The Stamina of Non-Gasketed, Flanged Pipe Connections

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Most international design codes for pressure equipment, such as BS 5500, ASME VIII and the new European standard prEN 13455, provide rules and equations to allow gasketed joints to be safely designed for a given internal pressure design and loading. In general they do not take into account the effect of axial plus lateral loads and bending moments acting in conjunction with the main pressure load. This paper is based on two years of experimental and finite element studies[3] of both standard/conventional ANSI (gasketed) and modern VCF (non-gasketed) flanged joints (according to the VERAX Compact Flange System). This has led to a deeper understanding of the requirements for a successful assembly and long term usage. ANSI and VCF joints have been subject to internal pressure, axial and lateral forces, with these loads having been applied both singly and in combinations. Mode of load acting in the joint i.e. static or dynamic has been studied for both kinds of joints. Experimental and analytical results are compared. Some practical considerations on the use of important emerging technology i.e. non-gasketed pipe joints in comparison to conventional gasketed systems are presented. Of fundamental importance is an insight into the mechanism of the bolted joint showing the effect from an external load on a preloaded bolt. It is found that it can be made near zero hence, in a properly built, non-gasketed bolted joint a static mode rules, and therefore the stamina of such a joint is unlimited. This paper demonstrates that the novel system is an efficient and well-engineered alternative to traditional joints.

Introduction

It is well known that flanges have been used for more than three hundred years for joining pipes. They are simple to fabricate and assemble requiring simple tools, moderate skills but a lot of common sense. They can be designed to sustain a variety of fluids for a range of temperatures and pressures and can be made to operate successfully under most conditions.

Early attempts to design flanges were based on a variety of very crude and simplified assumptions, that were altogether unsatisfactory. Typically, flanges were

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assumed to consist of a number of discrete radial beams in bending, completely ignoring the hoop forces. This resulted in extravagant dimensions. In 1927, Waters and Taylor presented a method for calculating flange stresses and deflections, that were in agreement with experimental results. Waters, Westrom, Rosshein and Williams further refined this analysis. When published in 1937, it became known as the **'Taylor and Forge method'** [1]. It still is the basis of the present ASME and BS5500 codes regarding flange design.

There are many different designs of flanged joints available. A flanged joint can perform well for very many years in a particular application, but in a different application, the same joint may perform miserably. Experience from offshore gas installations has made it evident, that the conventional gasketed, pipe joint, has several weaknesses, which can cause problems in service. Having examined the relative literature and after discussing problems faced by the oil and gas industry, it was decided to compare results from testing a standard ANSI gasketed joint with an equivalent alternative, non-gasketed VCF joint (provided by VERAX Ltd in Sweden).
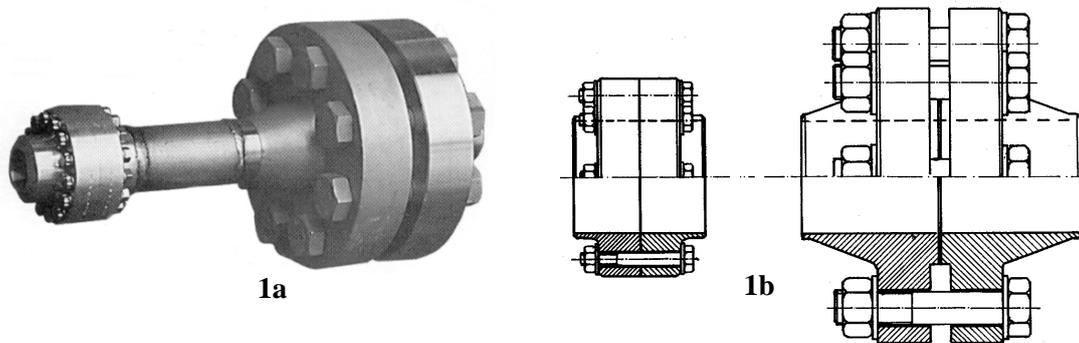

