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Master S-N Curve Fatigue Life Prediction of a Railroad Car Bogie Based on Mesh Insensitive Structural Stress Method

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Abstract. Fatigue life prediction of a welded structure is a complex phenomenon due to the nature of fatigue and the welding process. Additionally, Finite Element Method (FEM) results are extremely sensitive to the size of elements. Therefore, it is difficult to adopt a method to estimate the fatigue life, especially for welded structures. Besides, mesh size independence is a critical issue to perform fatigue life prediction methods that eliminates the need for excessive element numbers in the mesh. This paper investigates the Master S-N Curve Approach (MCA) using the output parameters of the mesh insensitive Structural Stress Method (SSM). MCA based on SSM employs structural stresses recovered from nodal forces and nodal moments. To recover these inputs, FEM model should be established properly. Thus, boundary conditions and applied loads were prepared for the model according to the BS EN 13749:2021. The submodeling technique in ANSYS software was used to analyze the bogie structure. To justify the mesh independence for the model, different mesh sizes were tested. In a specific range for shell bodies, SSM was shown to provide sufficient mesh independence feature. Furthermore, MCA was compared with Hot Spot Stress Method and Nominal Stress Method based on their fatigue life estimations.

Introduction

Fatigue is one of the most challenging mechanisms that affects the serviceable life of components due to variable loading; thus, it has been the subject of many studies in the literature [1]. Furthermore, fatigue life prediction of welded structures is even more complicated due to the nature of the welding process. Since the welding process requires heat to combine bodies, region called Heat Affected Zone (HAZ) is formed. In these regions, the fatigue life is influenced by heat input [2]. In the design validation and verification phase, fatigue life assessment is often aided by the stress results of the Finite Element Method (FEM). Even though FEM is a useful tool, the method needs a relatively fine mesh for the fatigue life assessment procedure which can be undesirable in terms of computational time and cost-effectiveness. Following, very fine mesh usage around singular geometries such as weld toe where most fatigue cracks initiate can cause unreliable stress singularities [3].
The International Institute of Welding (IIW) recommends two methods for the estimation of fatigue life which are namely the Nominal Stress Method (NSM) and the Hot Spot Stress Method (HSSM). The NSM calculates the stress at interested areas regardless of the local stress raising effects of the welded structure but still includes the stress raising effects of the macro geometric shape of the joint. In simple components, the method can be used to determine the average stress in the weld throat or the plate at the weld toe. The mesh can be relatively coarse but one should pay attention to assure that all stress concentration effects are eliminated from the welded area. In addition, if relatively coarse mesh is used for fillets, nodal forces should be used instead of element stresses to avoid dealing with underestimated stresses [4]. If there are problematical geometries or structural discontinuity that is not comparable to a classified structural feature, the nominal stress cannot be determined. In such cases, the HSSM can be used for fatigue life evaluation. The method calculates the stress field including all stress raising effects of structural details but still excludes the effects due to the local weld profile itself. Hot spot stress can be determined employing the stress values at reference points by the extrapolation to the weld toe depending on mesh size, i.e., coarse or fine mesh. Care must be

taken that if the weld is not modeled, the extrapolation path should be extended to the structural intersection spot to avoid stress underestimation caused by the missing stiffness of the weld [4]. These methods need to be classified according to the geometry, fatigue loading mode, or connection type. As a result of the classification process, a related fatigue strength curve should be chosen. Thus, the assessment may lead to misjudgment that will cause erroneous fatigue life prediction.

This paper investigates the adoption of FEM for the Master S-N Curve approach of a railroad car bogie based on mesh insensitive Structural Stress Method (SSM). The mesh-insensitive SSM is developed by Battelle [5]. The method can be used in both shell and solid element models with relatively coarse mesh. The benefit of this approach is to keep the designer away from stress singularities and also to provide computational time and cost efficiency. Later, the Master S-N Curve Method was developed by Dong et al. [6]. This method is an alternative approach to estimating the fatigue life strength of welded components. The method uses recovered structural stresses from nodal forces and nodal moments. Furthermore, the method eliminates the misjudgment of the welding classification process and also enables the usage of relatively coarse mesh models.

