FATIGUE EXAMINATION OF WELD SEAMS

Ervin Kerekes
Structural Analyst
NABI Rt.

Received: November 8, 2000.

ABSTRACT

The weld seams in a big structure always mean weak points, therefore we must design very carefully the position of them. It is especially true in dynamic cases. We need to make lots of calculations with very detailed model to find the best solution. The detailed model increases the requested capacity, time and costs. How could we simplify the process? Which method should we use?

Keywords: Fatigue estimation, BS5400, DIN15018, FEMFAT, Radaj, Eichlseder, Submodel analysis, FE, ANSYS

1. INTRODUCTION

In the literature and standards different calculation theories were published. In my paper I would like to give some useful information about these theories. You will see the advantages and disadvantages of them.

2. WELD SEAM ANALYSIS

2.1. Requirements

The requirements of weld seam analysis are:
- Model (FEM, BEM, FDM etc.)
- Load case definition (uni- and multi-axial load conditions, etc.)
- Material database (Non-linear material laws, test data, etc.)
- Calculation result (Stress result, strain result; time domain, frequency domain; quasi-static, transient etc.)
- Weld seam definition
- Fatigue calculation theory (BS5400 [1], DIN15018 [2], FEMFAT [4], etc.)
- Check (Tests).

2.2. The model

Force Method, Finite Element Method, Boundary Element Method, and Finite Differential Method. All of them are applicable to generate and solve a model. But, nowadays the most popular calculation theory in Hungary and in the world is the finite element method. The theory is not too old, but the estimation processes, the modeling techniques are well known. The most famous softwares (e.g.: ANSYS, NASTRAN,
PATRAN, COSMOS etc) work in this way. Practically every structural problem can be solved and the accuracy is not bad.

2.3. Load Case

This is one of the most important and hardest steps in the simulations. Tests or dynamic simulations can define it. Generally this one predestinates the fatigue calculation theory too, because some theory is not acceptable for example for multiaxial load conditions or transient results.

2.4 Material database

The fatigue tests are very expensive, because during the measurement we have to work with probability values, therefore one result is not enough. Properties for general materials can be found in the literatures and standards. In special cases we can use some proposed equations too for estimation of material properties from static attributes, but let us be careful and let us see the validity!

2.5 Calculation result

The American literatures prefer the strain based fatigue analysis, while the German (European) prefers the stress based analysis (Wöhler curves). Some theory and software can solve only the uniaxial, quasi-static problems, while others have more possibilities. Before the weld seam analysis we have to check our possibilities.

2.6. Weld seam definition

In the real structure the weld seams mean weak points. Normally, during the stress estimation this effect is not possible to respect or it requires lot of effort. Therefore we need some special techniques to assign the welds.

3. FATIGUE CALCULATION THEORIES

3.1. BS5400

This standard is the most common and popular in the British regions. The biggest advantage of it the simply calculation method, but this is the disadvantage of it too. The first step during the calculation is the weld seam definition. We can choose from the appendix for the predefined joints a detailed class (9 different classes: W, G, F2, F, E, D, C, B, S). This seems very simple, but in the reality it is not so easy.

The range (σr)-cycles(N) relationships have been established from statistical analyses of available experimental data (using linear regression analysis of log σr and log N) with
minor empirical adjustments to ensure compatibility of results between the various classes.

The equation may be written in basic (SN curve) form as:

$$N \cdot \sigma_r^m = K_0 \cdot \Delta^d$$

where

- $N$ is the predicted number of cycles to failure of stress range $\sigma_r$
- $K_0$ is the constant term relating to mean-line of the statistical analysis results
- $m$ is the inverse slope of the mean-line log($\sigma_r$) – log($N$) curve
- $\Delta$ is the reciprocal of the anti-log of the standard deviation of log($N$)
- $d$ is the number of standard deviation below the mean line.

The theory does not contain the mean stress effect! This is a big disadvantage of it.

3.2. DIN 15018 German Standard

The DIN15018 is a standard basically for Cranes, but some part is also correct for other field of the design and research. The inputs are the next: detail class, mean and amplitude stress, cycles.