Figure 1 Comparison of VERAX (VCF) Joint and Standard ANSI Joint

The Simple Bolted Joint

Main Features

The elementary bolted joint is detailed in Figure 2. Typically the bolt is resilient and the abutment is solid, with no soft element between the members bolted together. Being an application of **'the principle of the preloaded structure'** there is no significant displacement, one of the bolted members relative to the other, providing that external load does not exceed bolt preload. Webjorn has shown that in such a bolted joint, the force felt by the bolt can be made almost zero [1],[4-6]. This means that such a joint can not possibly fail from fatigue.
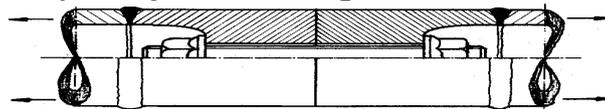
Figure 2 Elementary bolted joint

The Mechanism of the Gasketed Pipe Joint

A conventional pipe joint, using a pair of raised-face flanges, is characterised by forces being transmitted from one flange to another by way of a gasket of some kind. As the bolts in the joint are being preloaded, the flanges rotate about a primary centre, approximately one third of the gasket width from its outside edge, its angle of rotation is proportional to pre-load (see Figure 3). However, when exposed to loads from internal fluid pressure, from an axial/lateral load or from a bending moment, flanges rotate about two secondary centres, under the nut and under the bolt head, respectively. It follows that -- apart from that tensile stress is induced in the fillet between the flange itself and the hub -- high bending stress does form in the fillet under the bolt head and at the first thread near the nut. As much as fluid pressure, loads and moment vary with time, a dynamic mode does rule in such a bolted joint, what sooner or later may result in failure from fatigue.
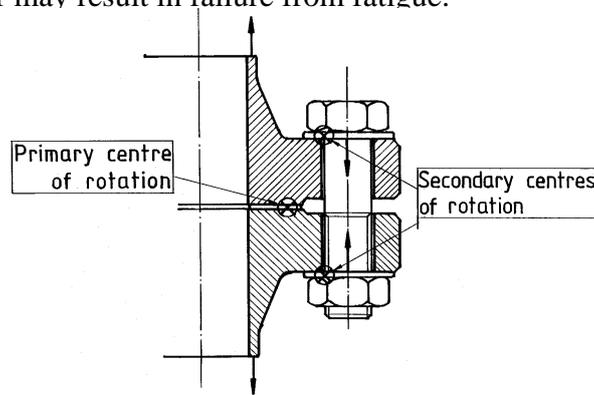

Figure 3 Mechanism of the Gasketed Pipe Joint

Fatigue of the Bolted Joint

In a rigid joint, a properly tightened bolt will not fail from fatigue as it is exposed to static load. Initial bolt tension will stay relatively constant until the external tensile load on the joint does exceed the bolt pre-load. This has also been proved from the experimental finite element studies concluding that, the higher the pre-load, the safer the joint is [6]. The basic principle of design is that the calculated service load must not be allowed to exceed the bolt pre-load. The bolt will experience no appreciable stress variation, and without stress variation, there can be no failure by fatigue, regardless of the number of load cycles on the joint. This is not the case where flexibility is present. Variable stress in screw or bolt fastenings increases with the flexibility of the connected parts. If flexibility is too great, the variable stress present may be high enough to cause eventual fatigue failure of the fastener regardless of the initial bolt pre-load. The greatest single factor that can eliminate cyclic stress variation due to cyclic loading is proper pre-tensioning or pre-loading of the fasteners. Test results indicate that rigid members bolted together by relatively elastic bolts offer the best method to prevent fatigue failure. In order to increase the reliability of a bolted joint, three main areas must be considered:

1. The nature and magnitude of external loads acting on the joint, shall be established, either by detailed computations, actual measurements and experiments or by "guesstimates".