In this study, a railroad car bogie 3D model was simplified and turned into a shell model at first. Then, boundary conditions and loads were defined for FEM analyses using ANSYS software according to BS EN 13749:2021 [7]. After analysis of a full model, the FEM model was reduced into a submodel for the areas of interest. Using the submodel, analyses were repeated for different mesh sizes to ensure that the mesh independence feature of the SSM was guaranteed. After that, related fatigue life was calculated utilizing the SSM and ASME B&PV Code, Section VIII, Div2. C & h parameters [8]. Lastly, the Master S-N Curve was compared with the NSM and HSSM in terms of estimated fatigue lives.

Materials and Methods

The bogie is made of S355 steel. Material specifications are given in *Table 1*.

Table 1. Material Specifications

Properties	Value	Units
Density	7850	kg/m^3
Young's Modulus	200	GPa
Poisson's Ratio	0.3	-
Tensile Yield Strength	355	MPa
Tensile Ultimate Strength	550	MPa

Firstly, a railroad car bogie was simplified for FEM analyses in ANSYS v22 software. Then, boundary conditions and loads were defined according to BS EN 13749:2021. Following, the methods were adopted for the fatigue life evaluation which are NSM, HSSM, and Master S-N Curve based on mesh insensitive SSM.

Railroad Car Bogie Finite Element Method. Before starting the FEM analyses, the railroad car bogie model in *Figure 1* should be simplified. However, during the simplification process, one should care about keeping all the structural details. Because if these details were neglected, it might lead to misjudgment. In addition, through the simplified model, the meshing and calculation process will become easier. If the thickness of a solid model is much smaller than its length, it can be modeled as a shell. Following, the railroad car bogie model is suitable to turn into a shell model except for supports.

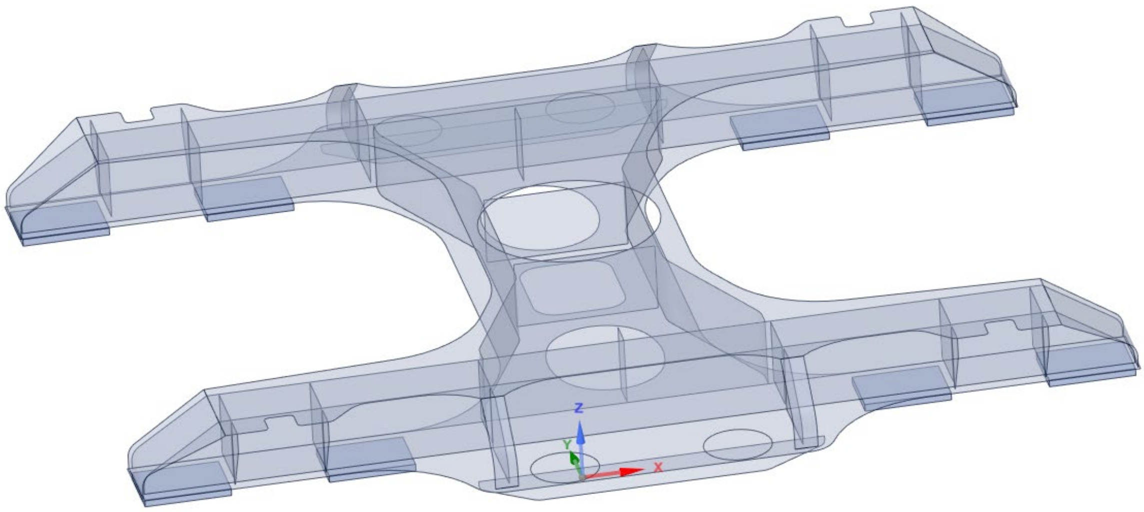
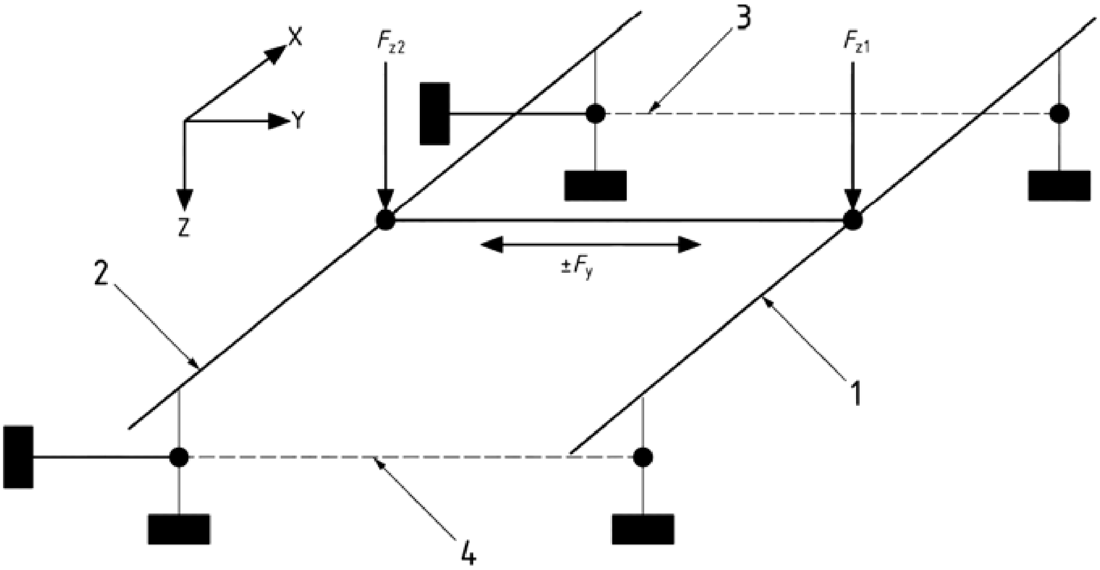


