

Behaviour of bolted flange joints in tubular structures under monotonic, repeated and fatigue loadings

I: Experimental tests

Hoang Van-Long, Jaspert Jean-Pierre and Demonceau Jean-François
ArGEnCo Department, University of Liège, Belgium

Abstract

This paper presents a test program on bolted flange plate connections used in circular tubular structures. In the tested joints, the connected tubular elements are made of high strength steel (TS590) while normal steel grade (S355) is used for the flanges. The tests were performed under monotonic loadings to obtain the mechanical properties of the connections and under cyclic loadings to characterize the behaviour of these connections for low cycle and high cycle fatigue. The test results are first presented herein, including a critical analysis of the latter; then, design methods available in literature are applied for the tested configurations with the objective to investigate the accuracy of these methods through comparisons with the experimental results.

Keywords: flange bolted joints; tubular elements; high-cycle fatigue; low-cycle fatigue; hot-spot stress; experimental tests.

1. Introduction

In practice, circular tubes are used in various types of structures, such as buildings, bridges, wind mill towers, radio/television masts, offshore structures, etc. The external loads acting on these structures are obviously from different types (for instance, exploitation loads, wind loads, wave loads, etc) and from different natures (monotonic, cyclic or repeated); according to the considered loads, the latter may induce low cycle or high cycle fatigue phenomena in structural details.

Also, bolted joints using circular flanges are frequently met within the mentioned structures, in particular for column splices. Such a type of connections has been the subject of many researches during the past three decades, e.g. [1, 2, 3, 4, 8, 10, 13 and 16]. Through these researches, design models have been proposed and/or improved using experimental, analytical and/or numerical

approaches, leading to different guidelines for the design of such connections. However, significant differences between the results obtained through these methods are observed. Moreover, most of the performed studies have been dedicated to the behaviour of such connection under monotonic loads while the behaviour of the latter under cyclic or repeated loads have been rarely considered, in particular for connections with High Strength Steel (HSS) tubes.

Besides that, Eurocode 3 part 1.8 [5] (dealing with the design of joints) doesn't give yet design guidelines for bolted joints using circular flanges leading to difficulties for the practitioners which have to deal with such connections.

It is the reason why, in 2008, a European project entitled "HITUBES - Design and integrity assessment of high strength tubular structures for extreme loading conditions" and funded by RFCS (Research Fund for Coal and Steel) was launched for three years; one of the objectives of this project was to investigate the behaviour of the previously mentioned connection configuration under monotonic and cyclic loadings. In particular, a test program has been performed in Liège on bolted flange connections between HSS tubes. The tested specimens were made of circular tubes with HSS and flanges with "normal" mill steels (S355). During the tests, the connections have been subjected to axial tension forces applied monotonically, to obtain the mechanical properties of the joints, and cyclically, to characterise the behaviour of the tested connections under low-cycle and high-cycle fatigue.

In the present paper, the performed test campaign is first described with a critical analysis of the obtained results (Section 2). Then, in a second step, a summary of current methods for the design of the tested connection configuration is reported in Section 3, with comparisons with the previously presented experimental results.

In a companion paper entitled "Behaviour of bolted flange plate connections in tubular structures under monotonic, cyclic and fatigue loadings. II: Numerical investigations" and published in the same journal, numerical studies performed on the investigated connections are also presented. The two companion papers constitute a global overview of the investigations conducted in Liège, in the

framework of the previously mentioned European project, with the objective to derive design recommendations for bolted flange connections.

2. Performed test program

The present section details the test program performed at the M&S laboratory (laboratory for Material mechanic and Structures, University of Liège, Belgium). The tested connection configurations will be first described in detail; then, the obtained test results will be presented and discussed.

2.1. Preparation of the tests

Description of the tested specimens and of the testing strategy

Two bolted joint configurations were tested with the objective to highlight different collapse modes. The properties of the tubular elements (356x12mm circular tube), the diameter of the circular flanges (556mm) and the number of bolts (12 bolts) are the same; only the thickness of the flanges and the size of bolts are varied between the two configurations (Fig.1):

- in Configuration 1, the flanges thickness is 15mm and M27 8.8 bolts are used;
- in Configuration 2, the flange thickness is 20mm and M20 8.8 bolts are used.

Full penetration welds were realised between the flanges and the tubes for both configurations. As mentioned previously, the steel grade for the tubes is TS590 (i.e. nominal yield stress equal to 590N/mm²) while the steel grade for the flanges is S355 (i.e. nominal yield stress equal to 355N/mm²).

Each configuration was tested under monotonic, high cycle fatigue and low cycle fatigue loadings. Therefore, in total, six specimens were fabricated and tested as presented in Table 1. Beside the main tests reported in Table 1, some supplementary tests aiming at obtaining information about the hot-spot stress distribution in critical zones were also performed in the elastic linear domain.

Tests on base materials

Tensile tests were carried out on coupons extracted from the tubes and the flanges composing the tested specimens. Also, tensile tests on bolts were performed; among the tested bolts, some were instrumented using the BTM-6C embedded strain gauges (Fig.2) in order to have more information on the stress-strain curve. The tests on the coupons extracted from the tubes and the flanges were performed following the recommendations given in ISO 6892 [11]. The mean values of the main characteristics of materials are presented in Table 2, more information can be found in [8].

Measurement of the initial geometrical deformations of the flanges

As illustrated in Fig.3, a significant deformation of the flanges has been observed; this deformation is due to the heat-effect of the welding process. The deformations of each flange at 12 points on the flange edges and uniformly distributed around the tube were measured using an electronic calliper and one straight ruler (Fig.3). The measured values (two flanges per specimen) are given in Table 3; the deformations for specimens M1 and M2 were not measured.

Instrumentations of the specimens

Four types of strain gauges were used to record the deformations on the tubes, on the flanges and in the bolts, as reported in Table 4. The locations of the strain gauges on the specimens are presented on Figs.4-7. The strain gauges were placed so as to measure deformations in critical zones, where the development of cracks/failure could have been expected, as in the bolt shanks, on the tubes at the weld toe or on the flanges at the weld toe. The strain gauges are only located on one side of the connections as the latter are symmetric. Among the 12 bolts present in each tested joint, there are 4 bolts are instrumented as illustrated in Figs.4-7. These instrumented bolts have also been used to control the applied torque moment during the bolt tightening procedure.

In order to obtain the global deformation of the joints, four displacement transducers are set-upped as shown on Fig.8.

Tightening procedure

A dynamometric wrench was used to tighten the bolts. The applied torque force was controlled using the measurements got from the strain gauges embedded in the instrumented bolts. According to Eurocode-3, part 1-8 (Article 2, Section 3.6.1) [5], the design preloading has to be taken as equal to $F_{p,Cd} = 0.7 f_{ub} A_s / \gamma_{M7}$, in which f_{ub} , A_s and γ_{M7} are the ultimate strength of the bolt (nominal value), the threaded section of the bolt shank and the safety coefficient (=1.1) respectively. With this force, the stresses in the threaded and unthreaded portions of the bolts are 509N/mm² and 407N/mm² respectively. Assuming a Young modulus equal to $E=2.1 \times 10^5$ N/mm², a strain equal to 1.9×10^{-3} can be computed for the unthreaded part, i.e. where the strain gauges are placed in the bolts. A “round-by-round” procedure is used for the tightening, until the value in the strain gauges in the instrumented bolts reach the previously computed value of strain (1.9×10^{-3}). During the tightening procedures, the indications from all the strain gauges were recorded.

Test sep-up

A SHENCK machine with a capacity of 2500 kN in both tensile and compression is used for the tests. The tests set-up is illustrated in Fig.9. Two tensile plates welded to the tubes are used to connect the specimens to the machine and to introduce the applied load. The distance between the tensile plates and the flanges of the tested connections is equal to 1000 mm in order to avoid affecting the stress distribution in the tubes near the tested connections. Accordingly, with this test set-up, the load acting on the specimens may be considered as a perfect axial load, in supposing that the tolerances are small.

Loading procedure

A quasi-static loading is adopted for the monotonic tests (Specimens M1 and M2). For the low cycle fatigue tests (Specimens L1 and L2), displacement control is used and a velocity of about 0.4mm/s was applied; so the applied loading may be considered as a quasi-static loading too. A frequency of 1.4 Hz was applied for the high cycle fatigue tests (Specimens H1 and H2). A constant load variation of 900 kN with a maximal value of 1000 kN is maintained for specimen H1

(configuration 1), while two constant load bands are used for Specimen H2 (configuration 2): (1) first band with a load variation of 600 kN (maximal load of 700 kN), and second band with a load variation of 1100 kN (maximal load of 1200 kN), as the show in Fig.10.