2. A sufficient number of sufficiently strong bolts shall be selected.

3. The layout of the joint shall be such that clamped members are stiff and bolts slender.

The present research has emphasised that a critical factor involved is the achievement at assembly of preload **above** the minimum specified [1],[4-6] that is the figure used in the design calculations. The recognition of this fact has encouraged the development of special electronic controls, which monitor the tightening process and provide very accurate bolt preload. See Figure 4a and Figure 4b.
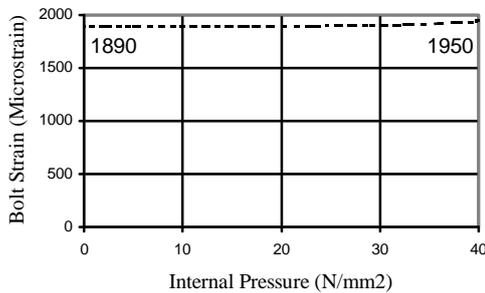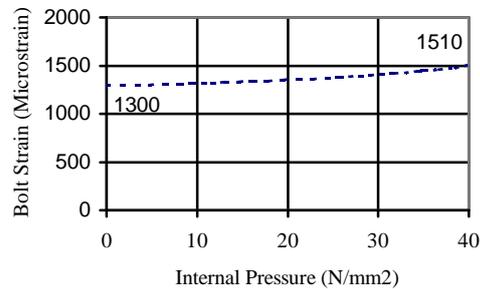
Fig 4a VCF Joint　　　　　　　　**Fig 4b ANSI Joint**

Figures 4a,b Internal Pressure versus Bolt Strain

Experimental Programme

To examine the effect of differing combinations of externally applied load, a test rig was designed to work in tandem with an existing tensile testing machine. Four pairs of strain gauges were attached at hub and at pipe sections at 90° angles. Two strain gauges were attached on bolts for the non-gasketed joint, whereas four strain gauges were attached on bolts of ANSI gasketed joint to examine the bending behaviour of bolts. Pressure loading was applied to the assembled joint via a manually operated hand pump, with a 500 bar capacity. Pressure gauges on the pump and on the assembled test vessels recorded the fluid pressure. Axial tension load was applied via two symmetric parallel shafts loaded by hydraulic cylinders. This tensile load was transferred to the pipe by the use of heavy end plates and a pin-type connector, which located the assembly and the loaded shafts. The end plates are deemed rigid enough to transfer the load from the shafts to the pipe assembly, however, these were strain gauged as a precaution and load levels obtained. Three and four point bending was achieved by the use of the testing machine cross head together with a custom-built load applicator. This arrangement applied load to the upper portion of the joint and was reacted by two frictionless loose saddles that allowed the joint to rotate in the axial plane. The layout of the test rig is shown in Figure 5.

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The technical benefits and drawbacks of the joint styles can be demonstrated by comparing the two main characteristics i.e. joint strength under working pressure and sealing ability. The object or purpose of these tests was to verify these quantities. It was therefore decided to conduct a number of experiments by building an appropriate model and measuring their performance for the VERAX four inch, class 900# joint without an O-ring seal and with an O-ring seal and also one standard ANSI four inch class 900# gasketed joint.

The following loading conditions were selected and applied to all flange styles:

- •Internal pressure only   •Internal pressure plus bending
- •Axial force only   •Internal pressure plus axial force
- •Bending moment only   •Internal pressure plus bending plus axial load

Experimental and Analytical Comparison

Although there are many different flange designs available, all except VERAX use the concept of the gasketed joint. From full 3-D and 2-D finite element studies of both standard gasketed and non-gasketed joints[3], it has been shown, that non-gasketed joints show better behaviour in contact pressure, low stresses at flange hub, flange ring, bolts and flange rotation compared to other designs. Both joints under examination have been subject to loads by internal, axial tension and bending moment, with such loads having been applied both singly and in combinations.