Fig. 1. Simplified railroad car bogie model

Afterwards, boundary conditions and loads should be defined according to the requirements of interest. This study takes into account the standard of BS EN 13749:2021 which includes the method to specify the structural requirements of bogie frames. According to the standard, boundary conditions and load scenarios were defined in *Figure 2* and *Table 2*, respectively.



Key

1 side 1 2 side 2

3 axle 1 4 axle 2

Fig. 2. Boundary conditions [7]

Table 2. Load scenarios [7]

Load case	F_{z1}	F_{z2}	F_y
1	$F_z/2$	$F_z/2$	0
2	$(1+\alpha-\beta)F_z/2$	$(1-\alpha-\beta)F_z/2$	0
3	$(1+\alpha-\beta)F_z/2$	$(1-\alpha-\beta)F_z/2$	$+F_y$
4	$(1+\alpha+\beta)F_z/2$	$(1-\alpha+\beta)F_z/2$	0
5	$(1+\alpha+\beta)F_z/2$	$(1-\alpha+\beta)F_z/2$	$+F_y$
6	$(1-\alpha-\beta)F_z/2$	$(1+\alpha-\beta)F_z/2$	0
7	$(1-\alpha-\beta)F_z/2$	$(1+\alpha-\beta)F_z/2$	$-F_y$
8	$(1-\alpha+\beta)F_z/2$	$(1+\alpha+\beta)F_z/2$	0
9	$(1-\alpha+\beta)F_z/2$	$(1+\alpha+\beta)F_z/2$	$-F_y$

where F_z, F_{z1}, and F_{z2} are vertical forces, F_y is transverse force, α is roll coefficient, β is bounce coefficient and μ is adhesion or friction coefficient [7].
In this study, only the first load case was considered. Hence, *Fz* needs to be calculated.

$$F_{z1} = F_{z2} = \frac{F_z}{2} = \frac{(M_v - 2m^+)g}{4} \qquad (1)$$

where M_v is the mass of the vehicle in working order, m^+ is the bogie mass without any secondary spring masses and g is the acceleration due to gravity [7].
Because the railroad car bogie model is huge, a submodeling technique can be used to reduce the requirement for computational resources for future analyses. *Figure 3* shows the FE submodel.