2.2. Test results and discussion

2.2.1. Test results

The measured load-displacement curves are presented in Figs.11-14 for the monotonic and low cycle fatigue tests (the displacement reported in these figures is the average value of the measurement, $((D1-D4)+(D2-D3))/2$, – see Fig. 8). The test on Specimen M1 had to be stopped because the load reached the maximal capacity of the testing machine (i.e. 2500 kN), but after reaching a significant deformation of the tested specimen (see Fig. 11). 37 cycles and 3 cycles of loading were applied to Specimens L1 and L2 respectively (see Fig. 13 and Fig. 14). For Specimen L2 (cyclic loading on configuration 2), the application of cycles with a constant displacement was initially planed. However, after the first cycle, no residual deformation was observed (see Fig. 14); accordingly, it was decided to increase the applied displacement for the second cycle but the same phenomenon was observed. At the third cycle, the displacement reaches the maximal point reached during the monotonic test, and the test was stopped. For Specimen L1 (configuration 1 under repeated loading), a typical behaviour for bolted joints (that have been presented in some researches, e.g [13, 15]) was observed with the development of hysteresis (see Fig. 13).

For the high cycle fatigue tests, 960000 cycles of loading were applied to Specimen H1 (configuration 1) while 2404000 cycles of the first loading band and 277600 cycles of the second loading band (Fig.10) were applied to Specimen H2 (configuration 2).

The failure modes of all tested specimens are presented in Figs.15-17; the corresponding observations are reported in Table 5. According to the T-stub concept as defined in the Eurocodes [5], a failure mode 2 was observed for configuration 1 (yielding of the flange plate + failure of the bolts) while a failure mode 3 occurred for configuration 2 (failure of the bolts with no yielding in the flange plate). This observation is not in agreement with the predicted failure modes obtained

through the analytical procedure used for the specimen design. The analytical methods will be summarized in Section 3.2.

By the limitation of space, the strain gauge records are not reported herein. However, the tendency of the strain gauge records at critical zones is illustrated in Fig.18. Noting that with respect to fatigue investigation, the stress variation (stress range) is more important than the absolute value of stress.

It is also interesting to highlight the following observation: the plates and the bolts of the joints were designed so as to have failure in the joints for the monotonic tests. Indeed, the ultimate load capacity of the joints was equal to less than 30% of the axial resistance (under monotonic loading) of the tube (= 10000kN). However, the cracks develop in the tubes during the fatigue tests meaning that the tubes were not over-designed as it looks like.

2.2.2. Discussion

Some specific aspects about the behaviour of flange bolted joints observed have been observed during the tests:

- The initial stiffness of the joints is very high, and the linear branch observed when unloading the specimens in the non-linear domain is not parallel to the initial branch (Figs.11 and 12).
- The stress in the bolts and the stress in the flange at the weld toe are almost constant in the linear domain while the stress in the tube at the weld toe considerably increases (Fig.17). The concentration of stresses in the tube near the weld toe is less important during the loading state in comparison with the tightening state. The way the stresses develop needs to be accurately understood and predicted as it influences the fatigue strength of the joints.

The above phenomenon may be explained as explained here below.

Due to the initial deformation of the flanges as previously described, the pre-stressing of the bolts during the tightening state induces pre-stresses not only in the bolts but also in other components:

- the bolts are principally pre-loaded in the axial direction;

- the flanges are principally pre-loaded in bending;
- the tube walls are principally pre-loaded in bending, in the longitudinal direction.

It is obviously that the stress components already pre-stressed in the tightening state will slowly increase in the beginning of the loading state, i.e. when the tensile load is applied to the specimen. Therefore, the stresses in bolts and the stresses in the flanges at the weld toes are almost constant in the linear domain. Also, the bending moment in the tube wall is almost constant but the axial force is progressively increases due to the application of the external load; this leads to the stress ranges in the tubes which are significantly higher than the ones in the bolts and on the flanges.

In order to clarify the above explanations, the behaviour of the joints can be illustrated as shown in Fig.19, in which two cases (with and without initial deformation of flanges) are distinguished. The forces applied to the joints are (see Figs.19 and 20): P , the external load, F_1 , the prying force, F_2 , the sum of the force in the bolt and of the contact force between the two flanges around the bolts, and F_3 , the sum of the contact forces between the two flanges inside the tubes. F_3 is equal to 0 in the case of perfect flanges, i.e. with no initial deformation, while it appears in joints with deformed flanges. This force (F_3) increases from zero to a maximal value at the end of the tightening state. In the loading state, F_3 progressively decreases, and may vanish at some moments. As the increase of the external load is almost equilibrated by the decrease of F_3 , F_2 almost doesn't change at the beginning of the loading state, leading to the non-variation of the moment in the flanges at the weld toe. Concerning the effects of the welding deformation of flanges on the joint behaviour, the interesting discussions can be also found in [1].

In summery, the influence of the initial deformation on the stress development in the different joint components is illustrated in Fig.21. If the initial deformation of flanges is omitted, as it is traditionally done for design, the stress range in the flange at the weld toe is very important in comparison with the stress range in the tube at the weld toe, and maybe the flanges would be identified as the critical component under the fatigue loading, which may not fit with the actual response of the joint including the initial deformation of the flanges.

3. Comparison of the test results vs. predictions from existing design methods

In this section, existing methods for the design of bolted joints is firstly summarized. The comparison between the predictions obtained through these methods and the experimental test results is then presented.

Under monotonic loading, the global behaviour of joints is described by the load-displacement curve characterised by its main parameters, i.e. the initial stiffness, the plastic resistance and the ultimate resistance. As the plastic resistance is strongly influenced by the “convention” used for its estimation when analysing test results, only the ultimate resistances are compared and discussed here after.

3.2. Existing methods to determine the resistance of joints

As the objective is to compare the method predictions to the experimental results, all the safety factors are taken equal to 1.0 and they are not appearing in the formulas reported here below. Moreover, the yield strength and the plastic moment are replaced by the ultimate strength and the ultimate moment respectively as the objective is to make comparisons on the predicted ultimate resistance of the joints.

3.2.1. Notices

Geometrical dimensions (Fig.22):

A : cross-sectional area of a tube;

A_s : threaded area of a bolt;

d : diameter of a bolt;

D : outside diameter of a tube;

D_p : diameter of bolt pitch circle;

D_f : diameter of a flange;

a_w : throat thickness of a weld;

e_1 : distance from the bolt axis to the outside surface of tube;

e_2 : distance from the bolt axis to the flange edge;

t : tube thickness

$$m = e_1 - 0,8\sqrt{2}a_w;$$

$$e = \sqrt{D_f^2/4 - m^2} - D_p/2;$$

n_b : number of bolts.

Geometrical coefficients:

$$k_1 = \ln \frac{D_p}{D-t};$$

$$k_2 = \frac{1}{2k_1} \left[2 + k_1 + \sqrt{k_1^2 + 4} \right];$$

$$k_3 = \ln \frac{D_p/2 + n}{D/2 - t/2}.$$

(n is the distance from the prying force to the bolt axis, see [2])

Mechanical characteristics:

B : total force in all the bolts;

B_0 : total preload in all the bolts;

B_u : ultimate resistance of all the bolts;

T : applied tensile force;

$f_{u,f}$: ultimate stress of a flange;

m_u : ultimate moment of a flange (per unit length);

ν : Poisson's ratio.

Fatigue parameters:

D_d : fatigue damage;

$\Delta\sigma_c$: detail category;

$\Delta\sigma_R$: stress range ($\Delta\sigma_{Ri}$ is the stress range in loading band i);

N_{Ri} : endurance (in cycles) of detail with the loading band i ;

n_1 : number of loading bands .

3.2.2. Igarashi method [10]:

By applying limit analysis for the flanges, Igarashi proposed analytical formulas to determinate the flange thickness and number of bolts to be used as follows:

$$t_f \geq \sqrt{\frac{2T}{f_{u,f} \pi k_2}};$$
$$B_u \geq T \left[1 - \frac{1}{k_2} + \frac{1}{k_2 \ln(D_f / D_p)} \right].$$

For a joint where the geometry and the material are defined, the capacity of the joint (T) can be obtain from above equations.

The Igarashi development has been adopted by CIDECT as a guideline for designing bolted circular flange joints [16].

3.2.3. Cao and Belle method [3, 4]:

Cao and Bell use a model in which the material is assumed to be linear elastic; the prying forces and bolt forces are assumed to be uniformly distributed at the flange edge and at the bolt pitch circle respectively. Analytical formulations for bolted joints were developed on the basis of these assumptions and are reported here below. Tables and charts were also established for practical purpose.

Firstly, the bolt force (B) is assumed to vary linearly with the applied tension force T following two straight lines:

$$B = 1.1B_0 \quad (\text{for } B \leq 1.1B_0)$$

$$B = \mu T + 0.1B_0 \quad (\text{for } B > 1.1B_0)$$

with μ , the slope of the curve depending of the geometrical dimensions of the joints (detailed formulas for the computation of μ can be found in [3]).