In the standard there are 7 different classes: W0, W1, W2, K0, K1, K2, K3, K4. The worst class is the K4 and best is the W0. The W0 means the basic material, the W1 and W2 are for the Rivet joints, the K0-K4 classes are for different type of weld seams.

The calculation way is different from the BS5400, because the calculation method is based on the classic Smith diagrams not on SN curves (Fig.1.). Originally this method is correct for the well-known parts with well-defined loads in 2D. The standard is adopted for ANSYS and extended for general 3D model too.

![Smith diagram from the DIN15018](image)

Figure 1. Smith diagram from the DIN15018

The final damage (which is not similar to the damage estimation of BS5400) is calculated in 2D from the next form:
\[
\left(\frac{\sigma_x}{\sigma_{x\text{ lim}}}\right)^2 + \left(\frac{\sigma_y}{\sigma_{y\text{ lim}}}\right)^2 = \left(\frac{\sigma_x \cdot \sigma_y}{\sigma_{x\text{ lim}} \cdot \sigma_{y\text{ lim}}}\right) + \left(\frac{\tau}{\tau_{\text{lim}}}\right)^2 \leq 1.1
\]

where

- \(\sigma_x, \sigma_y\) are the calculated stress in x, and y direction
- \(\sigma_{x\text{ lim}}, \sigma_{y\text{ lim}}\) are the calculated limit stress in x, and y direction by DIN15018
- \(\tau\) is the calculated shear stress
- \(\tau_{\text{lim}}\) is the calculated limit shear stress by DIN15018

The limit values are modified in the function of the cycles, stress ratio \(\frac{\sigma_{\text{min}}}{\sigma_{\text{max}}}\) and the class definition.

The class definition and the damage result can be seen on Fig. 2, 3.

3.3. Theoretical background of FEMFAT

The FEMFAT is finite element based fatigue analysis software. Dr. Eichlseder started the developing the theory of this software some years ago. The traditional fatigue estimations are based on nominal stresses, simple loads, notch factors and well-defined simple parts. In a real structure these are not known. Generally we have complex assembly with complex load. We can not define the nominal stresses and the notch factors, only the local dynamic- and static stress tensors and the stress distribution in the structure.

The method of analysis based on experimental and theoretical research as well as real scale tests and experience data. The analysis is essentially a modified procedure of [7], which additionally includes the influence factors of TGL19340 [12] and FKM94 [11].

The program package starts from the tension/compression S/N curve of a specimen and this curve will be modified to local S/N curves (Wöhler curves) for the nodes of the FE model. The local S/N curves are influenced by properties of the structure and the load.
behaviour. A component s/n curve is calculated in each node of the structure for each combination of amplitude and mean stress tensor defined by the rain-flow matrix and this s/n curve is used for the accumulated damage of this spectrum matrix element. The damage is accumulated linearly, optionally according to original, modified or elementar Miner rule.

The program accounts the secondary influences such as mean stress effect (Haigh graph with cutting-plane method), local plastic stresses (Neuber hyperbole).

Further influences on the lifetime of components, such as mechanical or thermal surface treatments, are accounted by various rules and standards.

The estimation of the notch effect basing on experiments or experience is possible by helping of relative stress gradient. This method was validated for numerous examples, see e.g. [8,9,10].

### 3.3.1 FEMFAT WELD

FEMFAT-WELD is a part of the software package FEMFAT, which facilitates the prediction of whole-component failure through fracture. The following tasks have been eliminated by the program.

- Recognition of the weld seams and weld seam types from the whole FE structure of complex components
- Determination of the condition of local welds seam in space
- Analysis of the definitive structure stress component $\sigma_{\text{NORMAL}}$, $\sigma_{\text{LONGITUDINAL}}$ and $\tau$ in the local weld seam
- Fatigue strength evaluation of local weld seam area, including the relevant weld seam notch stress at the seam junctions, roots or ends.