Summary of Results

Bolt pre-loading

The required tightening torque for the VCF joint (65 Nm) was significantly less than for the ANSI joint (505 Nm). When pre-loading the bolts in the ANSI joint, bending was noted in the bolts, as compared to no bending in the VCF bolts. From experiments it is evident that for VCF joint the required pre-load **can be achieved** whereas for ANSI joint it is very difficult to achieve due to gasket and flange rotation.
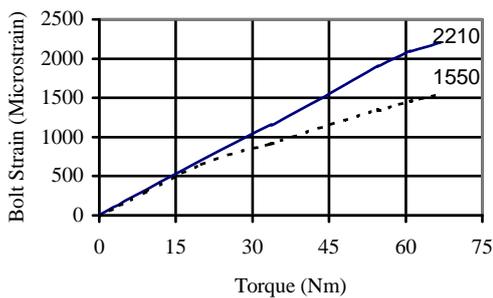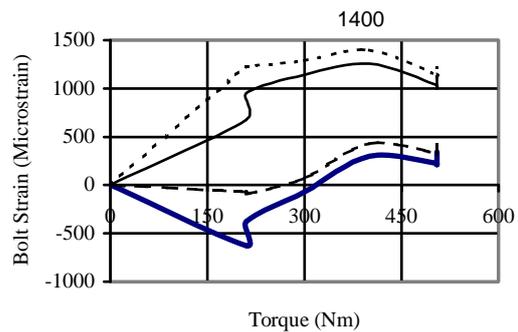
Fig 6a   VCF Joint   **Fig 6b ANSI Joint**

Figure 6a,b Bolt Pre-loading versus Bolt Strain

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Loading by internal pressure only

Up to the maximum proof test pressure, the strain in the ANSI joint bolts was about 10 times as high as in the VCF joint. For a pressure above the proof test pressure, strain in VCF joint bolts went up but was still lower than in the ANSI joint bolts. The higher strain noted in the ANSI joint bolts may be due to bending, which does result from flange rotation. Because of a large number of small bolts, in the VCF joint, various loads become well distributed among the bolts. Up to a proof test pressure of 230bar, the hoop strain at hub of flange was less than at the pipe section, whereas the axial strain was a little higher at the hub than at the pipe section. At a pressure of 400bar (= 1,74 times the proof test pressure), strain at the hub of the flange and also at the pipe section were nearly the same. Axial strain at hub was three times higher than in the pipe section. For the ANSI joint, due to its large dimensions and heavy welding neck, both the axial and hoop strains were lower than in the VCF joint. Overall maximum stress calculated from the maximum strain (either hoop or axial) were less than 2/3rd of the yield stress of flange and pipe material. The maximum stress for the bolts in the both joints was lower than the allowable stress for the bolt.

Bending moment only (vertical load)

The strains measured in the bolts of the VCF joints were lower than those of the ANSI joint, but the maximum stress for the bolts in the both joints was less than the allowable stress for the bolt material. Hoop strains at the hub of a flange, were nearly the same for both joints, whereas the axial strain in the VCF joint were substantially higher (about 3 times). Overall maximum stress calculated from the maximum strain (hoop or axial) were less than 2/3rd of the yield stress of flange and pipe material.

Tensile load only

The strain measured in the bolts was similar for both the joints and the maximum stress for the bolts of both the joints was less than the allowable stress for bolt material. Hoop strains at hub of flange were at the same level, whereas the axial strains in the VCF joints were about four times as high as in the ANSI joint. Strains in the pipe section were almost the same with all the joints. Maximum stresses from the maximum strain (hoop or axial) were less than the 2/3rd of the yield stress of flange and pipe material.

Internal pressure plus tensile load

In this case, the strain measured in the bolts of the ANSI joint was higher than in those of the VCF joint. Hoop strains at the hub of VCF flange were double those of the ANSI flange, whereas the axial strains were very large, about 5 times those of the ANSI joint. The hoop strain at a pipe section of a VCF-joint was twice that in the ANSI joint, whereas the axial strain was nearly the same for both joints. The axial force applied was 525kN, which is 4.3 times the maximum recommended axial load (= 121kN). At an axial load = 121kN, the maximum axial strain of about 230 micro-strain and a maximum hoop strain of about 350 micro-strain were noted. These strains in the ANSI joint were lower than in the VCF joints at the same parameters.

