A PARAMETRIC STUDY ON FATIGUE LIFE BEHAVIOR OF SPOT WELDED JOINTS

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ABSTRACT

Despite the availability of other joining methods such as laser beam welding, bolted, riveted, and adhesive bonding, resistance spot welding (RSW) remains the primary method to join panels especially in automotive, railroad, and airplane structures, which contain hundreds, even thousands of spot welds.

In service, mechanical components usually experience cyclic loading. This makes fatigue failure prevention the foremost design requirement. Considering that spot welds provide localized connection, they lead to stress concentration and consequently to initiation of fatigue cracks. The designer should then be able to predict the fatigue life of spot welded joints and determine the impact of design changes on fatigue resistance.

The objective of this study is to examine the effects of design parameters for spot welded joints on their fatigue life. For this purpose, stress analyses of spot welded plates were carried out using a finite element method for various plate geometries and spot weld diameters. Based on the predicted stress and strain states, fatigue life analyses were performed. The results of this study will provide the designer with some guidelines in designing spot welded joints.

Keywords: ANSYS, Fatigue failure, FEM, MTS, Spot weld, TS.

1. INTRODUCTION

The resistance spot welding (RSW) is the most important joining method for joining sheets of metal. According to studies, about 90 % of the welds used in an automotive body assembly are RSWs. A typical vehicle contains more than 3000 spot welds [1-6]. This number may reach 5000 and sometimes 8000 in couch and bus bodies. The advantages of using spot welding are that it is a quicker joining technique, no filter material is required, and that the low heat input implies less risk for altered dimensions during welding [1-5, 7-9].

Because a spot weld provides a localized connection it is a source of stress concentration, and thus fatigue crack initiation under fluctuating loading. For instance, the failure of automotive components takes places in approximately 80 % of all cases close to or around a spot weld, in 15 % of the cases failure is due to a low-quality-cut edge and only in 5 % fatigue cracks initiate in sheet metal [10]. As a result, it can be concluded that under normal service conditions the fatigue strength of spot-welded joints determines the durability of the whole structure.

In order to determine the fatigue life behavior of spot welded structures, accurate stress analysis and then systematic fatigue strength evaluation are needed. Different approaches are available for fatigue life assessment such as the nominal stress approach, the structural or hot-spot stress approach [11], the notch stress approach which takes into account stress concentration effects [9, 11-14], the notch strain approach [9, 11, 13, 15-17], volumetric fatigue approach [11, 18] and finally the crack propagation or fracture mechanics approach [8, 9, 11, 19-22].

In this study, simply the fatigue failure behavior of spot welded modified tensile shear (MTS) and tensile shear (TS) specimens were analyzed numerically. In order to see the effects of the spot weld diameters on fatigue life values, the analyses were repeated for different spot weld diameters for both MTS and TS specimens. Firstly a FE analysis was carried out using a commercial software, ANSYS, to determine the stress and strain states within the specimens. The material nonlinearity, plastic deformations, and residual stresses and strains developed after unloading were taken into account. Then, based on the predicted stress and strain states, fatigue analyses were performed using several models for fatigue life assessment.

2. MODELS FOR FATIGUE LIFE ASSESSMENT

Different fatigue life prediction techniques have been suggested in literature. They are classified as strain-based and stress-based approaches in general.

2.1 Stress-Based Approaches

These are the earliest, but the most widely used approaches. They are based on the assumption that the range of values for stress controls the fatigue behavior of a component. They involve empirical relations between uniaxial fully reversed stress and fatigue life (S-N curves), which are obtained through the standard rotating bending testing. However, mechanical components are usually subject to stresses oscillating about an average stress level. For these cases, a number of models were proposed [23-25]:

$$\frac{S_a}{S_f}+\frac{S_m}{S_{ut}}=1 \text{ (Modified Goodman, England-1899)} \tag{1}$$