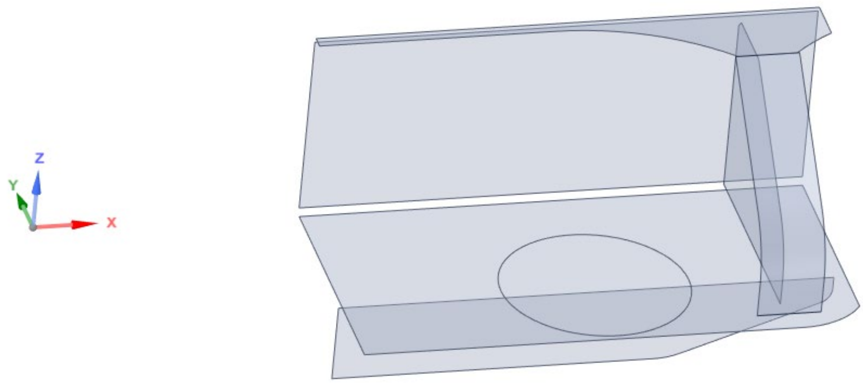


Fig. 3. Railroad car bogie FE submodel

Submodel mesh sizes, type of elements, total number of nodes, and elements are shown in *Table 3*.

Table 3. Submodel mesh details

Mesh Size [mm]	Element Type	Total Number of Nodes	Total Number of Elements
10	Linear Quad & Tri	5715	5334
8	Linear Quad & Tri	8677	8205
4	Linear Quad & Tri	33884	32940
2	Linear Quad & Tri	132098	130240

International Institute of Welding Fatigue Life Evaluation Methods. IIW offers two fatigue life estimation methods which are NSM and HSSM. For the simpler geometries, NSM can be used to determine the fatigue life. In other cases, i.e., problematical geometries or structural discontinuity, HSSM may be used to evaluate the fatigue life.

Nominal Stress Method. In simple components, the nominal stress can be calculated using fundamental theories of structural mechanics based on linear-elastic behavior. The average stress component can be determined in the weld throat or the plate at the weld toe using NSM [4].

$$\sigma_W \text{ or } \tau_W = \frac{F}{A_W} = \frac{F}{a \, . \, l_W} \tag{2}$$

where σ_W or τ_W is the stress in weld throat a for a weld of length l_W and F is the force in weld.
In other cases, i.e., hyperstatic structures or macrogeometric discontinuities, FEM can be used. In FEM aided approach, a path should be defined that extends to the weld toe. Afterwards, the nominal stress will be the stress that the graphic remains linear. A related demonstration is given in *Figure 4*.

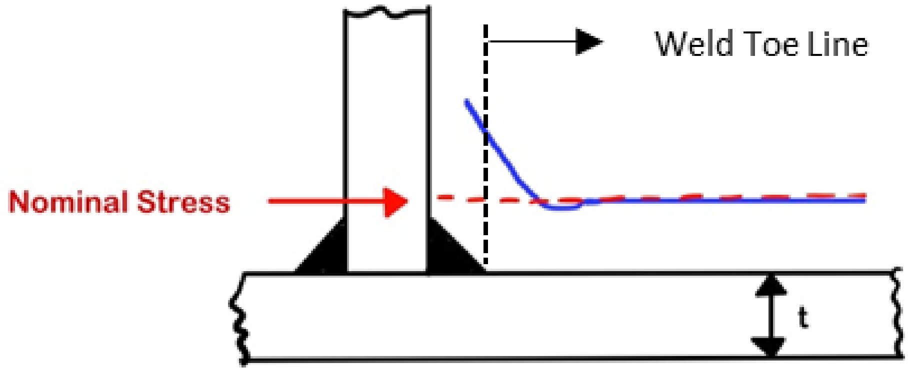


Fig. 4. Nominal stress definition

Hot Spot Stress Method. The hot spot stress can be calculated using reference points by extrapolating to the weld toe area of interest at those points. *Figure 5* shows those points.