Internal pressure plus bending (vertical load)

The strains in the bolts were a somewhat higher in the ANSI joint than in the VCF joint. Hoop strains at the hub of a VCF flange were double those of an ANSI flange, whereas in the VCF flange axial strain at the hub was about 5 times as high as in an ANSI flange. Overall maximum stresses calculated from the maximum strain (either hoop or axial) were lower than 2/3rdrd of the yield limit of the flange material. Maximum stresses in the bolts of the ANSI joint also were lower than 2/3rd of the yield limit of the bolt material. The maximum bending (vertical load) applied was higher than the recommended value of 65kN. The vertical load actually applied was 134kN with the ANSI joint and 156kN with the VCF joints.

Internal pressure plus bending (vertical load) plus tensile load

This is the most critical condition as all the loads were applied simultaneously on the test rig. For the ANSI joint, internal pressure and axial force were applied up to the limit specified, whereas bending (vertical load) was applied to above the level recommended. For the VCF joints internal pressure was limited, whereas the tensile load and bending applied up to 1.4 times higher than the specified limit. Overall maximum stress calculated from the maximum strain (either hoop or axial) was below 2/3rd of the yield limit of the pipe material. The maximum stress for bolt also was less than the allowable stress of the bolt material. Maximum stress for flange material was found to be above 2/3rd of the yield limit, but was slightly below the yield limit of the flange material. Some of the strain gauges in the axial direction were showing some type of residual strain. It is important to note that axial strains noted are quite high as compared to the hoop strains in all the cases. For the ANSI joint, high strains were noted in the bolts as compared with for the VCF joints. At the hub of a VCF-flange, hoop strains were slightly higher than with an ANSI flange, whereas for a VCF-flange, axial strains were comparatively high as compared with an ANSI flange.

Regarding the load capacity of a pipe joint the strains and stresses were found well within the allowable ranges. Stresses at high pressure loading were also found satisfactory. For *'internal pressure plus tensile load'* or *'internal pressure plus bending load'*, high axial strains in the VCF joints were noted although -- with the ANSI joint -- strains in the bolts were well below the specified allowable stress of bolt material. Strains in the ANSI bolts were quite high for every type of loading, but were within the allowable range.

Comparison of the Non-Gasketed and Conventional Gasketed Joint

Small dimensions, low weight (1/10th of the weight of an equivalent conventional one), easy handling, reduced maintenance and downtime for fixing leaks, less material and cost of manufacture, increased process temperature and pressure etc. are the main characteristics of the novel, non-gasketed pipe joints as compared to conventional designs. With the introduction of the novel, VCF non-gasketed joints, the monitoring of leakage may be reduced as compared with ANSI joints. This is the

result of metal to metal contact between flanges, what gives a good no-leak joint. As there is no gasket in the VCF joint, the removal and re-assembly may be done quickly. This means reduced cost of surveillance and maintenance. Neither flange faces nor bolts are exposed to internal media or the environment. A static mode of full metal-to-metal contact between flange faces does eliminate interface corrosion. For the same pressure rating class of 900#, torque required to pre-load the bolts for the VCF joint using small but recommended spanners was 65 Nm, which is very low as compared to the 505 Nm required for the conventional ANSI joint using a heavy torque wrench.

The weight of the modern joint is about $1/10^{th}$ of the weight of the equivalent, conventional one. With the VCF-joint no gasket is needed, but (if required) a seal or an O-ring may be added, whereas a gasket is a part of the conventional joint. No extra lubrication of threads is required with VCF joints, while lubrication of bolt threads is recommended with conventional joints. As no visible bending is observed in the VCF bolts (due to no flange rotation), no significant strains were expected or recorded. High strains in an ANSI flange are due to flange rotation from bolt pre-load plus other various loads. This can often cause leakage and failure from fatigue. Also it may result in gasket crushing or wear. Axial and hoop strains in the ANSI flange are low thanks to excessive flange dimensions and strength.

Comparative Reliability Assessment

A comparison of the weaknesses inherent in the conventional and the modern designs, can be made, although suitable quantitative failure rate data may not be available, by considering their basic design features with respect to:

- structural integrity and failure modes
- plant maintainability (make and break) criteria
- ability to cope with radial and angular misalignments on assembly, fluid hold up, complexity of assembly and skill level required, susceptibility to damage, speed of assembly and disassembly and interchangeability of component parts etc.

