variation at hub-flange fillet during bolt up of gasketed (first as per Sequence-1 and then as per Sequence-2 for last pass) and non-gasketed flanges (for any sequence adopted). Top, bottom and side are referenced to the locations shown in Fig. 3.

stresses at side location are relaxed and vice versa. Although at all the locations stresses are tensile, however stress variations showed stress relaxation. Each time, for next first higher torque values, the stress is maximum close to the bolt and minimum at an angle of 90°, however this varied continuously as each bolt is tightened. From stress results at the same locations for the two flanges assembled, stress variation due to unavoidable flange rotation is observed.

3.2. Hub-centre

Gasketed joint: From Fig. 7, almost same principle stress variation pattern as for hub-flange fillet is observed with the difference that a relatively less axial (141 N/mm^2) and hoop (148 N/mm^2) stress recorded during bolt up. During operating conditions up to 400 bars, axial and hoop stress variation of 50 N/mm^2 and 33 N/mm^2 around flange observed. Maximum stress measured at design, proof test, re-tightening and high pressure is 171, 176 and 209 N/mm^2, which is less than the allowable stress of the flange material.

Non-gasketed joint: Principle stress variation results are plotted in Fig. 7. A maximum principle stress (axial and hoop) increase of 33 N/mm^2 observed in the flange during bolt up. At the design pressure, a maximum axial and hoop stress increase of 26 N/mm^2 and 40 N/mm^2 is observed with the maximum stress variation of 5 N/mm^2 along axial direction between top and side flange locations. At the proof test pressure, a maximum axial stress increase of 41 N/mm^2, and a hoop stress increase of 62 N/mm^2 observed with maximum stress variation of 10 N/mm^2 along the axial direction between the top and side flange locations. At high pressure, a maximum axial stress increase of 130 N/mm^2 and a hoop stress increase of 151 N/mm^2 are observed with maximum stress variation of 25 N/mm^2 along hoop direction between top and side flange locations. The overall maximum stress (184 N/mm^2) observed is less than the allowable stress (248 N/mm^2) of the flange material at this location.

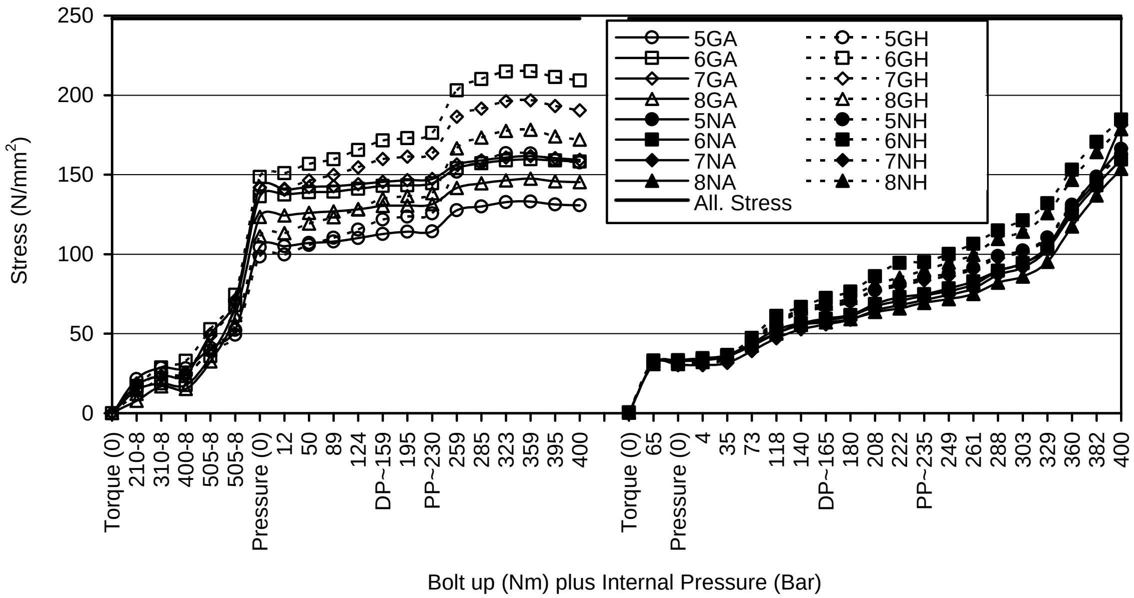


Fig. 7. Principle stress variation in axial and hoop directions at hub-center of flange during bolt up and operating condition for gasketed and non-gasketed flange joints.

