# Fatigue behavior of a bolted assembly – A comparison between numerical analysis and experimental analysis

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ABSTRACT

The fatigue behavior of a bolted assembly can be analysed, either by fatigue tests, or by computing the stress variations and using a fatigue curve.

This paper presents the fatigue analysis of a stud-bolt and stud-flange of a steam generator manway carried out with the two methods.

The experimental analysis is performed for various levels of load, according to the recommandations of the ASME code section III appendix II.

The numerical analysis of the stresses is based on the results of a finite element analysis performed with the program SYSTUS. The maximum stresses are obtained in the first bolt threads.

In using these stresses, the allowable number of cycles for each level of loading analysed, is obtained from fatigue curves, as defined in appendix I section III of the ASME code.

The analysis underlines that, for each level of load the purely numerical approach is highly conservative, compared to the experimental approach.

## 1 - INTRODUCTION

Pressure vessel manways are generally closed by a bolting system. According to section III of the ASME code [1], a fatigue analysis of this system is required, particularly at the thread fillet radius where stress concentration exists.

This analysis can be performed by two approaches :

- a numerical approach which consist in calculating stresses at the thread fillet radius and determining usage factor by using a fatigue curve,

- an experimental approach, in using representative specimens of the actual configurations.

This paper compare results obtained by the two approaches.

## 2 - PRESENTATION OF THE STUDY

As we want to compare experimental and numerical analysis, the model used to perform the latter analysis, is representative of the test configuration shown in figure 1.

This configuration is subjected to constant range axial cyclic loadings. From the number of cycles applied when failure occurs, we calculate in using appendix II ASME section III rules, the "experimental" allowable number of cycles for the specimen.

From the maximal stresses obtained by numerical analysis at the thread fillet radius, we also determine the numerical allowable number of cycles in using fatigue curves given by appendix I of the ASME code section III [1].

Then, the comparison of the minimum allowable number of cycles which results from the two approaches shall determine the conservatisme of the numerical approach when compared to the actual specimen behavior.

## 3 - EXPERIMENTAL ANALYSIS

### 3.1 Tests presentation

A cyclic tensile force is applied to the test rig shown in figure 1. The tests are performed for three cyclic load levels :

$F_{max} = 6{,}45.10^5$ N $\quad F_{min} = 0$

$F_{max} = 6{,}45.10^5$ N $\quad F_{min} = 2{,}62.10^5$ N

$F_{max} = 5{,}14.10^5$ N $\quad F_{min} = 0$

and two tests are carried out for each load level.

The range of the mean stress induced in the threaded parts of the stud by these loads are as follows :

- load n° 1 : $\Delta\sigma$ = 442 MPa
- load n° 2 : $\Delta\sigma$ = 263 MPa
- load n° 3 : $\Delta\sigma$ = 352 MPa

These ranges are representative of the cyclic loadings sustained by studs under plant operating conditions.

During the tests, different types of non-destructive examinations are used to detect crack initiation in studs :

- ultrasonic inspection,
- acoustical emission examination,
- liquid penetrant examination after detection by the other methods,
- eddy current examination.

### 3.2 Results

The table here after gives the number of cycles corresponding to the initiation detection.

| | | initiation cycle number | |
|---|---|---|---|
| load level | $\Delta\sigma$ | Test a | Test b |
| 1 | 442 | 36 000 | 57 300 |
| 2 | 263 | 227 700 | 474 900 |
| 3 | 352 | 1 212 800 | 193 100 |

During test 2a, one echo was detected by ultrasonic inspection after 227 700 cycles. As no defect was detected by liquid penetrant examination, the test was continued.

Then the stud failure was observed after 252 500 cycles, in a section located between the two first threads of the flange side.

In the others tests, the crack initiation in the stud was observed at the level of the first threads of the nut side.

### 3.3 Analysis of results according to appendix II of the ASME code section III

The method proposed by this appendix indicates how determine the allowable configurations :

- use the actual number of cycles and increase the load by a coefficient Ks,
- use the actual load and increase the number of cycles by a coefficient Kn,
- increase both load and cycles.

The configuration is allowable if the test does not lead to failure of the specimen.

Here we use the first two methods to verify from test results which configurations are acceptable with respect to appendix II ASME III code criteria.

For that purpose we consider for each load level, the minimum number of cycles corresponding to crack initiation resulting from the two tests. Then we deduce acceptable configurations either in dividing the load by Ks or in dividing the number of cycles by Kn.

