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Fatigue Life Prediction of Notched Components after Plastic Overload Using Theory of Critical Distance

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Abstract

The aim of this work is providing a suitable method based on the Theory of Critical Distance (TCD) to estimate the fatigue life of notched components in the presence of residual stress induced by the overload cycle. Presented method is relied on a virtual stress field to equalize the effects of real stress distribution in the hot spot around the notch root. This method is noted as Equivalent Virtual Stress (EVS) method. The EVS method combined with some versions of the TCD method, such as the Point Method (PM) and Line Method (LM), were applied for two types of notched samples. Samples were made with U- and V-notches with axial stress concentration factors as 2.45 and 5.55, respectively. The test material was AA 2024-T3 due to its considerable resistance to fatigue failures. The PM was found more conservative, while LM showed more accuracy, especially when the critical distance parameter locates far from critical regions. A good agreement was observed between experimental and predicted results.

Nomenclature

E	Young's modulus	$G(R)$	Multiplier function
L_{PM}, L_{LM}, L	Characteristic lengths	N_f	Experimental number of failure cycles
$N_{f,p}$	Predicted number of failure cycles	R	Stress ratio
$R_{y,eff}$	Effective stress ratio in the loading direction	R_{nom}	Nominal stress ratio
r	Distance from notch root	S_y	Yield strength
S_u	Tensile strength	ΔK_{th}	Threshold stress intensity factor
$\Delta\sigma_0$	Endurance limit	$\sigma(r)$	Tangential stress component
σ_m	Total mean stress	σ_{m0}	Applied mean stress
σ_{max}	Maximum applied stress	$\sigma_{y,res}$	Residual stress in the loading direction
$\sigma_{max,eff}$	Maximum effective stress		

1. Introduction

The stress concentrators are typically known as notches in engineering design. Notches are dramatically effective in reducing the fatigue life of components. Due to the complexity of components and structures, it is impossible to avoid notches entirely. On the other hand, cracks can also be considered as sharp notches with higher stress concentration factors. Therefore, studies on the notch effects not only improve the design by

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reducing the fatigue failure effects but also can help to better understand the fatigue crack growth mechanisms. The conventional method of study of notch effects is based on the notch sensitivity and related to correction coefficients. This method is described in detail in the machine design handbook [1]. The estimation methods of the notch sensitivity are explained in the FKM guideline [2]. The notch sensitivity method is based on empirical results, which cannot be considered adequate as an analytical method to investigate the fatigue behavior of notched components, especially when the geometry or stress distribution is more complicated. Therefore, different theories were presented in the last half-century to predict the fatigue life of the notched components to reduce this complexity. Some of these theories are explained and discussed in reference [3], proposing a virtual root radius based on the effective notch stress theory for welded notched thin-plates [4] or modified critical distance method for the fatigue life of bolted joints [5]. By taking a look into the current state of the art of the subject as reference [6], it seems that the fatigue prediction methods based on length parameter have attracted considerable attention in the last decades due to the ease in application and accuracy of estimation. The first approaches were taken by Neuber and Peterson as depicted in [7, 8]. Neuber's method is based on the average stress at a specific length on the notch center line, while Peterson is more concerned about the stress at a particular distance from the notch root. These methods were reformulated in recent years under the title of the Theory of Critical Distance (TCD) [9, 10]. The TCD methods based on the stress at a point or along a line are usually known as point method (PM) and line method (LM), respectively. In both approaches, the critical distance is assumed as a constant parameter which is approximated by material properties [10]. It has recently been shown that TCD can be employed with a proper formulation for the study of low cycle fatigue (LCF) problems [11].

The length parameter can also be considered dependent on the stress distribution around the notch root [12-14]. So the length parameter is not a material constant and depends on the notch geometry and loading conditions.

Despite many advantages of this approach, it seems that the studies about length parameter as a material constant are more practical in an engineering design due to the considerable ease in the application.

This study tries to provide a suitable method to apply the TCD when there are substantial scale of the residual stress fields around the notch root. This type of residual stress field is usually expected when the specimen experiences some plastic deformations due to different loadings such as a plastic overloading. For the evaluation of the accuracy and reliability of the current method, aluminum alloy AA 2024-T3 was used for making the test samples. It is assumed that the damage zone size in the fatigue mechanism depends on the effective stress which includes the residual stress to simplify the solution. It would be shown further that this assumption is found strongly supported by reasonable results of the macro-mechanical investigations.

2. Analytical Method

2.1. The Effect of the Residual Stress Field

The large scale residual stress fields appear after a plastic overload around the notch root. Although the notch affected zone is usually small, it cannot be ignored due to the high compressive stress at the notch tip. For taking into account the effects of the residual stresses, it is assumed that the residual stress in the loading direction ($\sigma_{y,res}$) can be represented as extra mean stress in the fatigue loading in the presence of the applied mean stress (σ_{m0}) [15]. Total mean stress (σ_m), can be expressed mathematically as follows:

$$\sigma_m = \sigma_{y,res} + \sigma_{m0} \tag{1}$$

This definition at some particular points is so advantageous because it brings the capability of engaged fatigue mechanism to hot-spot point independently from near points. Moreover, it can help to find the resultant failure easily and quickly at a hot spot.