A weld seam modelling guideline was developed in the TZS. The objective of the database for weld seam factors is to reduce the time required to built up the fine model, by using pre-analysed fine models. The local notch factors of the roots and junctions belonging to these pre-analysed models are stored in the database. The formation of the fine model is carried out according to the fixed Radaj process. The basis of the process is that the notch stress examination is based on "Mean Value/Range"- methods from KÖTTGEN, OLIVER and SEEGER, following the concept of RADAJ. In developing the database and modelling guidelines it was instead that the user could extend the database to include new individual joint types at any time.

### 3.4. “Submodel Method” Fatigue Analysis

The FEMFAT WELD is a very good and fast, but not the most accurate estimation for weld seam. The reason of it, the notch stress itself. The value of it depends on the load type, and the restraints of the model have also big influence for the results. The solving of this problem maybe the “submodel” technique with unit displacement loads.
Fig. 4 Finite element “submodel” structure with nodes

The relative stress gradient definitions are the next for load case 1, between 0-1 nodes:

\[ \chi_{11}^* = \frac{\sigma_{v01} - \sigma_{v11}}{\sigma_{v01}} \cdot \frac{1}{d_1} \]

Where \( d_1 \) is the distance between the two nodes, \( \sigma_{v01} \) is the equivalent stress in the No. 0 point in Load Case 1 and \( \sigma_{v11} \) is the equivalent stress in the No. 1 point in Load Case 1.

Similar form for node 2 to 8:

\[ \chi_{21}^* = \frac{\sigma_{v01} - \sigma_{v21}}{\sigma_{v01}} \cdot \frac{1}{d_2} \]

\[ \chi_{81}^* = \frac{\sigma_{v01} - \sigma_{v81}}{\sigma_{v01}} \cdot \frac{1}{d_8} \]

In a global form:

\[ \chi_{ij}^* = \frac{\sigma_{v0j} - \sigma_{vij}}{\sigma_{v0j}} \cdot \frac{1}{d_i} \]

(1)

where \( i = 1 \ldots \text{Max Node} \), the node number

and \( j = 1 \ldots \text{max. Load Case No.} \), the Load Case No.

The residual stresses come from the basic load cases with linear combination:

For example we have 3 load cases and we calculate the residual stress with the next equation:

\[ \left\{ \begin{array}{l} LC1 \\ LC2 \\ LC3 \end{array} \right\} \Rightarrow LC4 = A \cdot LC1 + B \cdot LC2 + C \cdot LC3 \]

The residual stresses from equivalent stresses can be written:

\[ \sigma_{v04} = A \cdot \sigma_{v01} + B \cdot \sigma_{v02} + C \cdot \sigma_{v03} \]

\[ \sigma_{v14} = A \cdot \sigma_{v11} + B \cdot \sigma_{v12} + C \cdot \sigma_{v13} \]

\[ \sigma_{v4} = A \cdot \sigma_{v1} + B \cdot \sigma_{v2} + C \cdot \sigma_{v3} \]

In general form:

\[ \sigma_{v(i+1)} = A_1 \cdot \sigma_{v1} + A_2 \cdot \sigma_{v2} + A_3 \cdot \sigma_{v3} + \chi + A_j \cdot \sigma_{vij} \]
The most simplest form for the residual stress in each node:

$$\sigma_{vi(j+1)} = \sum_{j=1}^{m} A_j \cdot \sigma_{vij}$$  \hspace{1cm} (2)

for \( i = 1..n \) and \( j = 1..m \), where \( n \) is the maximum node number and \( m \) is the maximum load case number.

The most simplest form for the residual stress in the “center” (0) node:

$$\sigma_{v0(j+1)} = \sum_{j=1}^{m} A_j \cdot \sigma_{v0j}$$  \hspace{1cm} (3)

Returning to the relative stress gradient and fill the equation (1) with equation (2) and (3):

$$\chi^*_{i(j+1)} = \frac{\sigma_{v0(j+1)} - \sigma_{vi(j+1)}}{\sigma_{v0(j+1)}} \cdot \frac{1}{d_i}$$  \hspace{1cm} (4)

The equation (4) gives us the residual relative stress gradient for the 0 points.

The Fig. 5 shows an example.