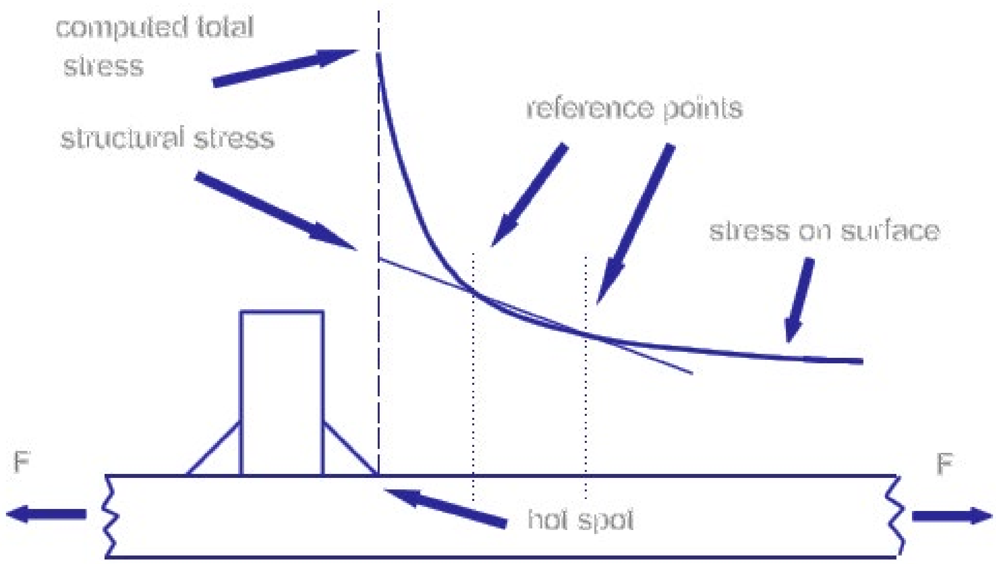


Fig. 5. Hot spot stress definition [4]

Like the nominal stress, a path should be drawn extending to the weld toe for the hot spot stress. An illustration of the process is shown in *Figure 6* for both shell and solid elements.

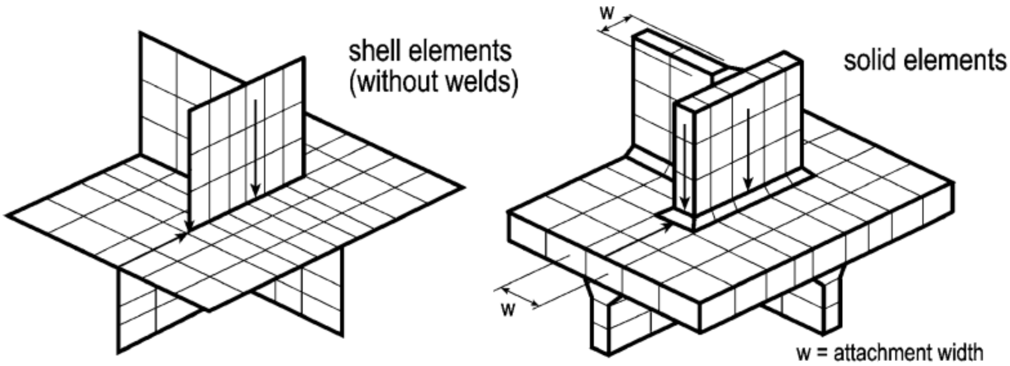


Fig. 6. Extrapolation paths [4]

There are two types of hot spot stress as shown in *Figure 7*.

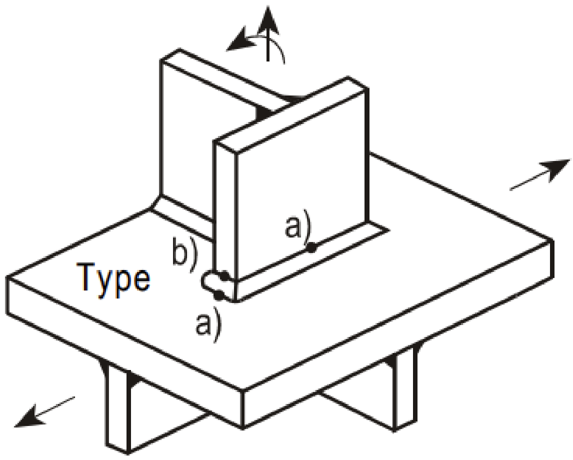


Fig. 7. Types of hot spot stress [4]