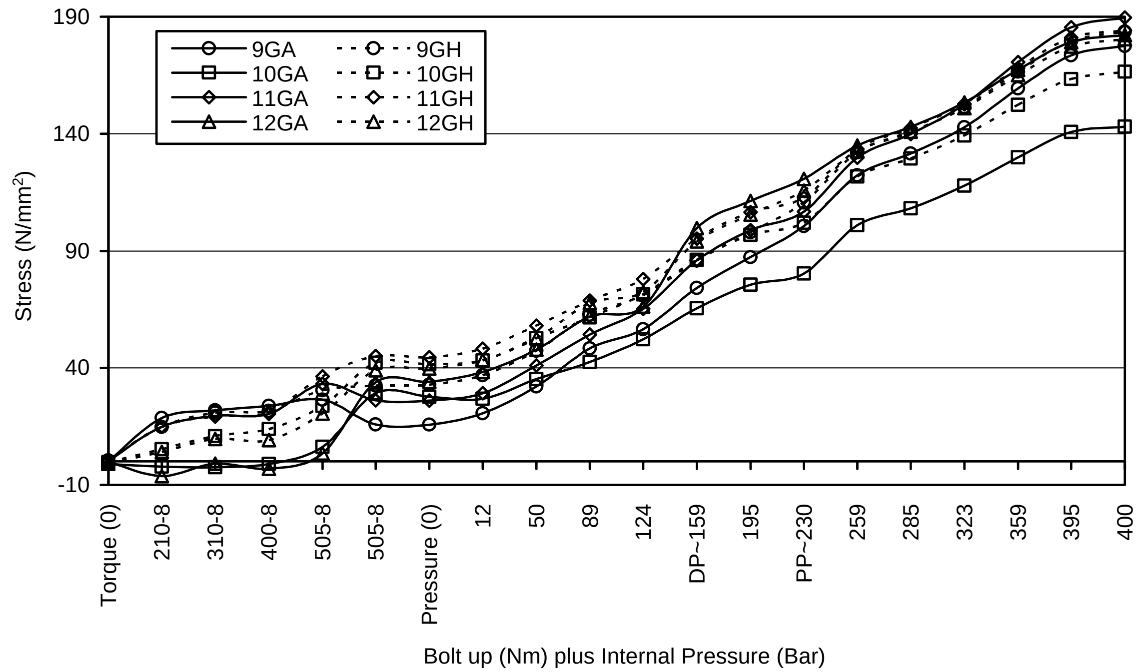


Fig. 8. Principle stress variation in axial and hoop directions at hub-pipe fillet of gasketed flange during bolt up and operating condition for gasketed flange joint only.