The effective stress ratio in the loading direction ($R_{y,eff}$) is introduced as the ratio of difference of minimum stress and mean stress to the difference of the maximum stress and mean stress at each point by Eq. (2).

$$R_{y,eff} = \frac{\sigma_{min} - \sigma_m}{\sigma_{max} - \sigma_m} \tag{2}$$

According to this equation, the maximum effective stress ($\sigma_{max,eff}$) is introduced as the difference between the maximum stress and effective mean stress by Eq. (3).

$$\sigma_{max,eff} = \sigma_{max} - \sigma_m \tag{3}$$

It must be noted that, the total mean stress (σ_m) can be tensile or compressive depending on the interaction of the applied mean and induced residual stresses at the specified locations.

Eqs. (2) and (3) with Eq. (1) provide an effective engineering tool to predict the fatigue life in the presence of the residual stress around the notch root. However, there is an obstacle that prevents this method from being applicable directly in the fatigue calculations of the notched components. Based on the definition for effective components in Eqs. 2 and 3, there is no certain constant values for these parameters in real samples for evaluation of the fatigue life. In fact, each point on the structure has its own effective maximum stress and stress ratio. Nevertheless, we need to obtain the minimum fatigue life of the whole structure by comparing the lives in all points. In this paper, it is tried to introduce a simple method for finding the minimum fatigue life and its location on the notched samples, which can be generalized for real structures. For applying this method directly, values of $R_{y,eff}$ and $\sigma_{max,eff}$ should be determined around the notch root as the location of the crack initiation; however, the stress value does not give reliable answers due to its high gradients [16]. It is almost accepted that the limited areas or volumes around the notch root (known as hot spots) are responsible for the fatigue calculations [17]. As explained before, there are different definitions for hot spots. Some of them depend on the material properties, while others are more concerned with the stress distributions. Both methods were reported to be reliable and accurate in the proper applications.

2.2. Definitions of PM and LM

According to the comprehensive review on the fatigue life modelling methods [6], it seems that the the researchers tended more to deploy the methods based on TCD due to its considerable easy usage with sufficient accuracy that were reported after massive efforts on its qualifying in the last decade [3, 18]. This feature made the TCD a practical engineering tool that removes the dominance of using the size effects. The most known methods based on TCD are PM and LM that can be expressed by Eqs. (4) and (5), respectively [3, 19, 20].

$$\sigma_{eff} = \sigma\left(r = \frac{L_{PM}}{2}\right) \tag{4}$$

$$\sigma_{eff} = \frac{1}{2L_{LM}}\int_{r=0}^{2L_{LM}} \sigma(r)dr \tag{5}$$

The L_{PM} and L_{LM} are length parameters of TCD in PM and LM methods, r is the distance from notch root and is the tangential stress component.

There are different methods to define L_{PM} and L_{LM} based on microstructure studies (like grains size and boundaries) and micromechanics [10].

The general and practical approach used by several investigations in the recent years is that these parameters are the same and can be presented by unique characteristic length as L [10]:

$$L_{LM} = L_{PM} = L \tag{6}$$

L can be computed by employing the linear elastic fracture mechanics (LEFM) for a specific material by applying Eq. (7) with some improvement to increase the prediction accuracy [3, 20].

$$L = \frac{1}{\pi}\left(\frac{\Delta K_{th}}{\Delta\sigma_0}\right)^2 \tag{7}$$

in which, ΔK_{th} is the threshold stress intensity factor and $\Delta\sigma_0$ is the endurance or fatigue limit for the smooth standard test specimen.

For applying TCD on notched components in the presence of the residual stress field, the hot spot loading conditions ($\sigma_{max,eff}, R_{y,eff}$) are equalized with the unified loading condition with the similar number of cycles to failure at a specific stress ratio. For instance, assume applied fatigue load alongside induced residual stress have the effective values as R_1, σ_{max1} at the hot spot location. The required equivalent stress with a specified stress ratio (usually $R = -1$) can be found for any values of the fatigue cycles from stress-life curves.

Therefore, this approach is not concerned with stress value or stress ratio alone by fixing the other parameters similar to the conventional TCD. This equivalent stress can be applied directly in the hot spot with PM and LM methods to study the fatigue life from the macro mechanistic point of view. In fact, the current method removes the obstacles for analyzing the fatigue lives in the presence of the residual stress field by considering the virtual stress distribution that has the similar influence of the actual stress distribution in the hot spot. For this reason, the current method is called Equivalent Virtual Stress (EVS) in the present study. It is also assumed that the hot spot size did not change in the EVS to be able to employ the general formulas of TCD. The reliability of this assumption will be investigated in the next sections by experiments in different notched geometries under various loading conditions.