Depending on the mesh element sizes, two or three reference points can be used. The following, steps for the determination of hot spot stress using the reference points and extrapolation equations are given below for type a) hot spot stress [4]:

1. Mesh element length is no more than 0.4*t (t: thickness) at the hot spot using linear elements:

$$\sigma_{HSS} = 1.67 * \sigma_{0.4*t} - 0.67 * \sigma_{1.0*t} \tag{3}$$

2. Mesh element length is no more than 0.4*t at the hot spot. Recommended for cases of pronounced non-linear structural stress increase towards the hot spot, sharp direction change of applied force, or for thick-walled structures with linear elements.

$$\sigma_{HSS} = 2.52 * \sigma_{0.4*t} - 2.24 * \sigma_{0.9*t} + 0.72 * \sigma_{1.4*t} \tag{4}$$

3. Mesh element length is equal to the thickness of the plate with quad elements at the hot spot.

$$\sigma_{HSS} = 1.50 * \sigma_{0.5*t} - 0.50 * \sigma_{1.5*t} \tag{5}$$

Using the proper equation given above, fatigue life will be estimated.

Master S-N Curve based on Mesh Insensitive Structural Stress Method. Master S-N Curve is a useful technique for the estimation of fatigue life. The methods recommended by IIW, i.e., NSM and HSSM, require a classification process by welding types. However, the Master S-N Curve provides a unique solution for fatigue life estimation regardless of welding type [6]. In addition to that, implementing the mesh-insensitive SSM to the Master S-N Curve offers a mesh-insensitive unique solution.

Mesh Insensitive Structural Stress Method. Battelle developed a method for both shell and solid bodies to provide mesh independence [5].

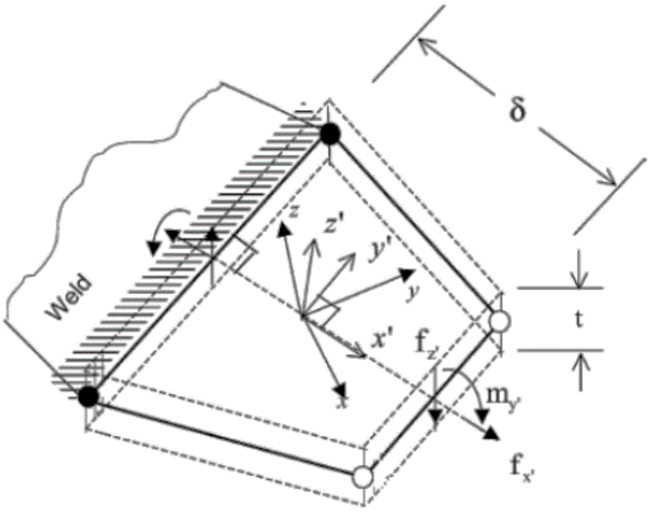


Fig. 8. Illustration of mesh insensitive SSM for an element [5]

As shown in *Figure 8*, models are often defined in the global coordinate system (*x, y, z*) depending on the finite element codes used. To be able to apply the method, components in *Eq. 6* should be calculated using the local coordinate system (*x', y', z'*). Local *x'* is perpendicular and *y'* is parallel to the weld direction. [9]. According to the defined local coordinate system, structural stress components are substituted as:

$$\sigma_S = \sigma_m + \sigma_b = \frac{f_{x'}}{t} + \frac{6\left(m_{y'} + \delta * f_{z'}\right)}{t^2} \tag{6}$$

where σ_s is the structural stress, σ_m is the membrane stress component, σ_b is the bending stress component, $f_{x'}$ and $f_{z'}$ are the sectional forces along *x'* and *z'* axis respectively, $m_{y'}$ is the sectional moment about *y'* axis and *t* is the thickness of the related element.