#### 3.3.1 Approach by load reduction

Ks results from the product of 4 coefficients which depend of the effect of size, surface finish, temperature and variation in test results.

Here, only the last coefficient is greater than 1. Thus, the number of test for each load level being equal to 2, we obtain : Ks = 1,47 - 2 x 0,044 = 1,382.

Therefore, according to appendix II of the ASME III code, the acceptable operating conditions are :

| Configuration | Minimum load ($10^5$ N) | Maximum load ($10^5$ N) | Number of cycles |
|---|---|---|---|
| S1 | 0 | 4,67 | 36 000 |
| S2 | 1,89 | 4,67 | 227 700 |
| S3 | 0 | 3,72 | 193 100 |

#### 3.3.2 Approach by cycle reduction

The reduction of the number of cycles is made as follows :

$$Kn = (Ks)^{4,3} \# 4$$

The results obtained are then :

| Configuration | Minimum load ($10^5$ N) | Maximum load ($10^5$ N) | Number of cycles |
|---|---|---|---|
| N1 | 0 | 6,45 | 9 011 |
| N2 | 2,62 | 6,45 | 56 996 |
| N3 | 0 | 5,14 | 48 335 |

## 4 - NUMERICAL ANALYSIS

The method consists in computing the maximum stresses at the thread fillet radius of the studs tested. For each load level applied during testing, the usage factor is further determined according to ASME section III NB 3200 rules [1].

Stress calculation is performed using the "SYSTUS system" [2] based on the finite element method.

### 4.1 Finite element model

Two calculations are performed with two models which correspond to "stud-to-nut connection and stud-to-flange connection.

These models are shown in figures 2 and 3 respectively. Figure 4 show the mesh of the stud threads. Calculations are made using linear elastic option of SYSTUS system [2].

Loading and ties applied to the models are specified in figures 2 and 3. Stresses induced by the loads defined above are obtained by a linear correction.

### 4.2 Maximum stresses

The maximum stresses are obtained at the thread fillet radius. We observe a rapid decrease of stresses from the fillet radius surface. In this area, stresses are divided by a factor 2 at 0,2 mm depth.

In the following, we use the maximum stresses observed at the fillet radius surface.

Figures 5 and 7 show the variation of the maximum to mean stresses ratio (stress concentration factor) versus thread location in stud and nut.

Figures 6 and 8 show the same variation for the stud-to-flange connection.

### 4.3 Fatigue analysis

From these results, we determine the usage factor at the thread fillet radius of the stud, for each level of load applied when testing, in using fatigue curve of the ASME III code [1].

Results obtained are :

- with the maximum stresses observed in stud (in using fatigue curve given by fig. I.9.4 of the Appendix 1 of the ASME III code) :

| Configuration | $\Delta\sigma$(MPa) | $n_r$ | $n_c$ | $n_e$ | u | u (k=4) |
|---|---|---|---|---|---|---|
| S1 | 1 928 | | 537 | 36 000 | 67 | 30 |
| S2 | 1 146 | | 1 498 | 227 700 | 152 | 68 |
| S3 | 1 535 | | 843 | 193 100 | 229 | 102 |
| N1 | 2 665 | 36 000 | 281 | 9 011 | 32 | 14 |
| N2 | 1 585 | 227 700 | 792 | 56 996 | 72 | 32 |
| N3 | 2 122 | 193 100 | 447 | 48 335 | 108 | 48 |

- with the maximum stress observed in flange (in using fatigue curve given by fig. I.9.1 of the appendix I of the ASME III code) :

| Configuration | $\Delta\sigma$(MPa) | $n_r$ | $n_c$ | $n_e$ | u |
|---|---|---|---|---|---|
| S1 | 911 | | 2 000 | 36 000 | 18 |
| S2 | 542 | | 9 108 | 227 700 | 25 |
| S3 | 726 | | 3 713 | 193 100 | 52 |
| N1 | 1 259 | 36 000 | 819 | 9 011 | 11 |
| N2 | 749 | 227 700 | 3 352 | 56 996 | 17 |
| N3 | 1 003 | 193 100 | 1 510 | 48 335 | 32 |

where :
- $n_c$ is the number of cycles obtained from fatigue curve with a stress range of $\Delta\sigma$
- $n_e$ is the number of cycles obtained from appendix II methode,
- $n_r$ is the minimum real number of cycles before crack initiation occurs during tests
- u is the usage factor corresponding to $\Delta\sigma$ and to the number of cycles $n_e$,
- u ($k = 4$) is the maximal usage factor obtained in the stud by using a stress concentration of 4 according to ASME III code.