![Fig. 5. Butt-Weld, V seam](image)

The most general case we have both end of the Butt-Weld, V seam 6 DOF, - 3 translation and 3 rotation. Normally this means 12 different unit load cases. Using the symmetric condition and reducing the problem in 2D, the DOF decreases into 3. If we want to know only in one area what is the stress situation and we use only 9 nodes to model this small detail (Fig. 4), we will get relatively large database. These are the next in 9 points: the stress matrixes for 3 unit load cases (\( 9 \times 3 \times 3 \) data), 3 \( \Lambda_i \) superposition constants for the linear combination of the amplitude stress and 3 for mean stress. The final number of the data is 87 (!). This is only for 1 detailed part!

If we want to store \( k \) details for a weld seam, we need \( k*81+6 \) data.

### 3.4.1. Database decreasing for “Submodel Method” Fatigue Analysis

The residual relative stress gradient from the load cases:

$$\chi^*_{ij} = \frac{A_i \sigma_{v01} - A_i \sigma_{v11} + A_2 \sigma_{v02} - A_i \sigma_{v12} + A_3 \sigma_{v03} - A_i \sigma_{v13} + \Lambda}{A_i \sigma_{v01} + A_2 \sigma_{v02} + A_3 \sigma_{v03} + \Lambda} \cdot \frac{1}{d_i}$$

Let’s write in another form!

$$\chi^*_{ij} = \left( \frac{1}{A_2 \sigma_{v02} + A_3 \sigma_{v03} + \Lambda} \cdot \sigma_{v01} - \sigma_{v11} \right) + \left( \frac{1}{A_1 \sigma_{v01} + A_3 \sigma_{v03} + \Lambda} \cdot \sigma_{v02} - \sigma_{v12} \right) + \left( \frac{1}{A_1 \sigma_{v01} + A_2 \sigma_{v02} + \Lambda} \cdot \sigma_{v03} - \sigma_{v13} \right) \cdot \frac{1}{d_i}$$
In simpler form:

\[
\chi_{ij}^* = \left( \frac{1}{A_2\sigma_{\sigma_{12}} + A_3\sigma_{\sigma_{13}} + \Lambda} \cdot \chi_{11}^* + \frac{1}{A_1\sigma_{\sigma_{01}} + A_3\sigma_{\sigma_{03}} + \Lambda} \cdot \chi_{12}^* + \frac{1}{A_1\sigma_{\sigma_{01}} + A_2\sigma_{\sigma_{02}} + \Lambda} \cdot \chi_{13}^* + \Lambda \right) \cdot \frac{1}{d_1}
\]

The relative stress gradient in a node with this form is defined very easy, and using this form the database size decreases. Now it is enough to store the next data, if we use the earlier conditions: Von Mises stresses in each unit load cases (3 data), relative stress gradients in each direction in each load cases (8 x 3 = 24), constants for the linear combination of the amplitude and mean stress (6 data). The final number of the data is 33. This is more than twice less than the original was! For k details k x 27+6 data must be stored.

Using the cutting plate method for the mean stress effect, the stress tensors storing are also important in the detailed area in each unit load cases (3 x 3 =9). This means 39 data. For k details: k*84+12 data.

5. CONCLUSION

<table>
<thead>
<tr>
<th></th>
<th>Advantages</th>
<th>Disadvantages</th>
</tr>
</thead>
<tbody>
<tr>
<td>BS5400</td>
<td>Fast, simple; no element size sensitivity; useable for multi-axial</td>
<td>no mean stress effect; not well defined stress input</td>
</tr>
<tr>
<td></td>
<td>dynamic analysis; very simple to coding in a program language</td>
<td></td>
</tr>
<tr>
<td>DIN15018</td>
<td>more accurate than BS5400; based on better elements; very simple to coding in a program language</td>
<td>Only for loads in phase</td>
</tr>
<tr>
<td>FEMFAT WELD</td>
<td>Fast for well defined weld seams</td>
<td>Notch factors; difficult to generate a new weld seam</td>
</tr>
<tr>
<td>“Submodel” Technique</td>
<td>Easy database generation; possibility to use the original FEMFAT BASIC or MAX theory; Correct for bigger submodel too (e.g. full T joint etc.)</td>
<td>Bigger database</td>
</tr>
</tbody>
</table>

6. REFERENCES