In the case of cyclic stress, one must multiply *Eq. 6* by (1-R), where *R* is the loading ratio, resulting structural stress parameter becomes [3]:

$$[\Delta\sigma_S] = [\Delta\sigma_m] + [\Delta\sigma_b] \tag{7}$$

Demonstration of mesh independence of the SSM, four different mesh sizes, i.e., 2mm, 4mm, 8mm, and 10mm, were employed in the submodel of the railroad car bogie. Structural stress components that are given in *Eq. 6* were calculated using the Mechanical APDL Module of ANSYS v22.

Master S-N Curve. Once the mesh independence is assured, outputs of SSM are then can be utilized for the derivation of a unique fatigue strength definition called an equivalent structural stress parameter, ΔS_S [3]. Related loading condition defined by Dong et al. [6] as:

$$\Delta S_S = \frac{\Delta\sigma_S}{t^{*\frac{2-m}{2m}}\, I\,(r_b)^{\frac{1}{m}}} \tag{8}$$

$$r_b = \frac{|\Delta\sigma_S|}{|\Delta\sigma_m| + |\Delta\sigma_b|} \tag{9}$$

where $\Delta\sigma_S$ is the structural stress parameter given in *Eq. 7*, *m* is the slope of Paris-Law in log-log scale generally taken as 3.6, t^* is the plate thickness ratio ($t^* = t/t_{ref}$ with $t_{ref} = 1$mm), r_b is the bending ratio. $I(r_b)$ is the dimensionless integral function of the bending ratio calculated as [8]:

$$I(r_b)^{1/m} = 0.0011 * r_b^6 + 0.0767 * r_b^5 - 0.0988 * r_b^4 + 0.0946 * r_b^3 + 0.0221 * r_b^2 + 0.014 * r_b + 1.2223 \tag{10}$$

for displacement-controlled conditions, and

$$I(r_b)^{1/m} = 2.1549 * r_b^6 - 5.0422 * r_b^5 + 4.8002 * r_b^4 - 2.0694 * r_b^3 + 0.561 * r_b^2 + 0.0097 * r_b + 1.5426 \tag{11}$$

for load-controlled conditions.

Afterwards, using the equivalent structural stress parameter ΔS_S, cycles to failure N can be evaluated by *Eq. 12* as:

$$\Delta S_S = C * N^h \tag{12}$$

where C and h are determined based on linear regression analysis. The following parameters provided by ASME [8] are shown in *Table 4*.

Table 4. Master S-N curve parameters according to the ASME [8]

Statistical Basis	C	h
-3 STD	11577.9	-0.3195
-2 STD	13875.7	-0.3195
-1 STD	16629.7	-0.3195
Mean	19930.2	-0.3195
+1 STD	23885.8	-0.3195
+2 STD	28626.5	-0.3195
+3 STD	34308.1	-0.3195

Result and Discussion

Firstly, analyses were carried out for different mesh sizes, i.e., 10mm, 8mm, 4mm, and 2mm to assure mesh independence. *Figure 9* shows the maximum equivalent von Mises stress results.