## 5 - COMPARISON OF THE TWO APPROACHS

The previous tables show a very large discrepancy between the number of cycles allowed by experimental approach (appendix II of the ASME III code) and the one obtained by usage factor calculation at the more stressed point of the test rig. In the following, we try to explain the large difference observed :

a) Requirements about the correction coefficient applied to the number of cycles are more severe in the numerical approach than in the experimental one. The number of cycles allowed by the appendix II of the ASME III code for only one representative test is the one obtained when failure occurs divided by :

$$(1,47)^{4,3} = 5,24$$

For two tests, this coefficient is reduced to 3,99.

Furthermore, the fatigue curves used to calculate the allowable number of cycles are deduced from mean curve corrected by a coefficient included in the range from 7 to 300 as shows by figure 9 which give ASME code fatigue curves and experimental results obtained for various bolting materials [3].

b) The fatigue curve which includes a mean stress correction corresponding to 2,7 Sm and 3 Sm. A correction taking into account the real mean stress shall give a more realistic usage factor.

c) The usage factor calculation is based on the maximal stress obtained at the surface thread fillet radius. Many works [4], [5] show that in an area where stresses are highly concentrated (notch effect), the crack initiation is piloted by the variation of the normal stress $\Delta\sigma_{\theta\theta}$ at a characteristic distance d from the notch surface. Calculations performed here, according to ASME rules are therefore highly conservative.

## 6 - INFLUENCE OF A PLASTIC DEFORMATION AT THREAD FILLET RADIUS

The stresses used to perform fatigue analysis result from elastic calculations. In high stress concentration area such as notches, elastic calculations lead to strain which are underestimated. Therefore, as crack initiation in such areas is piloted by real strain ranges, the risk that this damage appears is also underestimated. In order to take this fact into account, we can correct the elastic strain according to Neuber's method. We can also determine an elastic-plastic strain correction factor Ke which is defined as being the ratio of the real strain range over the theoretical elastic strain range.

Then the analysis is realised according to chapter NB 3228.5 of the ASME III code [1].

In this context, we performed an elastic plastic calculation by a finite element method in using a model which is representative of the configurations analysed here. This calculation leads to a Ke factor near the value of 2.

Therefore, if we calculate the usage factor at the thread fillet radius, taking into account this value of the Ke factor, we shall obtain much more margins with respect to the experimental approach that those carried out on previous tables (see § 4.3).

The elastic plastic calculation reveals also that the real load distribution on the thread side is more uniform that those obtained by an elastic calculation as we can see it in figure 10.

## 7 - CONCLUSION

The fatigue analysis of a bolted junction subjected to cyclic loadings, leads to result which are different whether we use the experimental approach proposed by appendix II of ASME III code [1] or the numerical approach described in chapter B 3200 of the same code. The results presented in this paper show effectively that the numerical approach yields allowalbe numbers of cycles much lower than those resulting from the experimental approach. When we take into account the NEUBER'S correction or the elastic-plastic correction, this difference is still more important.

For a greater coherence of results obtained from the two approaches, we suggest :

- to modify the fatigue curve recommended by appendix I of the ASME III code in order to reduce the overconservatism for large number of cycles,
- to modify the fatigue curves recommended by appendix I of the ASME III code, in cases where the maximal stresses are lower than 2,7 Sm, in order to reduce the effect of the mean stress correction,
- to admit fatigue analyses based on stress values obtained at a caracteristic distance from the thread fillet radius,
- to limit stress concentration factor at 4.

Furthermore, this paper shows that the difference between experimental an numerical approaches increase when we perform a correction in order to take into account the plastic strain at the thread fillet radius. Therefore, we suggest overlooking this correction until the parameters that monitor fillet radius fatigue are better defined.

## REFERENCES

[1] Asme Boiler and Pressure Vessel code. Section III 1986 edition.
[2] SYSTUS - Programme de calcul de structure par éléments finis. FRAMATOME.
[3] SNOW, LANGER - "Low cycle fatigue of large diameter bolts". Trans. of the ASME - Feb. 1967.
[4] d'ESCATHA and all - "A criterion for analysing fatigue crack initiation in geometrical singularities". C34/80 Int. Conf. on Pressure Vessel Technology. London 1980.
[5] NEUBER - Theoritical determination of fatigue strength stress concentration. Technical report AFML RT 68.20 - April 68.