The conventional (ANSI) joint is based on an international standard and therefore confidence generally is placed on its structural integrity. The lack of extensive field tests should not preclude the use of the results from an emerging technology such as the VCF joint.

Considering several different factors, it can be concluded, that regarding the structural integrity that the modern, non-gasketed joint does offer, it is a better option. This is based on the requirement that the initial bolt/joint tightening operation is done properly, as the reliability of the joint is not only a function of their manufactured quality; it also depends upon the quality of operations. Considering these points, it is concluded that both the conventional and the modern joints are of high integrity and may be expected to serve well under appropriate installation and maintenance conditions. This has also been verified from a number of experiments performed

[5],[6]. This is the reason why it is now required, that VCF joints should be defined and treated as a complete unit, delivered and being shipped with all the bolts and nuts in place. It follows that any welding to a pipe must be made with the two flanges bolted together. It has been noted that in applications, where these novel flange joints have been used, no problems have been experienced [9].

Conclusions

A number of analytical and experimental tests have been undertaken to examine the behaviour of standard, gasketed and non-gasketed pipe flange joints. Grasping the importance of the influence of bolt preload is crucial in understanding the overall behaviour of the joint. From the results, a static condition dominates in the non-gasketed joint during preloading and operation. In the standard, gasketed joint, this is not the case and a joint with high variability in performance and hence operational confidence results. When a standard gasketed joint is 'ideally' set up, with guaranteed preloading with for example hydraulic bolt tensioners, then its performance is dramatically improved. It is considered that recommended preload values should be specified for each joint as this has been proved from the non-gasketed joint results that the higher the preload, the better the joint.

It is concluded, that the stamina of the non-gasketed, flanged pipe connection is outstanding, providing that it is properly built and installed.

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Plate 1 Standard, gasketed joint under combined loading
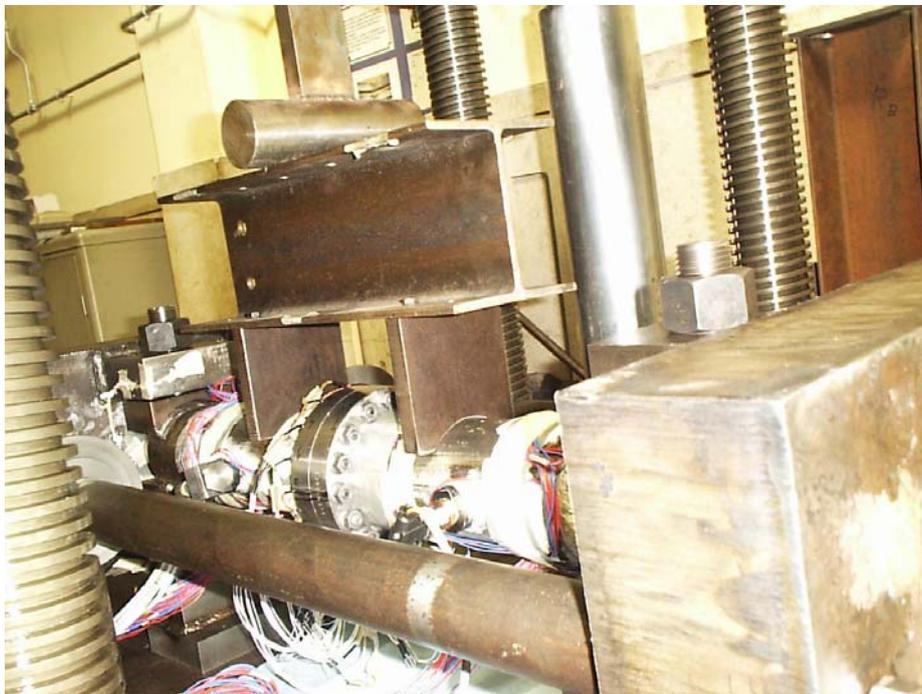
Plate 2 Non-gasketed joint under combined loading