$$\frac{S_a}{S_f}+\frac{S_m}{\sigma_f}=1 \text{ (Morrow, USA-1960s)} \tag{2}$$

where S_a is the alternating stress , S_m is the mean stress, S_f is the fully reversed fatigue strength of the specimen, S_{ut} is the ultimate tensile strength. There is a controversy on how the equivalent stress should be calculated. Use of the maximum principal stress

$$S_{ea}=\sigma_{a1} \tag{3}$$

maximum shear stress,

$$S_{ea} = \sigma_{a1} - \sigma_{a3} \tag{4}$$

or von-Mises stress

$$S_{ea} = \frac{1}{\sqrt{2}}\sqrt{(\sigma_{a1} - \sigma_{a2})^2 + (\sigma_{a1} - \sigma_{a3})^2 + (\sigma_{a2} - \sigma_{a3})^2} \tag{5}$$

were suggested as representative of equivalent alternating stress, S_{ea}. Here, σ_{a1}, σ_{a2} and σ_{a3} are principal alternating nominal stresses with $\sigma_{a1} > \sigma_{a2} > \sigma_{a3}$. For equivalent mean stress, S_{em}, similar suggestions were made [26]. For example, use of von Mises stress,

$$S_{em} = \frac{1}{\sqrt{2}}\sqrt{(\sigma_{m1} - \sigma_{m2})^2 + (\sigma_{m1} - \sigma_{m3})^2 + (\sigma_{m2} - \sigma_{m3})^2} \tag{6}$$

or sum of the normal mean stresses

$$S_{em} = \sigma_{m1} + \sigma_{m2} + \sigma_{m3} = \sigma_{mx} + \sigma_{my} + \sigma_{mz} \tag{7}$$

were suggested as representative of equivalent mean stress, S_{em} [26]. Equivalent stress approaches are extensions of the yield criteria to fatigue.

2.2 Strain-Based Approaches

According to these models, the range of values for strain controls fatigue life. They also take into account the effect of plastic strain. They are therefore especially suitable for cases where plastic effects dominate fatigue behavior. Although most engineering structures are designed such that the nominal stresses remain elastic, stress concentrations often cause plastic strains to develop in the vicinity of notches, e.g. spot welds in our case. Fatigue cracks usually nucleate due to plastic straining at the notches. The total strain amplitude can be resolved into elastic and plastic strain components, each of which was shown to be correlated with fatigue life in a linear fashion using log-log scale for most metals [26].

One method, referred to as "Morrow's mean stress method," can be written as [11, 24, 26];

$$\frac{\Delta\varepsilon}{2} = \varepsilon_a = \frac{\Delta\varepsilon_e}{2} + \frac{\Delta\varepsilon_p}{2} = \frac{\sigma'_f - \sigma_m}{E}\left(2N_f\right)^b + \varepsilon'_f\left(2N_f\right)^c \tag{8}$$

where σ'_f is the fatigue strength coefficient, ε'_f is the fatigue ductility coefficient, and *b* and *c* are exponents determined by experiments, and σ_m is the mean stress.

In the case of multiaxial stress and strain state, equivalent alternating strain is calculated [24, 26, 27] using either maximum principal strain,

$$\varepsilon_{qa} = \varepsilon_{a1} \tag{9}$$

or maximum shear strain

$$\varepsilon_{qa} = \frac{\varepsilon_{a1} - \varepsilon_{a3}}{1+\nu} \tag{10}$$

or octahedral shear strain

$$\varepsilon_{qa} = \frac{\sqrt{(\varepsilon_{a1} - \varepsilon_{a2})^2 + (\varepsilon_{a2} - \varepsilon_{a3})^2 + (\varepsilon_{a3} - \varepsilon_{a1})^2}}{\sqrt{2}(1+\nu)} \tag{11}$$

where ε_{a1}, ε_{a2}, and ε_{a3} are principal alternating strains with $\varepsilon_{a1} \rangle \varepsilon_{a2} \rangle \varepsilon_{a3}$.