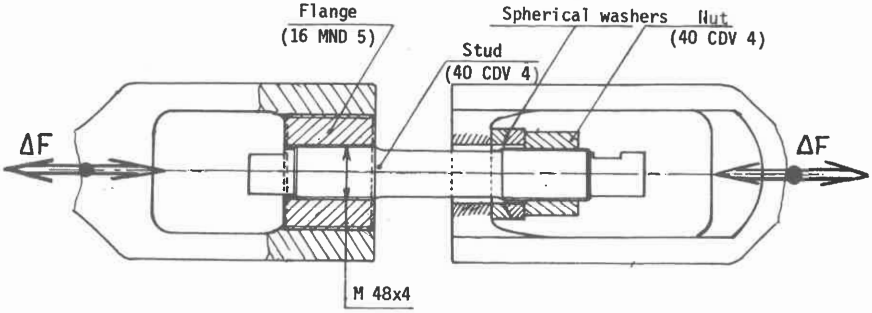


Figure 1 : Test rig

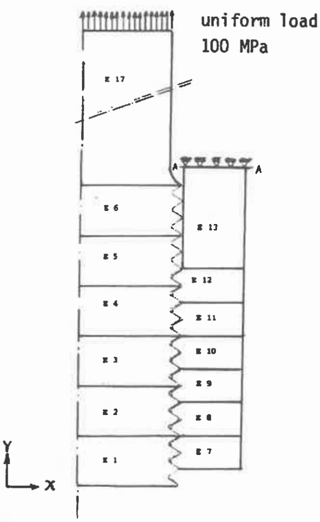


Figure 2 : Stud-to-nut connection model

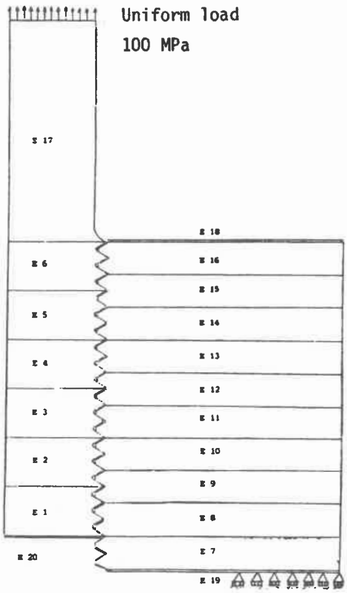


Figure 3 : Stud-to-flange connection model

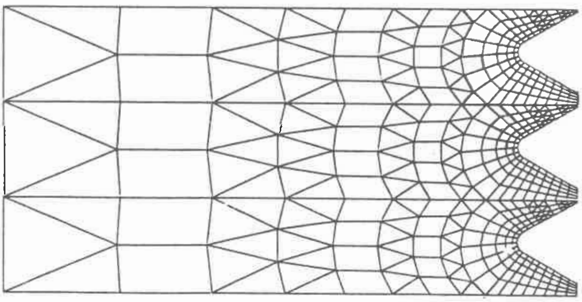

Figure 4 : Finite element mesh detail of the stud thread

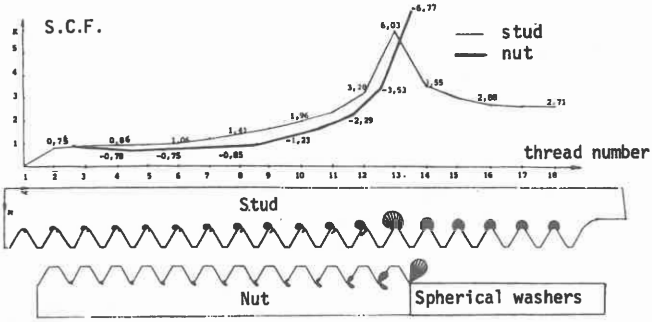


Figure 5 : Stud-to-nut connection
Stress concentration factors versus thread location