Secondly, the bending moment in the flange at the weld toe and at the bolt pitch circle may be determined as follows:

$$M_{ri} = \frac{-T}{8\pi} \left[(1-\nu) \frac{D_f^2 - (D-t)^2 - (D_f^2 - D_p^2)\mu}{D_f^2} + 2(1+\nu) \left(\ln \frac{D_f}{D-t} - \mu \ln \frac{D_f}{D_p} \right) \right]$$

$$M_{rp} = \frac{T}{8\pi} \left[(1-\nu) \frac{D_f^2 - D_p^2}{D_f^2} \left(\mu - \frac{(D-t)^2}{D_p^2} \right) + 2(1+\nu)(\mu-1) \ln \frac{D_f}{D_p} \right]$$

With a joint for which the geometry and the material are defined, the ultimate capacity of the joint is limited by the ultimate resistance of the bolts (B_u) and the ultimate bending moment in the flanges (M_u):

$$B \leq B_u;$$

$$M_r \leq M_u.$$

3.2.4. “Eurocode” concept [12]:

Referring to Eurocodes, no specific guidelines for the design of bolted flange joints are given. However, using the concept of T-stub as given in Eurocode-3, part 1.8 [5], CIDECT Research Project 5BP [12] proposes an approach for flange bolted joints fully in line with the Eurocode philosophy. Within the proposed approach, three failure modes are distinguished, and formulas for each collapse mode are proposed to determine the resistances of the joints (defined as the minimum of the resistances of the three possible collapse modes):

Mode 1 resistance – yielding of the flange (thin flanges):

$$T_1 = 2\pi m_p \left[1 + \frac{D + D_p}{2m} \right],$$

Mode 2 resistance – yielding of the flange + failure of the bolts (intermediate flanges):

$$T_2 = \frac{2\pi m_u + nB_u}{m+n};$$

Mode 3 resistance – failure of the bolts (thick flanges):

$$T_3 = B_u.$$

3.2.4. Couchaux method [2]:

Recently, Couchaux [2] proposed an alternative procedure for the design of flange bolted joints, combining the Igarashi model and the T-Stub concept as suggested in the Eurocodes. Five (5) failure modes are identified and characterised to compute the joint resistance. Only Modes 1, 2 and 3 are presented here after; Modes 4 and 5 are related to the connected tubes and the welds not decisive for the joint configurations considered herein.

Mode 1 resistance (thin flanges):

$$T_1 = 2\pi m_u \left[1 + \frac{2}{k_1} \right],$$

but $T_1 \leq 2\pi m_u n_b \min \left(2; 1 + \frac{2e}{\pi m} \right).$

Mode 2 resistance (intermediate flanges):

$$T_2 = 2\pi m_u \left[1 + \frac{1}{k_3} \right] + B_u \left[1 - \frac{k_1}{k_3} \right];$$

Mode 3 resistance (thick flanges):

$$T_3 = B_u.$$

3.3. Fatigue strength according to Eurocode-3 [6, 14]

As Eurocode-3, part 1.12 [7] don't give specific rules for the fatigue resistance of high strength steel, the rules given for normal carbon steel in Eurocode-3, Part 1.9 [6] are used.

According to the latter, the fatigue strength of joints may be calculated through the following steps:

- Identify the critical zones where cracks may develop;
- Assign to each critical zones a detail category as reported in Eurocode-3, part 1.9 (Tables 8.1 - 8.10 and Table B1 in [6]);
- Calculate for each detail category the reference stress ranges ($\Delta\sigma_c$), corresponding to a fatigue strength at two million cycles;
- Determine the stress ranges ($\Delta\sigma_R$) associated to the loading (the nominal stress range concept is used for the details of Tables 8.1-8.10 given in [6] while the hot-spot stress ranges is adopted for the details of Table B.1 given in [6]);
- Compute the endurance (in cycles) of each detail with a loading band for which the stress ranges are constant:

$$N_{Ri} = 2 \times 10^6 \left(\frac{\Delta\sigma_c}{\Delta\sigma_{Ri}} \right)^3 \quad (\text{for } N_{Ri} \leq 5 \times 10^6); \quad N_{Ri} = 2 \times 10^6 \left(\frac{\Delta\sigma_c}{\Delta\sigma_{Ri}} \right)^5 \quad (\text{for } N_{Ri} > 5 \times 10^6)$$

- Calculate, finally, the joint damage through the following formula:

$$D_d = \sum_i^{n_i} \frac{n_{Ei}}{N_{Ri}}$$

with n_{Ei} is the number of cycles associated to the stress range $\Delta\sigma_{Ri}$ for loading band i (in case of constant loading, i is equal to 1).

3.4. Comparison to the experimental results

With the geometric and material data of the considered joints (Section 2) and using the previously mentioned formulas, the ultimate resistances under monotonic loading and the fatigue strength of the investigated joints can be obtained. The so-obtained results are summarized in Tables 6 and 7.

To compute the fatigue resistance of the tube, the constructional detail number 11 in Table 8.5 of Reference [6] is used (tube socket joint with 80% full penetration butt welds). Even if a ring flange

(with a central hole) is considered for this detail, it is identified as the closest detail for the investigated connection. The stress range in the tube is calculated as the nominal stress by dividing the axial force by the tube area. When the hot-spot stress concept is adopted, the detail category of 100 is used (Table B1 of Reference [6]) while the stress range is deduced from the measured deformation given by the strain gauges (FL1, FL2 and QF1, QF2 on Figs. 4, 5 and 6).

Some remarks can be drawn:

- The results from the current methods and the experimental tests are quite different, in both ultimate capacity and failure mode (Table 6). It can be observed that the ultimate capacity of the joints is always underestimated through the analytical models, what is on the safe side. However, as the failure mode is not well predicted, it can lead to troubles in the design process; indeed, when the designer is looking for a ductile mode of failure (for instance, Modes 1 and 2 in the Eurocode concept), the fact that the actual failure mode is not well predicted can lead to a lack of ductility.
- There are significant differences between Eurocode and the test results concerning the fatigue strength of tube at the weld toe (Table 7). It is important to notice that the fatigue strengths given by Eurocode are not conservative in comparison with the test results. Even the geometries (ring/full flanges) and the material (normal/high-strength steel) are not corresponding between the used detail and that of the tested specimens, it cannot explain the observed differences. In fact, some inconsistency within the rules given in the Eurocode for this detail can be identified; indeed, the stress range in the tube at the weld toe is very sensitive to the geometries of the different components (tubes/flanges), what is not taken into account within the procedure of Eurocode 3, part 1.9 [6]. To improve the prediction given by the Eurocode, one solution would be to use the hot-spot stress (computed through FEM analyses or analytical approaches) in the concerned detail instead of the nominal stress as actually recommended in the Eurocode. The last opinion has been also recommended in some researches, e.g [2, 9].

4. Conclusions

Results of a test program on bolted joints with circular flanges were presented. The behaviour of these joints under monotonic, high cycle and low cycle loadings was recorded. The measured mechanical properties such as initial stiffness, ultimate capacity, failure mode, stress distribution and fatigue strength in the joint behaviour were given. The application of the current methods for the investigated joints was also presented and discussed.

It was observed that these existing guidelines give different values for the ultimate capacity of joints, but all methods are conservatives. However, the more important is the difference concerning the failure modes between the existing methods and the test results, which can lead to troubles in the design process as a ductile failure mode may be predicted through these methods while it is a brittle one which is actually occurring.

With respect to the fatigue resistance check for the tube at the weld toe by using Eurocode [6], some inconsistencies within the recommendations in Eurocode were identified; in particular, it was highlighted that the effects of the geometrical dimensions on the stress distribution are not well taken into account. To improve the prediction given by the Eurocode, one solution would be to use the hot-spot stress (computed through FEM analyses or analytical approaches) in the concerned detail instead of the nominal stress as actually recommended in the Eurocode; this solution is already suggested in other researches [2, 9].

Above remarks will be also clarified in the companion paper where numerical investigations conducted on the joints studied herein are developed.

Acknowledgements

This work was carried out with a financial grant from the Research Fund for Coal and Steel of the European Community, within HITUBES project: “Design and Integrity Assessment of High Strength Tubular Structures for Extreme Loading Conditions”, Grant N^o RFSR-CT-2008-00035.