Another equation suggested by Smith, Watson, and Topper (often called “SWT parameter”) is given by [16, 25, 26]

$$\sigma_{max}\varepsilon_a E = (\sigma'_f)^2 (2N_f)^{2b} + \sigma'_f \varepsilon'_f E (2N_f)^{b+c} \tag{12}$$

In addition to these equations, there are two more universal relations which are used in fatigue life calculations proposed in the literature based on experimental data [16, 26]. These relations are “Muralidharan & Manson” and “Brinell hardness” equations which written in the following respectively.

$$\frac{\Delta\varepsilon}{2} = \varepsilon_a = 0.623\left(\frac{S_{ut}}{E}\right)^{0.832} (2N_f)^{-0.09} + 0.0196(\varepsilon_f)^{0.155}\left(\frac{S_{ut}}{E}\right)^{-0.53} (2N_f)^{-0.56} \tag{13}$$

$$\frac{\Delta\varepsilon}{2} = \frac{4.25(HB)+225}{E}(2N_f)^{-0.09} + \frac{0.32(HB)^2 - 487(HB) + 191000}{E}(2N_f)^{-0.56} \tag{14}$$

3. NUMERICAL ANALYSIS

3.1 Model Descriptions

In the FE model, a 3D 10-node tetrahedral solid element, SOLID92, was used for the base metal. This element has plasticity, stress stiffening, large deflection, and large strain capabilities. On the other hand, the nugget was modeled using a two-node beam element, BEAM188. Contact and target elements, Targe170 and Conta175, were created on the inner surfaces of the plates around the spots. The beam element, BEAM188, is based on Timoshenko beam theory. Shear deformation effects are therefore included, which are especially important for short beams. This element has six degrees of freedom at each node. Because the nugget experiences low stresses, its material model was chosen as linearly elastic.

In welding process, heat treatment does not affect the mechanical properties of the material. Therefore, it is necessary to use the same material properties for the nugget and base metal.

Hence in this study, base metal and spot weld nugget are modeled using the same material properties which young’s modulus and poisson’s ratio are choused as 2.07E5 Mpa and 0.25 respectively.

The dimensions of the MTS and TS specimen are shown in Figures 1 and 2 respectively.

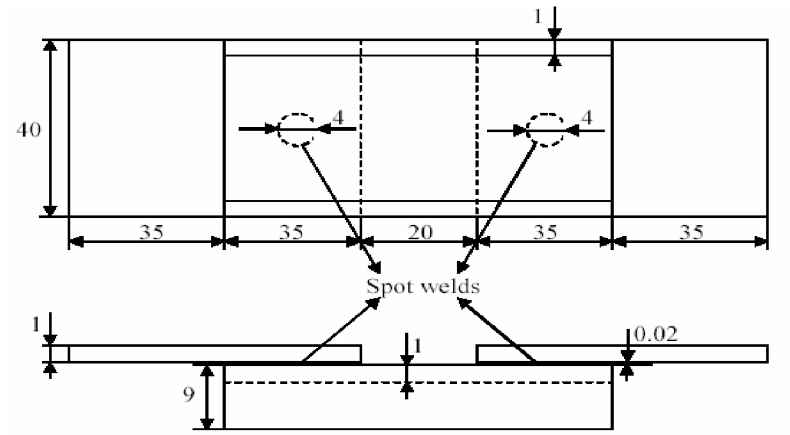

Figure 1. Geometry of MTS specimen (side and top views).

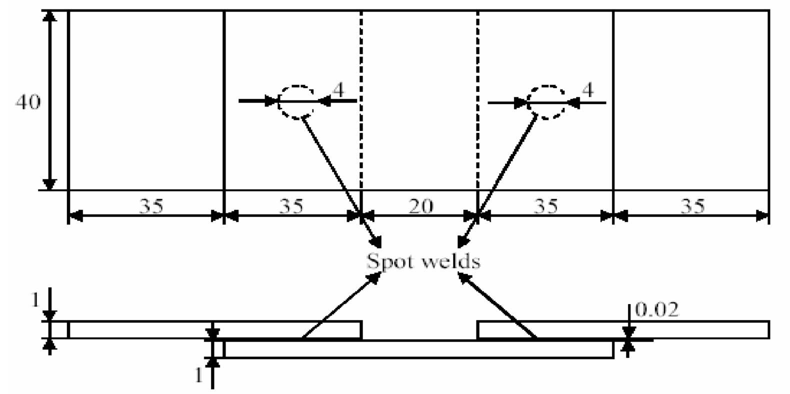

Figure 2. Geometry of TS specimen (side and top views).