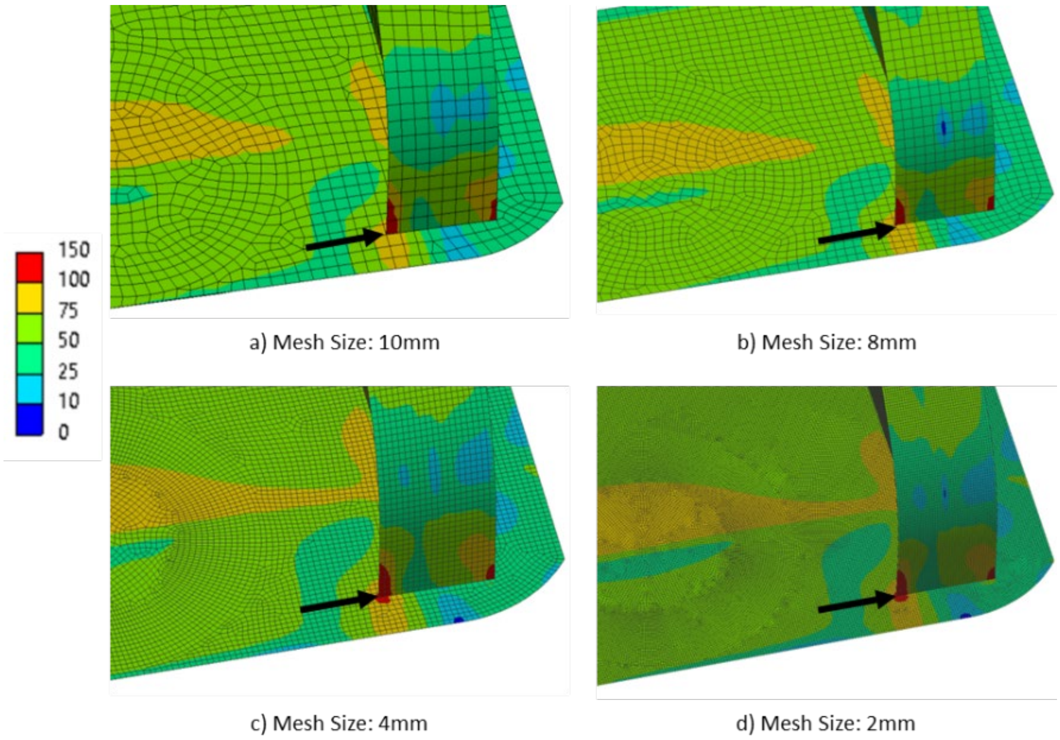


Fig. 9. FEM equivalent von Mises stress results

Nodal forces and nodal moments were obtained from ANSYS Mechanical APDL Module v22. Substituting these results in *Eq. 6* gave the corresponding structural stresses for each mesh size. Mesh convergence results are shown in *Table 5*.

Table 5. von Mises stress & structural stress results

Element Size [mm]	Structural Stress [MPa]	von Mises Stress [MPa]	Structural Stress Variation [%]*	von Mises Stress Variation [%]*
10	79,056	235,86	-2,447	-21,531
8	80,628	256,07	-0,508	-14,807
4	82,574	321,54	1,893	6,974
2	81,901	388,84	1,062	29,364

* Variation [%] = ((Stress – Average Stress) / Average Stress) * 100

After the mesh independence was achieved, using the SSM parameters in the given equations in *Section 0*, Master S-N Curve fatigue life estimation results could be substituted. Also, the NSM and the HSSM results were evaluated using the above-mentioned procedures which are presented in *Section 0* and *0*, respectively. All results are shown in *Table 6*.

Table 6. Numerical fatigue life estimation results

Method	Fatigue Life Estimation [Cycle]
Nominal Stress	3,50E+05
Hot Spot Stress	1.50E+06
Master S-N Curve, +3 STD	5E+07
Master S-N Curve, +2 STD	2.84E+07
Master S-N Curve, +1 STD	1.61E+07
Master S-N Curve, Mean	9.14E+06
Master S-N Curve, -1 STD	5.18E+06
Master S-N Curve, -2 STD	2.94E+06
Master S-N Curve, -3 STD	1.67E+06

According to *Table 6*, fatigue life estimation results show that the NSM is the most conservative approach among the three methods as expected. The Master S-N Curve Method provided a systematic distribution between +3 STD and -3 STD. The HSSM fatigue life estimation is in parallel with the Master S-N Curve negative standard deviation results.

Conclusion

In this paper, the mesh independence of SSM and a unique fatigue life prediction approach named Master S-N Curve were investigated. Using different mesh sizes, mesh independence was obtained. A unique fatigue life estimation was provided by the Master S-N Curve regardless of weld or load type classification likewise the NSM and HSSM. The results of this study can be verified by testing a railroad car bogie in service conditions that would provide a more realistic perspective. Using this perspective, an absolute comparison could be made to determine the accuracy of these methods in real service conditions. Additionally, the process can be expanded for a solid body using proper procedures.

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