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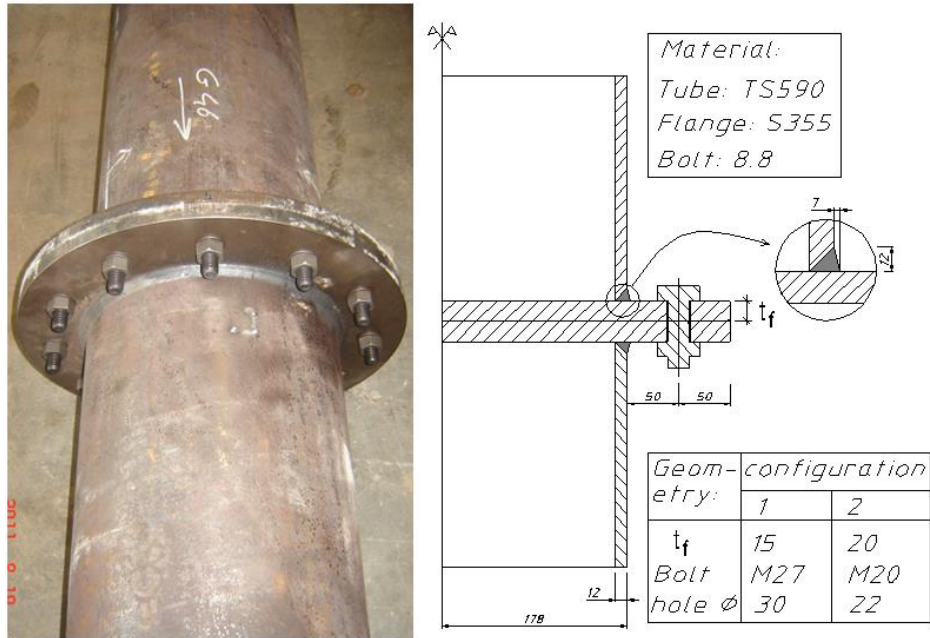


Fig.1. Specimen description

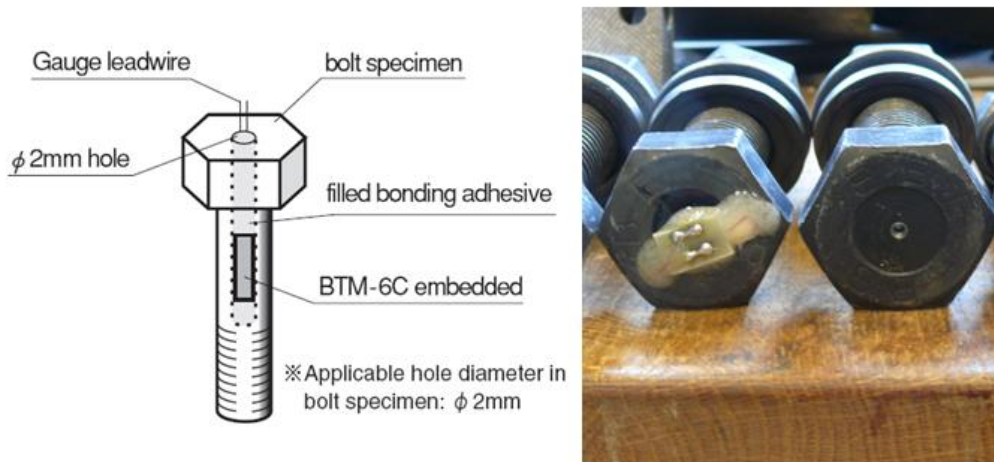


Fig.2. Instrumented bolts

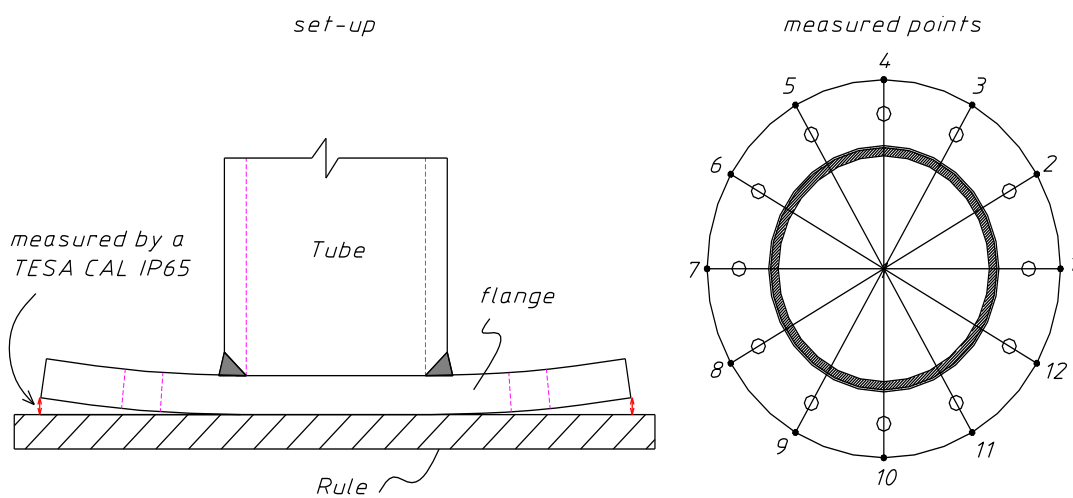


Fig.3. Measurement of the initial deformation of the flanges

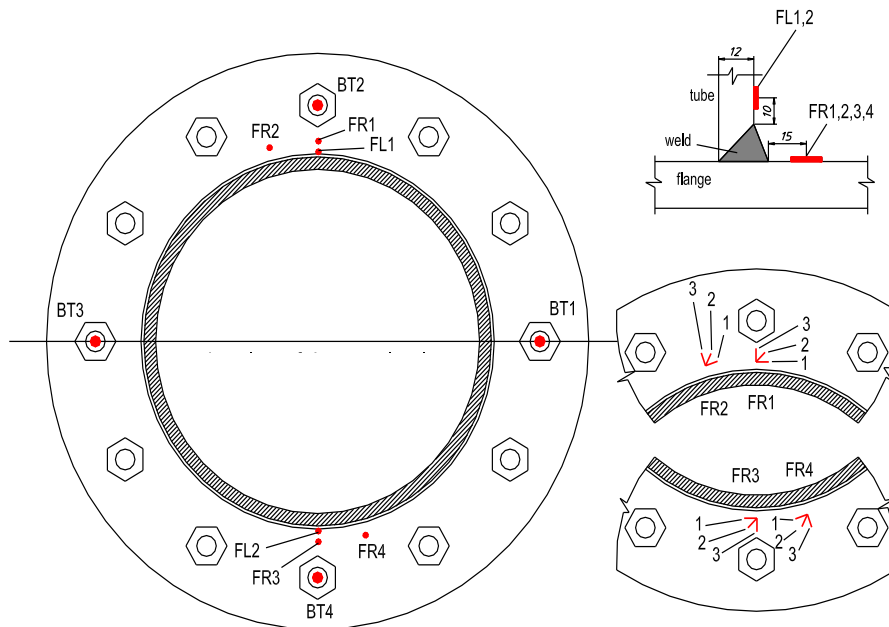


Fig.4. Location of strain gauges on specimens M1 and M2 (monotonic tests)

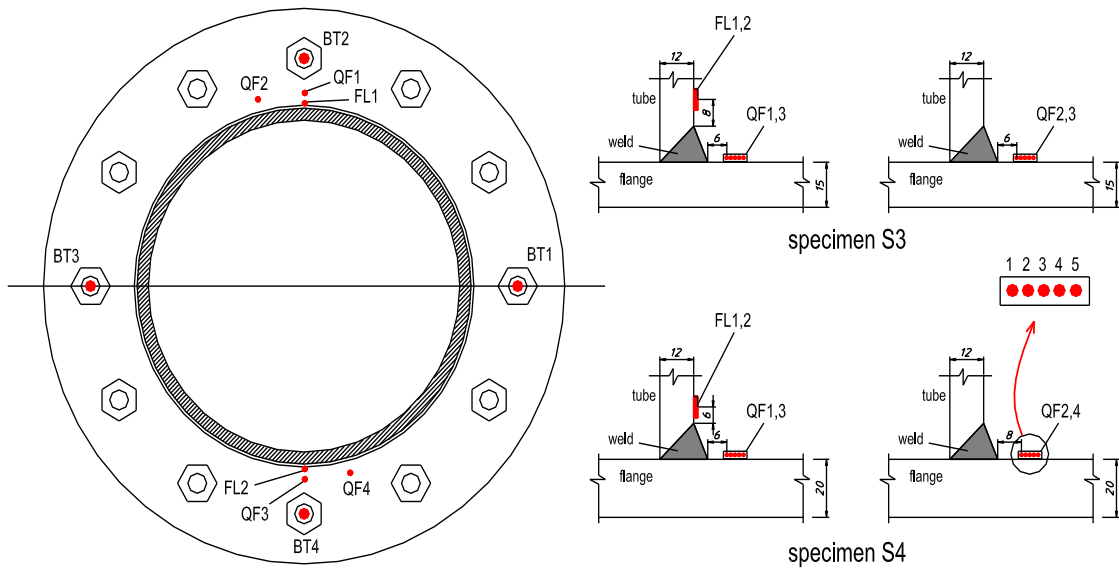


Fig.5. Location of strain gauges on H1 and H2 (fatigue tests)

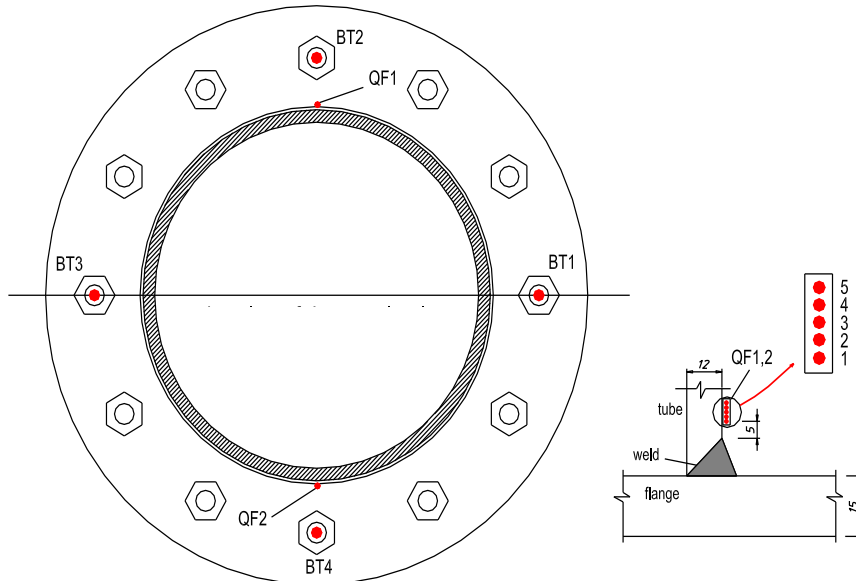


Fig.6. Location of strain gauges on specimen L1 (cyclic test)