3.2 Boundary Conditions

In the finite element models: All displacement and rotation degrees of freedom of the nodes on the surface of the left end of the MTS and TS were fixed: All degrees of freedom on the right end are also fixed except the loading direction. A tensile load is applied in-plane to the right end. The cyclic loading was applied in two load steps. First, the load was incrementally increased to its maximum value, $F_{\max}$, and the resulting stress state was obtained. In the second load step, the load was incrementally decreased to its minimum value, $F_{\min}$. In the next load cycles, stresses were assumed to fluctuate between the stress levels corresponding to maximum and minimum loads.

3.3 Analysis and Calculation of Fatigue Lives

After modeling the so called MTS and TS specimens using the above material data and techniques analysis was done.

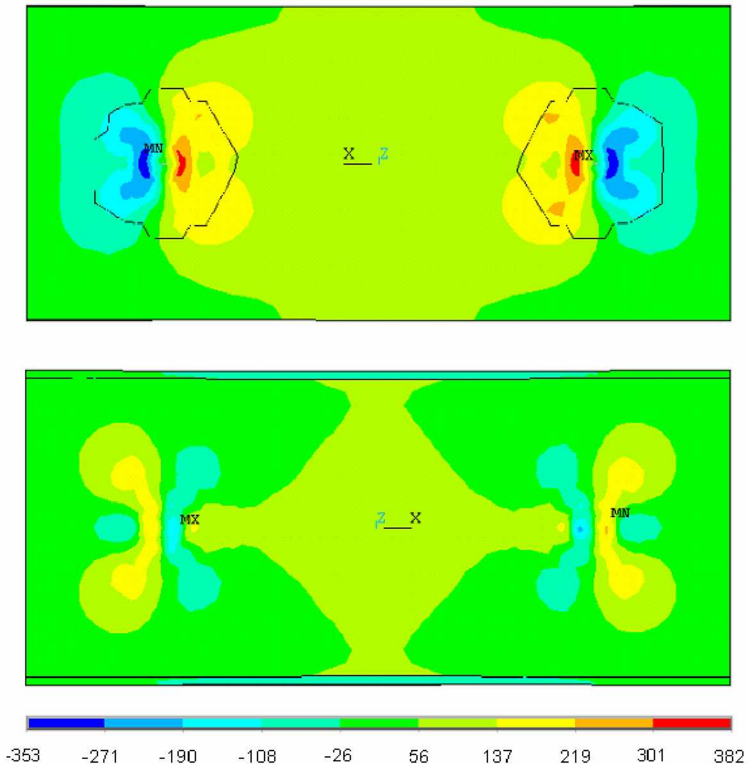

Figure 3. The distribution of σ_{xx} component (in MPa) of stress on the inner and outer surfaces of the central plate developed due to maximum load for MTS.