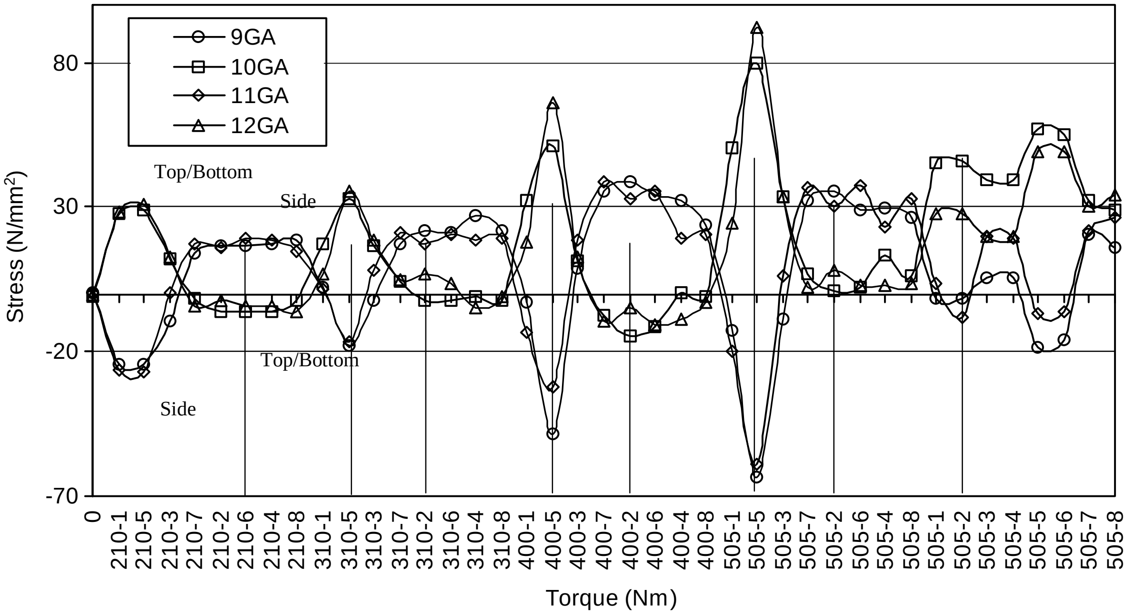


Fig. 9. Maximum principle stress (axial) variation at hub-pipe fillet of gasketed flange during bolt up (first as per Sequence-1 and then as per Sequence-2 for last pass). Top, bottom and side are referenced to the locations shown in Fig. 3.

3.3. Hub-pipe fillet of gasketed flange joint assembly only

Stress variation in the axial and hoop directions during bolt up (for bolt-8 of each tightening pass) and operating condition is plotted in Fig. 8. A rapid stress increase is observed at 505 Nm torque as per bolt tightening Sequence-2, with maximum stress variation of 15 N/mm^2 in both the axial and hoop directions along top, bottom and side locations. Almost linear stress increase (50–85 N/mm^2) at design pressure, (65–105 N/mm^2) at proof test pressure and (128–174 N/mm^2) at 400 bars observed. Maximum stress (99, 120 and 190 N/mm^2) measured is less than the allowable stress of the flange material (248 N/mm^2).

A more detailed flange stress variation phenomena during bolt up as per bolt tightening Sequence-1 and 2 for each bolt is plotted in Fig. 9. During bolt tightening Sequence-1, almost, same stress variation behaviour observed along all the locations. Both tensile and compressive (sinusoidal stresses variation) pattern showed relaxation. This concluded due to the flange rotation and any possible movement of the flange assembly in axial direction, as the assembly is placed on the frictionless saddles with one end fixed to the ground during joint tightening. Each time, for first three bolts (1, 5, 3) tightening as per sequence-1, positive axial stress variation of 60, 55, 130, 155 N/mm^2 between top, bottom and side locations 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, stress variation of 15 N/mm^2 observed, without stress reversing.

4. Re-tightening

In actual practice, effect of re-tightening is not realised, as the main concern is to minimise any leakage by further tightening as is found a common practice in actual applications for pipe joint assemblies. This may result in some temporary control but thereafter more severe leakage results and ultimately the joint requires replacement. Effects of re-tightening should be properly understood that, is it, better or worst,