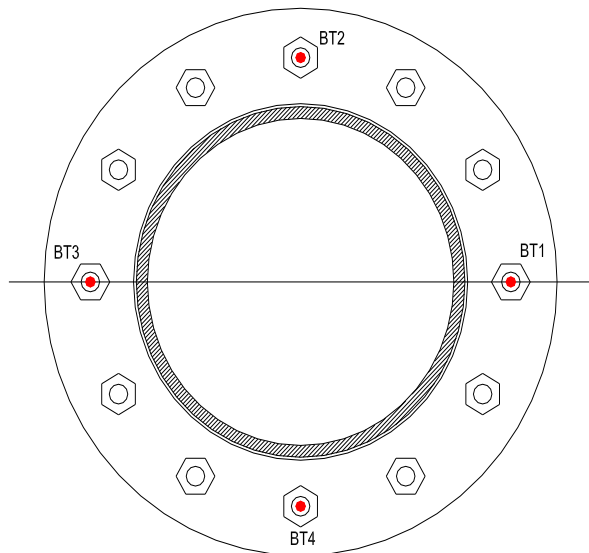


Fig.7. Location of strain gauges on specimen L2 (cyclic test)

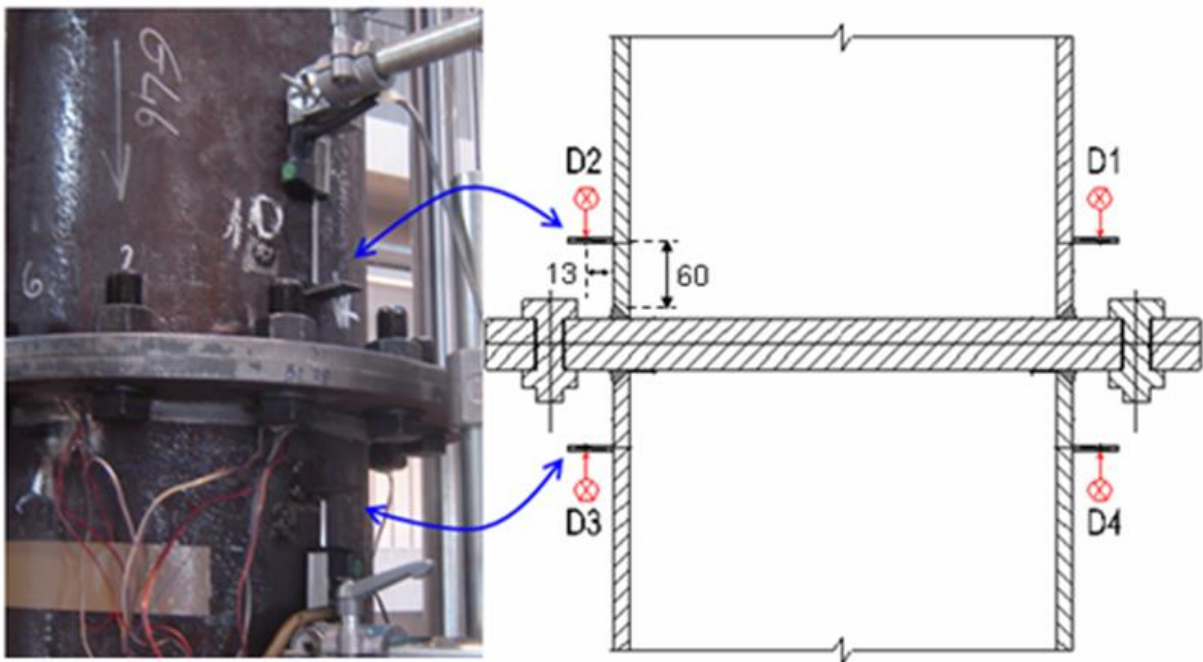


Fig.8. Location of displacement transducers on the specimens (four displacement transducers are used (D1, D2, D3 and D4))