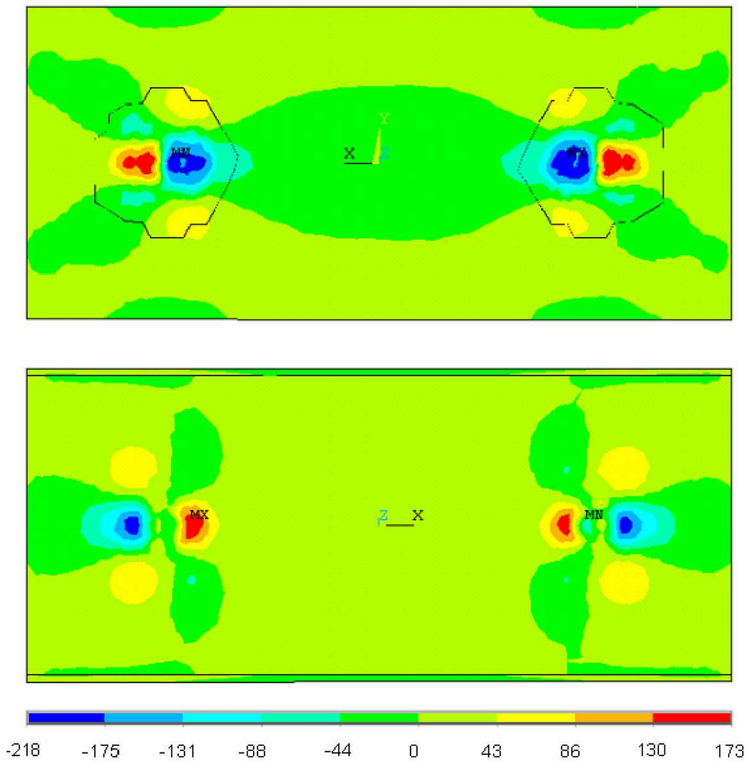

Figure 4. The distribution of σ_{xx} component (in MPa) of stress on the inner and outer surfaces of the central plate developed due to minimum load for MTS.

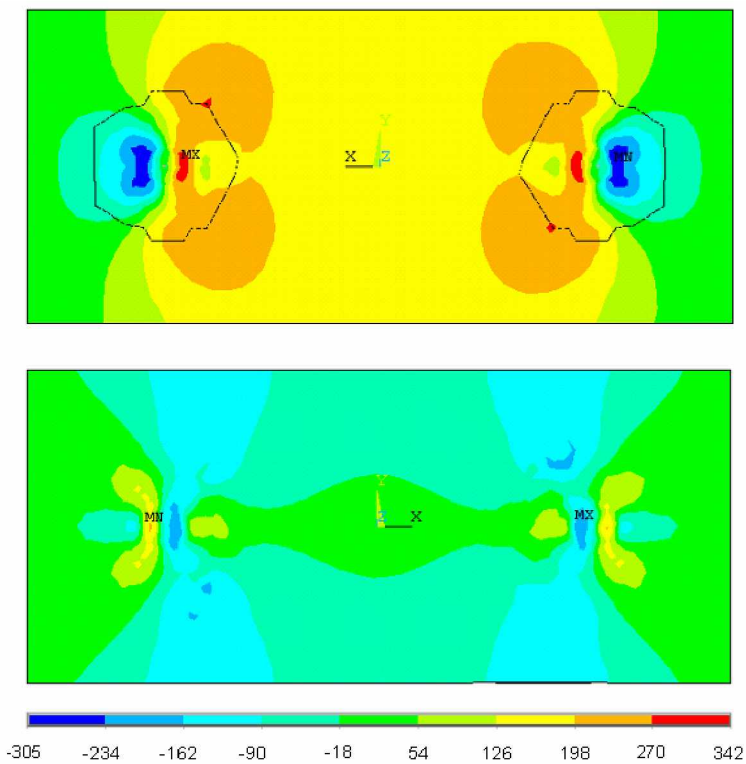

Figure 5. The distribution of σ_{xx} component (in MPa) of stress on the inner and outer surfaces of the central plate developed due to maximum load for TS.