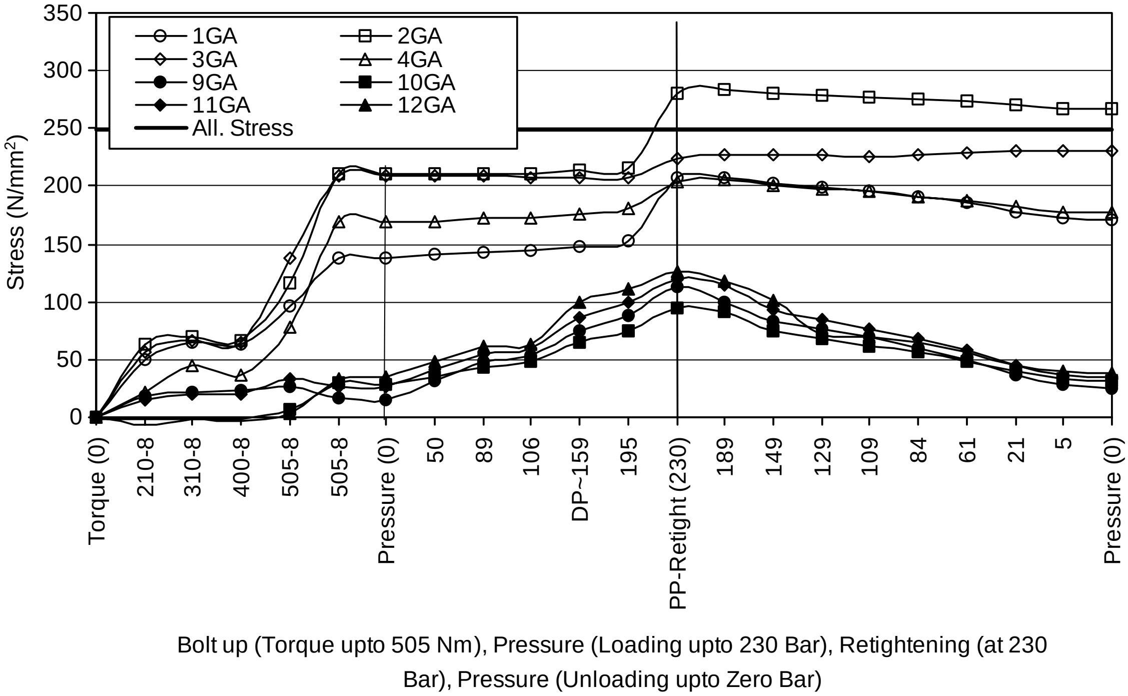


Fig. 10. Effect of re-tightening on maximum principle stress (axial) at hub-flange fillet and hub-pipe fillet of gasketed flange joint.

for the gasketed joint as this is not mentioned in any code and is under conflict as mentioned in European Sealing Association document (European Sealing Association, 1998).

Gasketed joints: Re-tightening of joint is carried out as per Sequence-1 when joint is 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 is surprisingly high (15–60 N/mm^2) even though the torque in the bolts is applied very smoothly and carefully without any sudden jerks. After unloading the flange joint, a residual stress of about 12–45 N/mm^2 and 0–10 N/mm^2 observed at hub flange and hub-pipe fillet locations (Fig. 10). Maximum stress (280 N/mm^2) is calculated at the top location which is more than the allowable stress of the flange material. This shows that the re-tightening of the gasketed joint during operating condition adds to the effect of flange straining or yielding. After unloading, joint is checked for any bolt relaxation and bolt-5 is found almost relaxed, while the remaining found reasonably tight. For new set of test, before loading (pressurizing), re-tightening is again performed as per Sequence-2. A small relaxation is observed in the bolts 5, 6, 7 and 8 with bolt-5 the most relaxed.

Non-gasketed joints: During re-tightening all the bolts are found reasonably tight, so no relaxation concluded.

5. Discussion

5.1. Flange stress and stress variation

In an ideal joint, with symmetrical dimensions, stress variation at all locations around flange should be the same. However, from the results presented herein, the variability is obvious especially for the gasketed joint. This is due to the flange rotation and possible movement of assembly as it is placed on frictionless saddles. The stress variation pattern at hub-flange fillet is significant; however, stress magnitude is close to the allowable stress of flange material. In summary, it appear that no matter how much care is taken; yielding at hub flange fillet cannot be avoided when using raised face gasketed joints. Based on the factors such as dimensional inconsistency, tightening methodology, fitters training, location of installation etc., a proper joint tightening procedure cannot be standardised. Although, with practice on the shop floor the confidence can be achieved which needs a long time for proper training of the personnel, for different sizes and then for different materials using extensive experimental testing. Apart from this all, the most important consideration of gasket flexibility factor cannot be eliminated at all, which is obvious from bolt relaxation, bolt bending and flange rotation (Abid, 2000). For the successful operation of a joint, the 'static mode' is essential, which avoids any fatigue mechanism in the joint, whereas a 'dynamic mode' in gasketed joint exists and is highlighted from this experimental study. However, a gasketed joint made in a very controlled environment can perform successful working, which however cannot be guaranteed and most joint failures observed during and just after pre-loading. In real situations where alignment of pipelines is difficult to ensure, and movement is constrained, flange rotation with more severe stress variation will be obvious, with the tightening methodology such as hammering adopted in the industry.