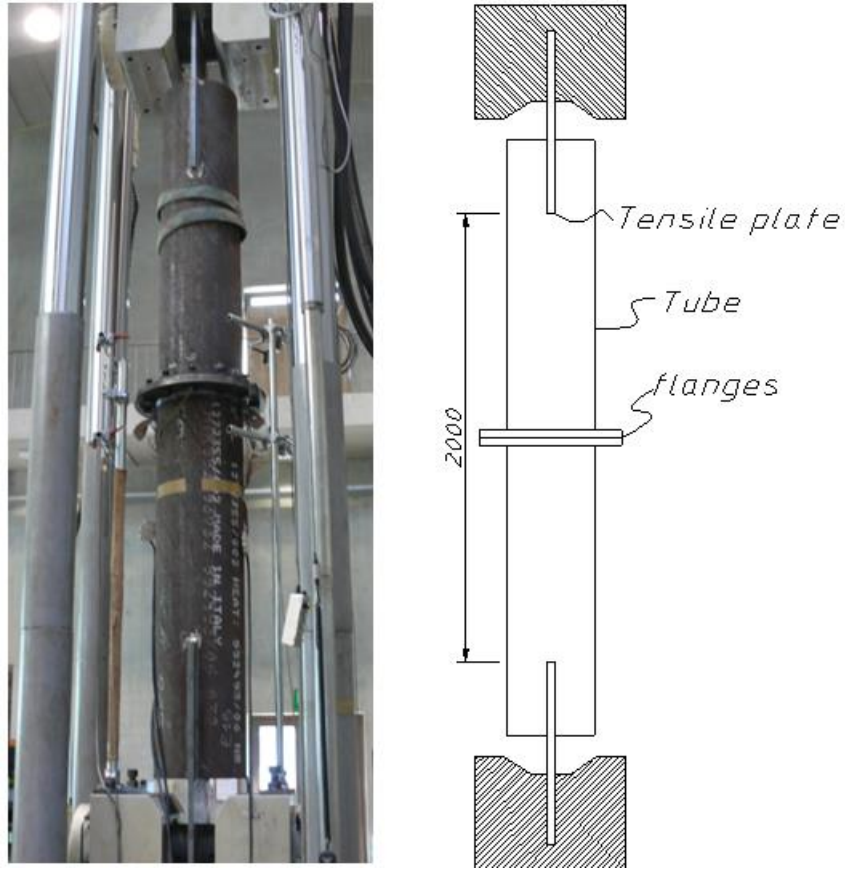


Fig.9. Test set-up

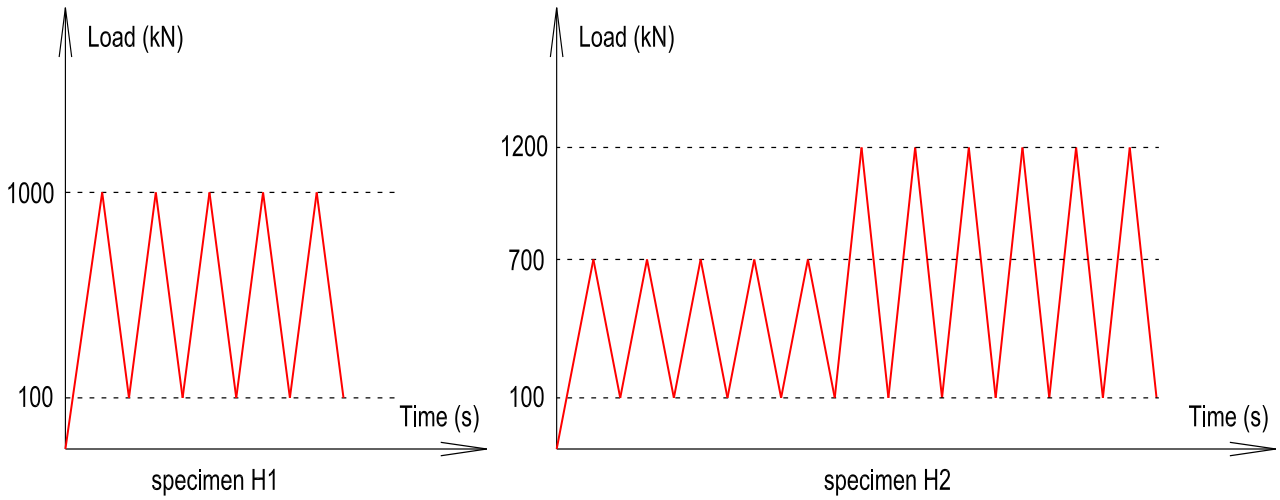


Fig.10. Loading procedures for high cycle fatigue tests

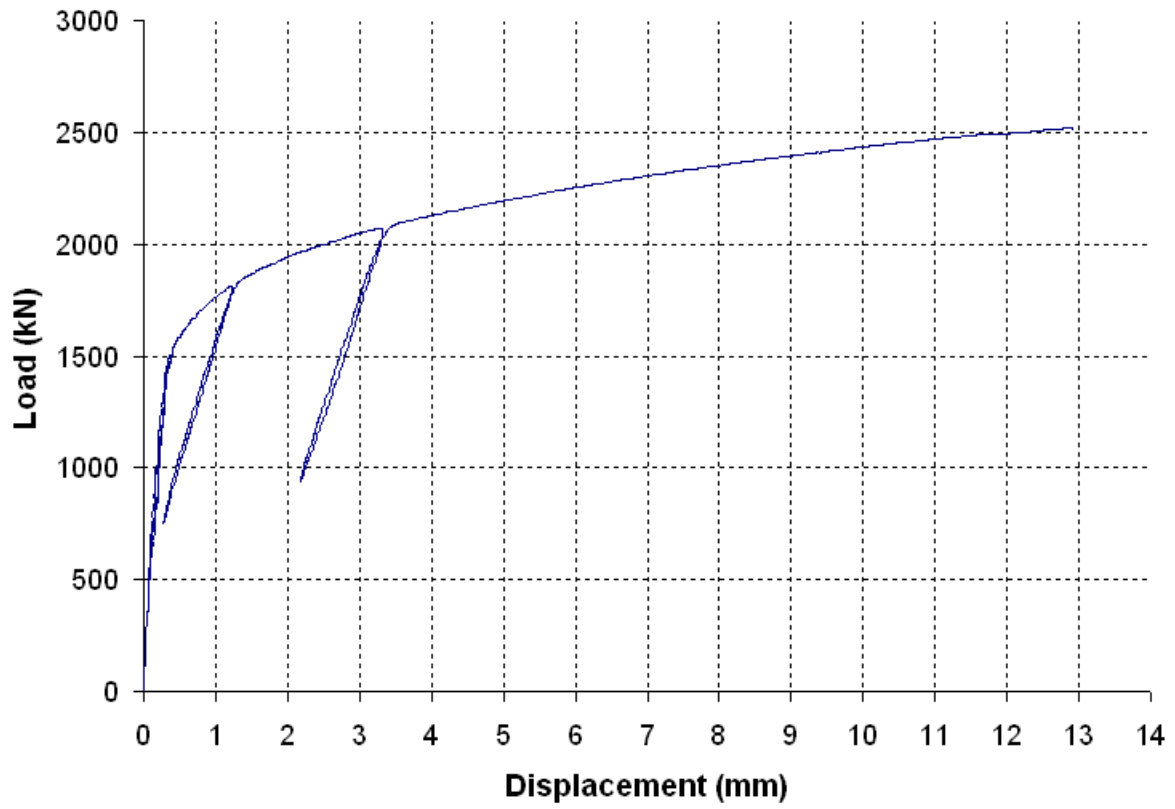


Fig.11. Load-displacement curve for Specimen M1 (configuration 1, monotonic loading)

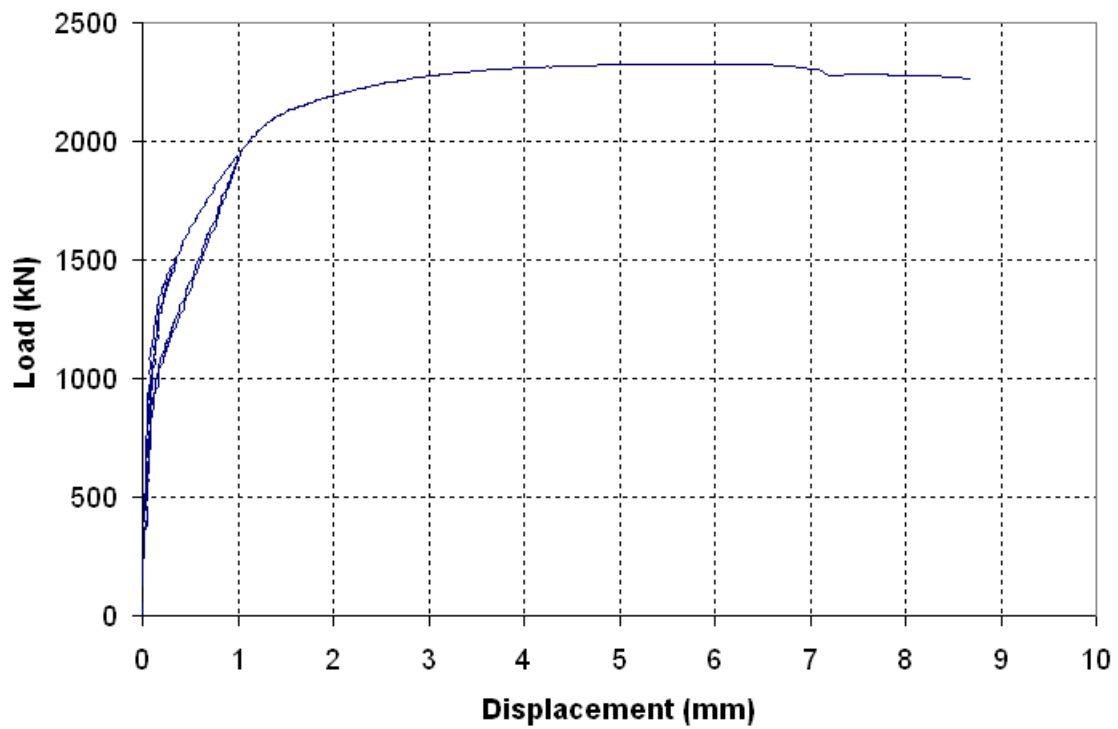


Fig.12. Load-displacement curve for Specimen M2 (configuration 2, monotonic loading)

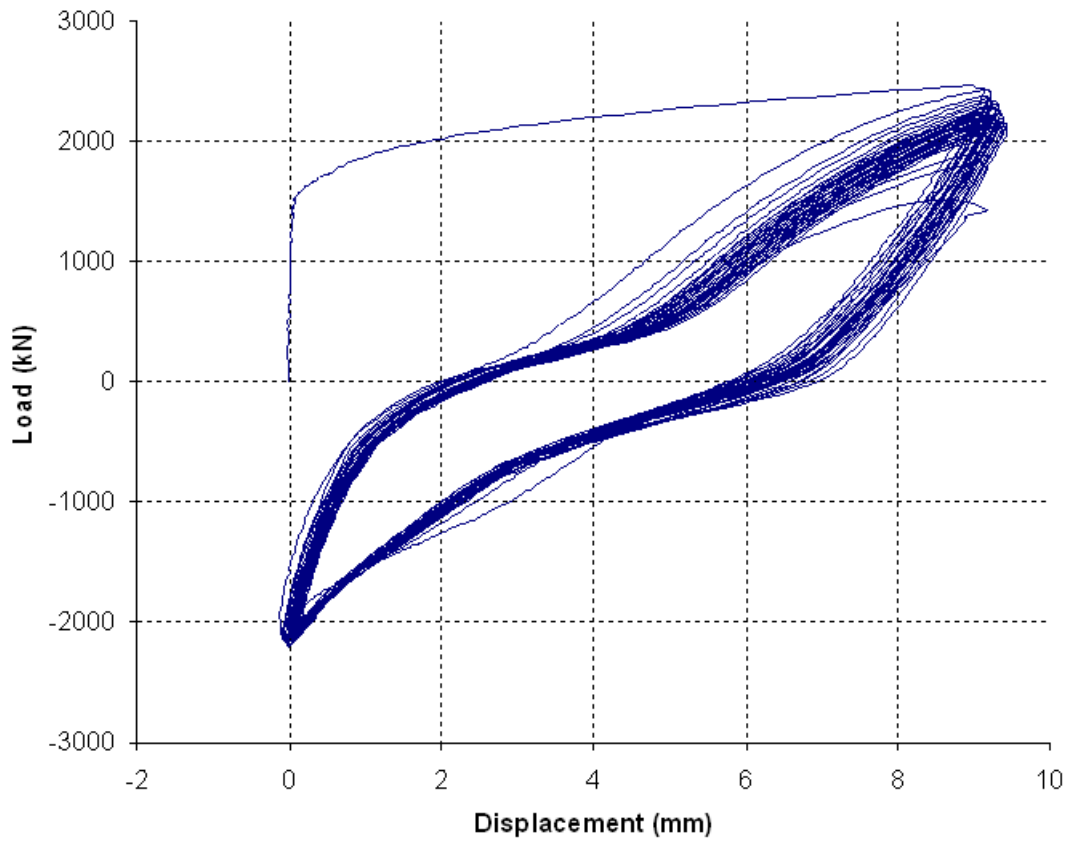


Fig.13. Load-displacement curve for Specimen L1 (configuration 1, repeated loading)