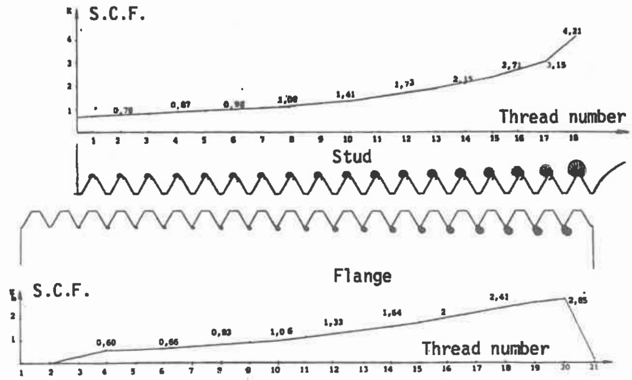


Figure 6 : Stud-to-flange connection
S.C.F. versus thread location

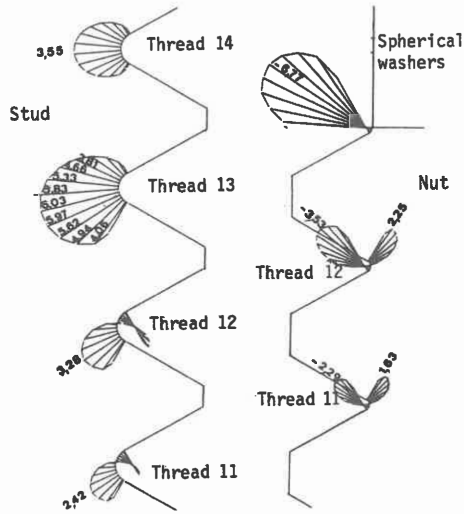


Figure 7 : Stud-to-nut connection
S.C.F. variations at the thread fillet radius

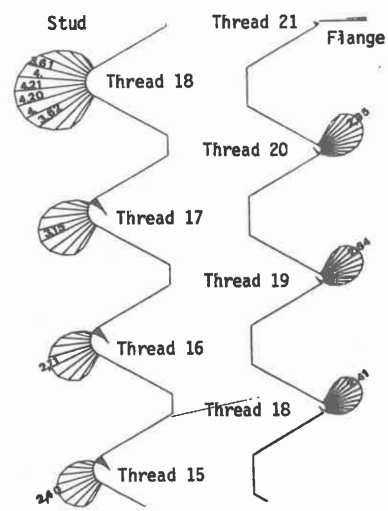


Figure 8 : Stud to flange connection
S.C.F. variations at the thread fillet radius

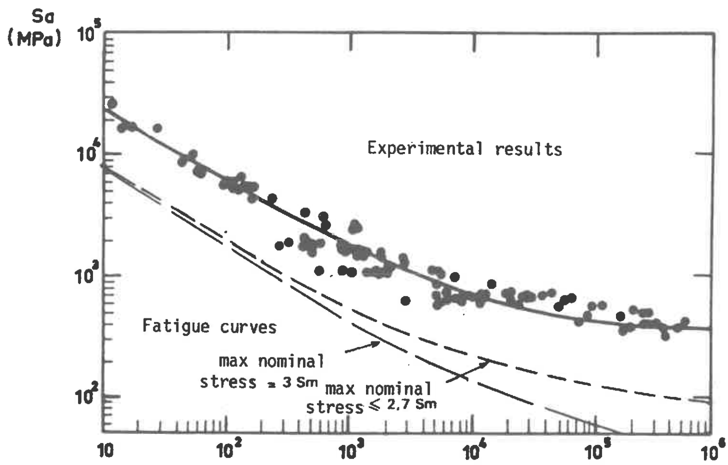


Figure 9 : Bolting fatigue curves and experimental results

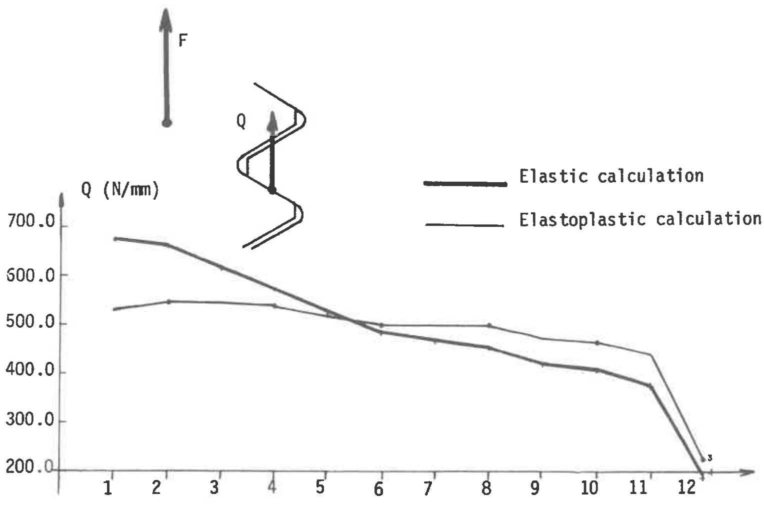


Figure 10 : Load distribution on thread side