From principle stress results, it is concluded that, during bolt up and operating conditions, at all locations maximum stress in a non-gasketed joint, lie within the yield strength of the flange material. Whereas, in the gasketed joint, stresses in flange during bolt up and up to the design pressure are close to the yield stress of the flange material at hub-flange fillet (Fig. 11). However, these are more than the allowable stress at proof test pressure, re-tightening and at high pressure of 400 bar (Fig. 12).

5.2. Re-tightening

In the non-gasketed joint, after unloading, bolt relaxation is confirmed, and even after re-tightening as per Sequence-2, relaxation remained. This is concluded due to the yielding of the flange, providing an additional effect to the relaxation of the joint during bolt up and any re-tightening (especially using hammering) makes the situation worst.

However for non-gasketed joints such re-tightening is not required as there is no gasket present and hence no flexibility and no rotation. Re-tightening of bolts is not prohibited as such and may be performed after a certain time if required as a precaution. However, it is the experience of the authors that such

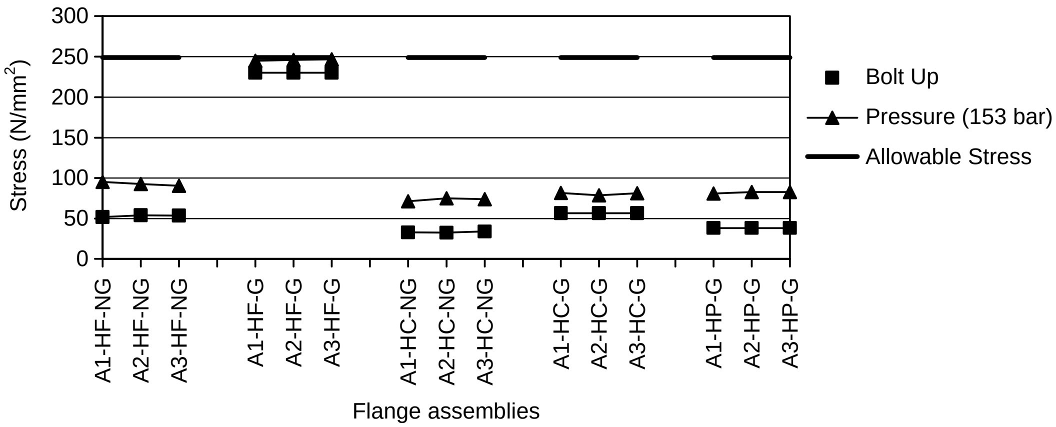


Fig. 11. Principle stresses at design pressure of 153 bar for three gasketed and non-gasketed flange joint assemblies tested. (A1–A3: flange joint assemblies).