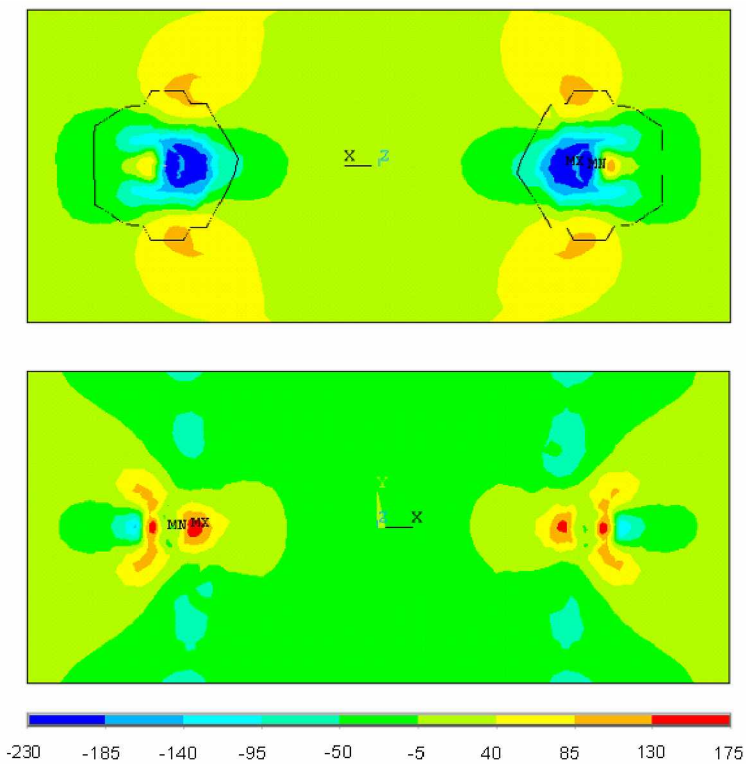

Figure 6. The distribution of σ_{xx} component (in MPa) of stress on the inner and outer surfaces of the central plate developed due to minimum load for TS.

Figures 3, 4, 5, and 6 show the distributions of σ_{xx} component of stress on the inner and outer surfaces of the central plate for MTS and TS geometries, having 4 mm spot diameter, respectively developed after the maximum (2700 N) and minimum (150 N) loads are applied. High stresses develop at regions on the inner surfaces of the sheets close to the peripheries of the spot weld nuggets because of stress concentration. Since minimum load is quite low (150 N), significant stresses existing after unloading may only be attributes to residual stresses developing due to nonuniform plastic deformation. Although load transfer occurs through the spot-weld joint, the nugget is subject to low stresses since its thickness is large in comparison to the sheet thickness. Besides, the nugget is mainly subject to shear loading and the plate is mainly subject to bending moment. As a result, bending induces larger stresses. Due to the effect of bending, stresses change from tension to compression through the thickness. The peak tensile stress develops close to the spot weld but not on its circumference. This location also conforms to the fatigue crack initiating sites observed in the experiments.

4. RESULTS AND DISCUSSION

First, the stress and strain values of the critical nodes of spot welded MTS and TS for different spot weld diameters were calculated using a nonlinear finite element analysis (FEM). When these calculations were done, it is clear that a stress concentration or singularity exists at the interception of the nugget boundary with the interface of joined sheets because all Von Misses stresses, tensile stresses, bending stresses etc. have their maximum magnitudes near the nugget boundary (that is the inner surfaces of flanged or center pieces of MTS and TS specimens on the peripheries of the spots). These stress distributions explain successfully the phenomenon that why the fractures are generally first created around the nugget for the spot-welded joints. Peak tensile-stress values on x, y, and z directions were found at the internal edges of the spot welds, while peak compression-stress values were at the external edges of the spot welds. Hence it can be said that the high tensile stresses together with the high tearing stresses are all located at the edges of the spot welds in the spot-welded joints, and will cause bending of the lap zones of the specimens even if this bending values in tensile shear specimens (TS) are not so critical and important as for modified tensile shear specimens (MTS) because of flanged pieces or centerpieces of MTS specimens.

Using different types of fatigue life prediction techniques typical spot weld fatigue results for different geometry and spot weld diameters are presented through Figures 7, 8, 9 and 10.