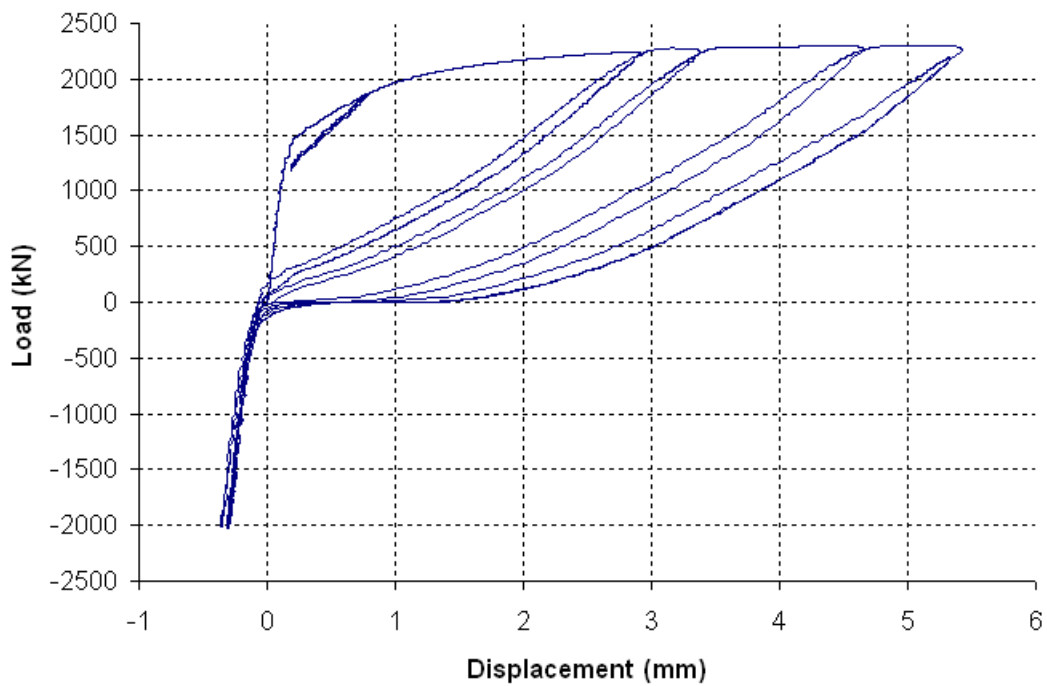


Fig.14. Load-displacement curve for Specimen L2 (configuration 2, repeated loading)



Specimen M1

Specimen M2

Fig.15. Specimens after monotonic testing



Specimen H1

Specimen H2

Fig.16. Specimens after fatigue testing



Crack in the flange of Specimen L1



A bolt of Specimen L2

Fig.17. Specimens after cyclic testing

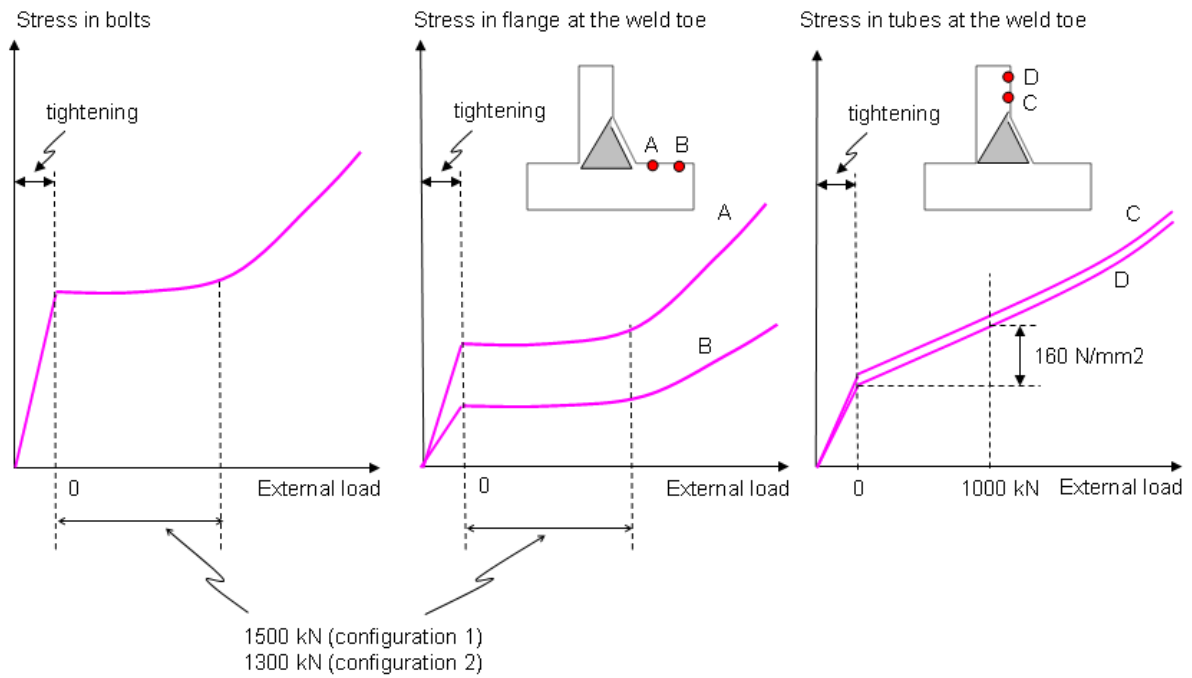


Fig.18. Illustrative view of the stress evolution in joints vs. the applied external load