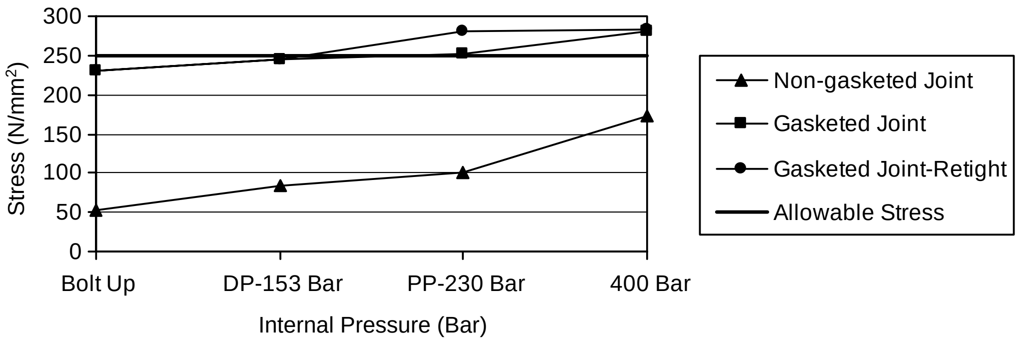


Fig. 12. Maximum principle stresses at hub flange fillet for one of the joint assemblies of both the gasketed and non-gasketed flange joints at design pressures (153 bar), proof test pressure (230 bar) and 2.6 times design pressure (400 bar).

re-tightening of itself does not cause any damage to the bolts or flanges in a non-gasketed joint. The same cannot be said for the gasketed joint.

5.3. Quality of bolts, proper tooling and tightening sequence

Gasketed joint: During experimental work, bolts used are of good quality and great care is taken using proper lubricant. During industrial survey (Abid and Nash, 2005; Abid, 2000), it is noted in some real situations that the bolts are so rusted that they required to be cut to dismantle the assembly for maintenance or repair (as they are exposed to the external environment). In addition, due to the poor quality bolts with improper surface treatment, an extra effort may be required to tighten them to the required pre-load level. With the additional effect of gasket quality and flexibility, as rotation is a must, providing stress variation in flange it is concluded that the bolt quality can have a major impact on the effectiveness of the joint operation, both in terms of bolt strength and fitness for purpose. Good quality bolts with proper surface treatment are concluded an essential for a successful long term joint. Similarly, use of proper tools and tightening procedures (avoiding hammering etc.) to make the joint assembly is recommended to control flange stress variation. In the end, last but not the least, to achieve proper joint pre-loading and control flange stress variation, bolt tightening sequence has a pronounced effect and should be considered as an important parameter.

Non-gasketed joints: As a high initial pre-load is recommended for the non-gasketed joint assembly, good quality bolts and proper tooling is a must (Abid, 2000; Webjörn, 1989). Without such practical considerations, the static mode of behaviour in the joint cannot be ensured. During the experimental work, conventional spanners are also used to tighten the bolts along with the special long handled ring spanner and electronic torque wrenches. It is concluded that when using only conventional spanners, a high initial pre-load cannot be successfully achieved even though experienced and trained fitters are employed. In addition, the use of socket sets is not recommended as this method of loading can give rise to bending in smaller diameter bolts. Furthermore, good quality bolts must be provided as inconsistency in bolt head, nut and spanner dimensions is also a common experience. Care must be taken to avoid bolt head and nut damage and to ensure that the correct high level of pre-load is achieved (Abid, 2000).

6. Conclusions

From results it is concluded that, during bolt up and operating conditions all the stresses in a non-gasketed joint, lie within the yield strength. Whereas, in the gasketed joint, during bolt up and up to the design pressure are close to but at proof test and higher pressures are more than the allowable stress at certain locations. From a consideration of the stress variation, a 'static mode of loading' is concluded in the non-gasketed joint, providing better sealing and joint strength capabilities. Whereas, stress variation (tensile and compressive) at different locations 'dynamic mode of load' in the gasketed joint is concluded, providing poor sealing and joint strength capabilities. Re-tightening provides worst effect for the gasketed joint in straining the flange under operating conditions especially due to the inherent 'dynamic mode of load'. Whereas, for the non-gasket joint, re-tightening helps to certain extent, however due to 'static mode' in the joint, no re-tightening is generally recommended. Practical matters of assembly, tooling and tightening sequence do have an important effect on the successful behaviour of the gasketed flange joint. Especially important is the gasket quality for the gasketed flange joint only and bolt quality with proper surface treatment which must be good to ensure that a high initial pre-load in the bolt is achieved with less effort and is strongly recommended for the non-gasketed flange joint.

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