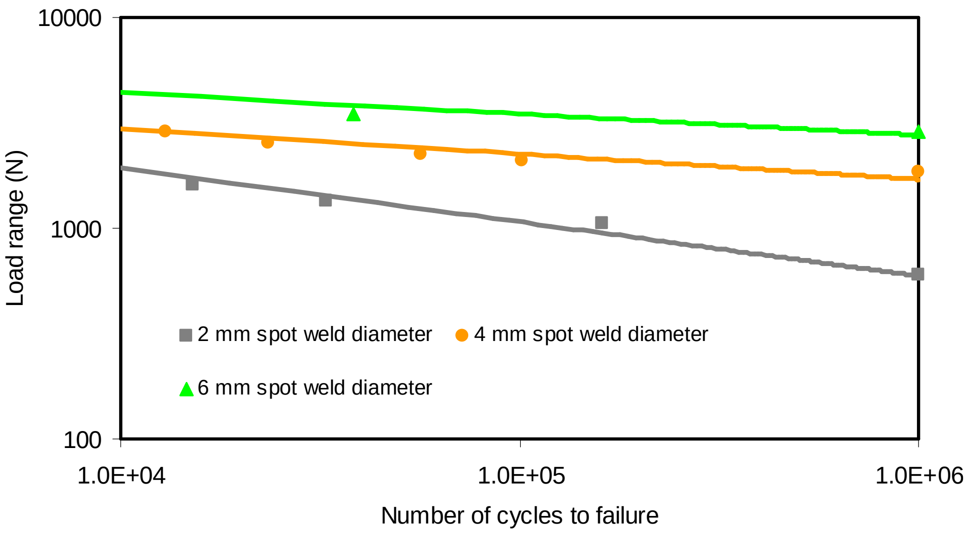

Figure 7. Fatigue lives predicted using “Coffin & Manson’s approach” for TS specimen.

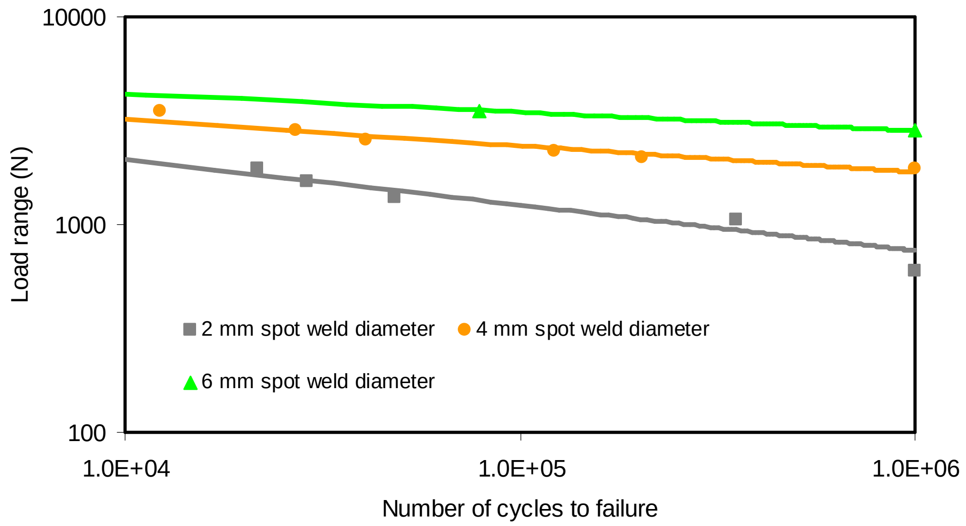

Figure 8. Fatigue lives predicted using “Coffin & Manson’s approach” for MTS specimen.