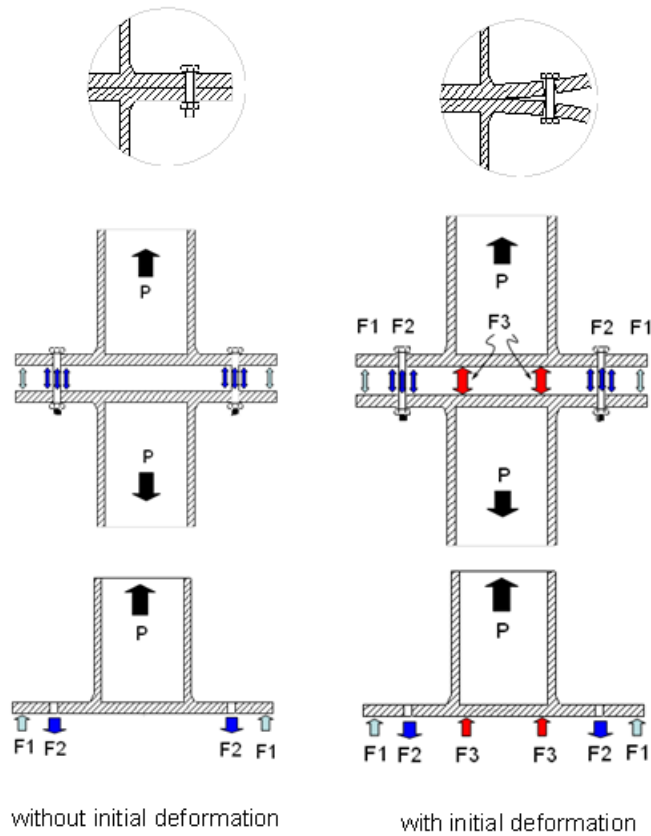


Fig.19. Forces acting on the joint

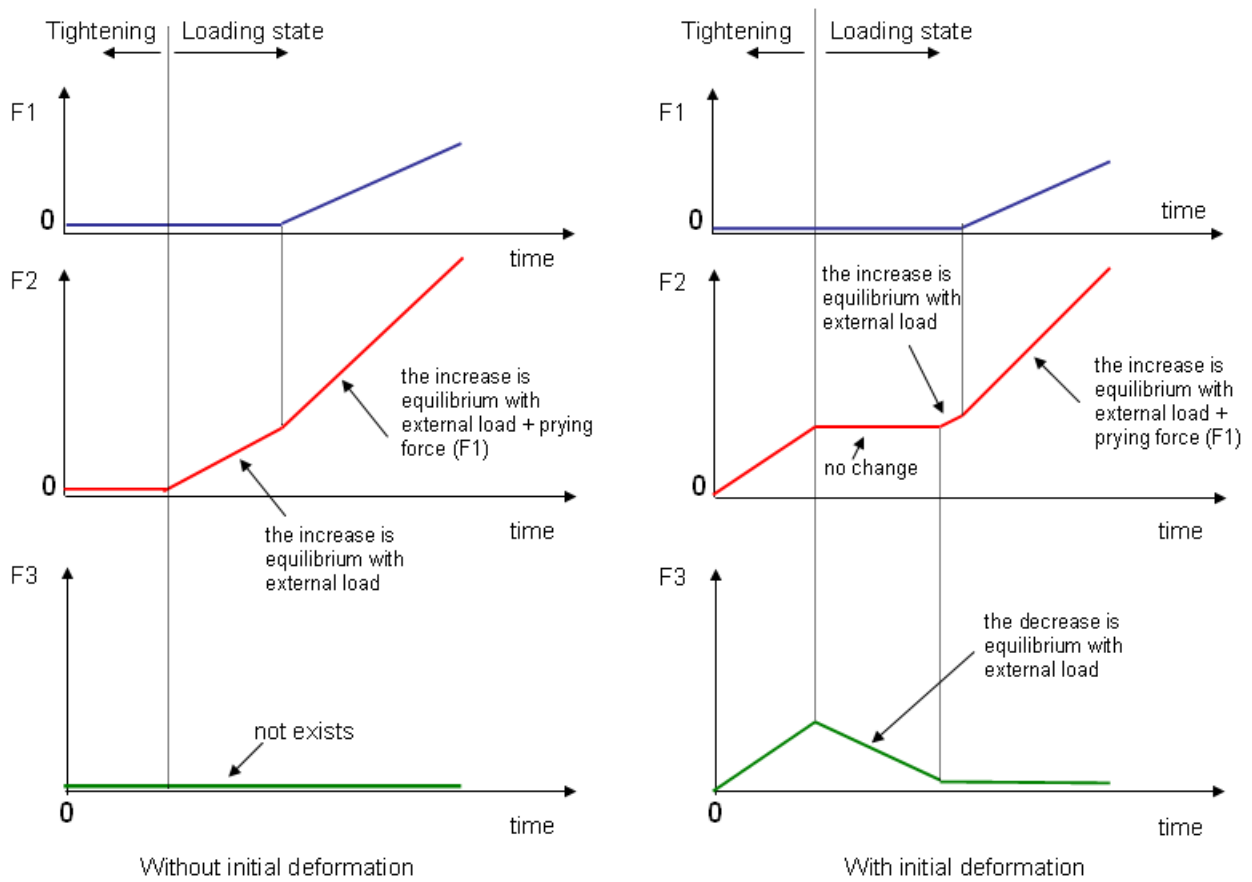


Fig.20. Development of the forces acting on joint

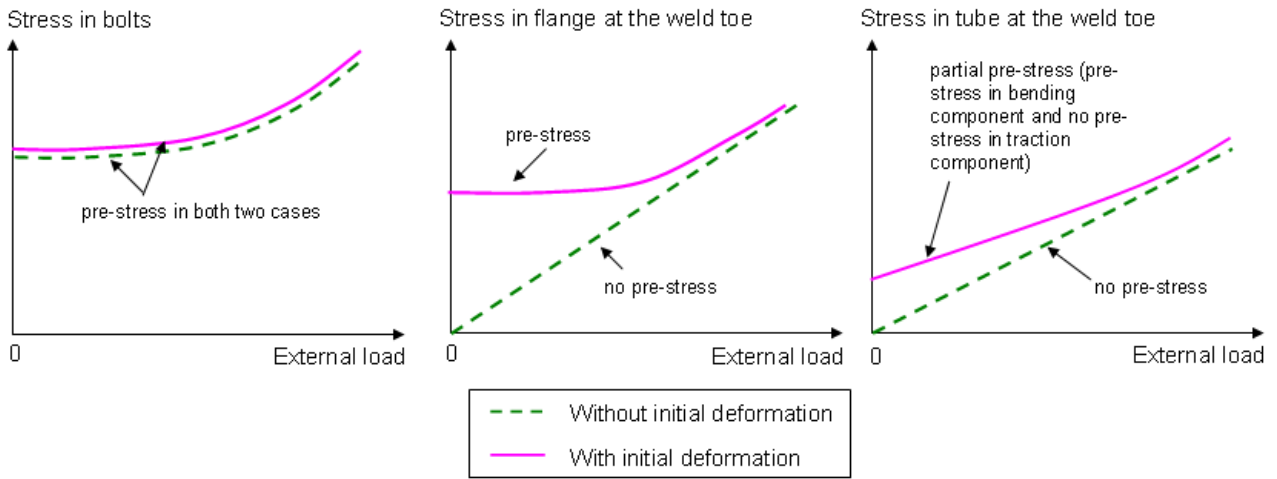


Fig.21. Comparison of the evolution of stresses in different components

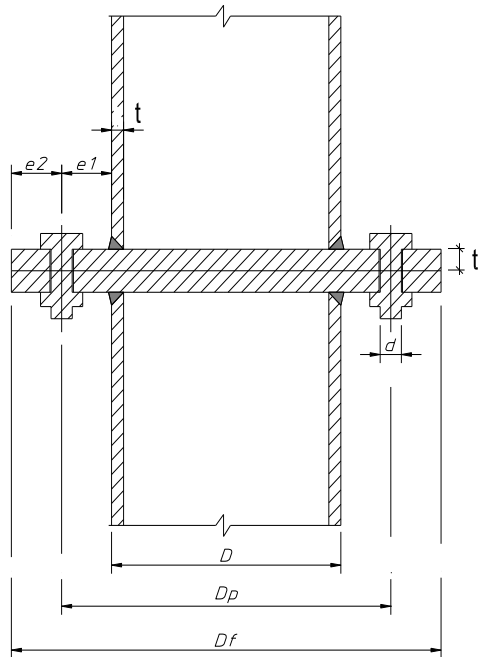


Fig.22. Geometrical parameters of the considered joint

Table 1: List of performed tests

Test number	Specimen name	Configuration (Fig.1)	Loading type
1	M1	1	Monotonic loading
2	M2	2	
3	H1	1	High cycle fatigue loading
4	H2	2	
5	L1	1	Low cycle fatigue loading
6	L2	2	

Table 2: Results of coupon tests

Parameters	Plate 15mm	Plate 20mm	Tube	M27 bolts	M20 bolts
Yield strength (N/mm ²)	387	384	822	857	850
Ultimate strength (N/mm ²)	544	529	881	930	930

Table 3: Initial deformations of the flanges

Spec.	Flange	Measured value in mm at the points given in Fig.3											
		1	2	3	4	5	6	7	8	9	10	11	12
H1	1	1,79	2,99	3,28	2,39	3,88	2,92	1,39	2,88	3,10	2,87	3,65	3,03
	2	2,25	2,67	2,65	2,35	2,84	2,78	2,57	2,72	2,92	2,12	3,26	2,79
H2	1	1,98	2,11	2,07	1,57	1,72	1,59	1,22	1,39	1,57	1,30	1,63	1,59
	2	1,86	2,46	2,30	1,52	2,35	2,29	2,20	1,96	1,82	1,82	1,56	1,71
L1	1	1,84	0,82	1,62	2,03	1,30	1,89	1,93	0,95	1,94	2,47	1,37	1,98
	2	1,68	2,80	2,5	1,63	2,56	2,49	1,59	2,98	2,82	1,62	2,68	2,68
L2	1	4,37	3,65	3,38	2,48	2,92	3,25	2,56	2,65	2,38	1,97	3,47	4,44
	2	1,74	2,55	2,95	1,67	2,45	2,60	2,30	2,83	2,57	1,66	2,05	2,21

Table 4: Used strain gauges

Type of strain gauge	Description	Symbol	Location
BTM-6C	axial strain gauge for bolts	BT	In the bolts (Figs.4-7)
FRA-6-11	0°/45°/90° 3-element rosette	FR	On the flanges (Fig.4)
FLA-6-11	Single element	FL	On the tubes (Figs.4 & 5)
QFXV-1-11	5-element single-axis	QF	On flanges and tubes (Figs.5 & 6)

Table 5: Description of tested specimens at the end of the tests

Specimens	Load type	Referent figures	Component states		
			Tubes	Flanges	Bolts
M1	Monotone	Fig.15	No particular sign	<i>Plastic deformation at weld toe</i>	<i>Plastic deformation occurs</i>
M2	Monotone	Fig.15	No particular sign	Very small deformation	<i>All bolts are failed</i>
H1	Fatigue	Fig.16	<i>Crack at the weld toe</i>	No particular sign	No particular sign
H2	Fatigue	Fig.16	<i>Crack at the weld toe</i>	No particular sign	<i>A bolt is completely cracked, others are partially cracked</i>
L1	Repeated	Fig.17	No particular sign	<i>Crack at the weld toe</i>	<i>Plastic deformation is observed</i>
L2	Repeated	Fig.17	No particular sign	Very small deformation	<i>Plastic deformation is observed</i>

Table 6: Comparison of the methods (monotonic loading)

Method	Configuration 1		Configuration 2	
	Ultimate capacity (kN)	Critical components	Ultimate capacity (kN)	Critical components
Igarashi [10]	1466	flanges	1801	bolts
Bell and Cao [3,4]	1493	flanges	1596	bolts
“Eurocode” [12]	1944	flanges	2249	flanges + bolts
Couchaux [2]	1561	flanges	2161	flanges + bolts
Experimental	>2500	flanges + bolts	2320	bolts

Table 7: Comparison of the method (fatigue resistance - high cycle loading)

Method	Damage index	
	Configuration 1	Configuration 2
Eurocode (nominal stress) [6]	0.45	0.38
Eurocode (hot-spot stress) [6]	1.32	1.67
Experimental	1.00	1.00