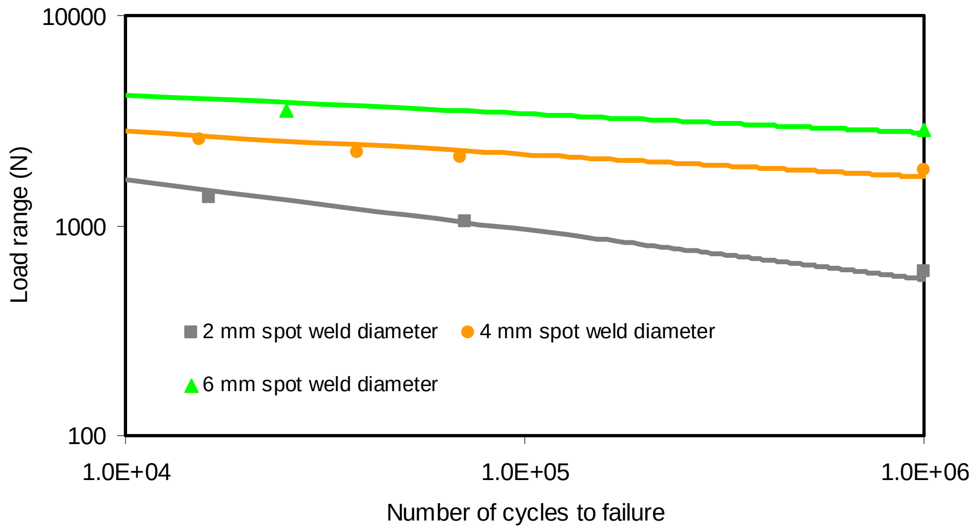

Figure 9. Fatigue lives predicted using “Morrow’s mean stress approach” for TS specimen.

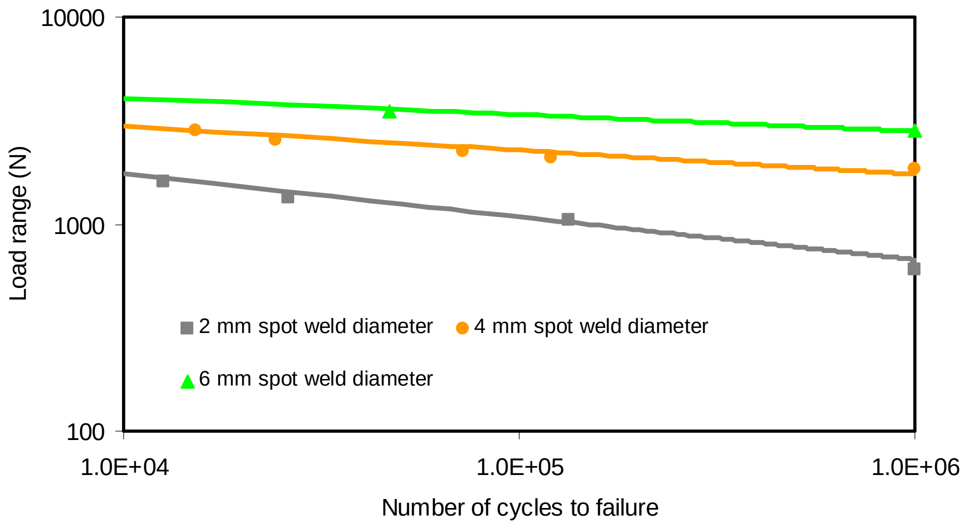

Figure 10. Fatigue lives predicted using “Morrow’s mean stress approach” for MTS specimen.

5. SUMMARY AND CONCLUSIONS

Fatigue life values of four different type of spot weld diameters of MTS and TS were calculated using monotonic test results with empirical relations introduced in literature. Based on the discussions in the preceding sections, the following conclusions can be drawn:

- The fatigue life of the spot welded specimens depended most significantly on the type of specimen, applied load amplitudes and off course the spot weld diameter.
- Equivalent stress approaches have been commonly used because of their simplicity, but their success in correlating multiaxial fatigue data has been limited to a few materials and loading conditions. In addition, they should be used only for proportional loading conditions, in which the principal axes directions remain fixed during the loading cycle.
- Similar to the equivalent stress approaches, equivalent strain approaches are also not suitable for nonproportional multiaxial loading situations, in which the principal alternating strain axes rotate during cycling.
- Failure in a spot-weld occurs in the heat affected zone (HAZ) of unmelted metal between the base metal and the weld metal. The HAZ is caused by the extreme heat of welding changing metallurgical properties on the microscopic level. The HAZ is subjected to higher stresses than the surrounding base metal and is the location of failure in most cases.
- It is known that the failure modes for tensile and modified tensile loading cases depended primarily on the magnitude of the applied mean load and the nugget diameter. It was observed that the amplitude of the applied load appears less of an effect on the final failure mode, although this had a significant effect on the specimen fatigue life.

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