3. Experimental Tests

Notched specimens made of sheets with a thickness of 3mm were tested under completely reversed loading ($R_{nom} = -1$). The AA 2024-T3 was chosen as the test material due to its widespread applications in the aerospace structures, having low density and remarkable fatigue life. The mechanical properties of the material are obtained from standard tensile test according to ASTM E8/E8M [21] and presented in Table 1. The stress-strain curve of the material and dimensions of the standard specimen are shown in Fig. 1. Furthermore, the chemical composition of the material is given in Table 2.

Table 1
Mechanical properties obtained from tensile tests for AA 2024-T3.

Young's modulus, E	73GPa
Yield strength, Sy	337MPa
Tensile strength, Su	471MPa
Reduction of area	17.5%

The fatigue tests were performed under load controlled condition according to the recommendations of ASTM E 466 [23]. The sinusoidal loading with the frequency between 10 to 30Hz was applied on each specimen using Zwick/Roel machine. This frequency range does not affect the material fatigue behavior according to the suggestions of ASTM E 466.

Table 2
Chemical composition of AA 2024-T3 [%weight].

Element	Al	Si	Fe	Cu	Mn	Mg
Reference [22]	Base	0.07	0.12	4.44	0.51	1.52
Sample	Base	Max 0.5	Max 0.5	3.8-4.9	0.3-0.9	1.2-1.8
Element	Cr	Ni	Zn	Ti	Zr	
Reference	< 0.01	0.01	0.03	0.04	< 0.01	
Sample	Max 0.1	Max 0.05	Max 0.25	Max 0.15	Max 0.05	

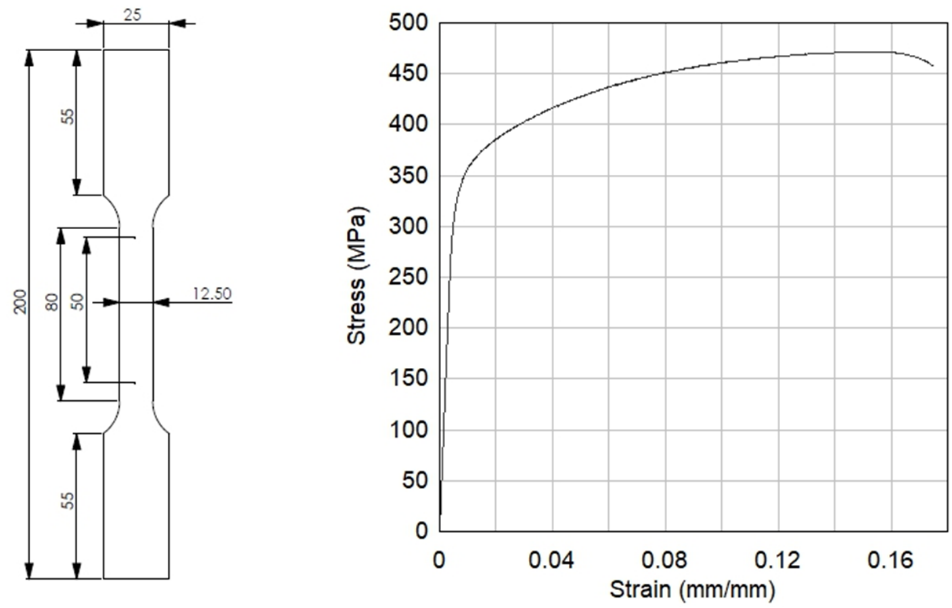


Fig. 1. Standard tensile test specimen and stress-strain curve of AA 2024-T3.

For evaluating the effects of the geometry on the fatigue prediction results, two different notched geometries with the shape of U and V were tested under various loading conditions. The U-notched specimens had a root radius of 5mm and a depth of 4mm. V-notched specimens had a root radius of 0.4mm, depth of 4mm and an opening angle of 60°. The adopted failure criteria were considered as observation of a surface crack with the length about $a_f = 1$mm and $a_f = 0.5$mm for U- and V-notched specimens, respectively. A cracked V-notched sample and machine apparatus for loading are depicted in Fig. 2.

Different loading conditions were applied to each notched specimen series to validate the accuracy of the proposed method. The loading conditions according to the notch geometries are listed in Table 3. The stress concentration factors were computed by the finite element method whose amounts for U- and V-notched specimens were as 2.45 and 5.55, respectively.

4. Failure Prediction of Experimental Tests

This section is concerned with the validation of the proposed method. The stress distribution analysis is a significant step of TCD analysis. It plays an important role in the methods employed in the current study. The stress distribution fields were obtained from numerical computations performed by Abaqus commercial software [24].

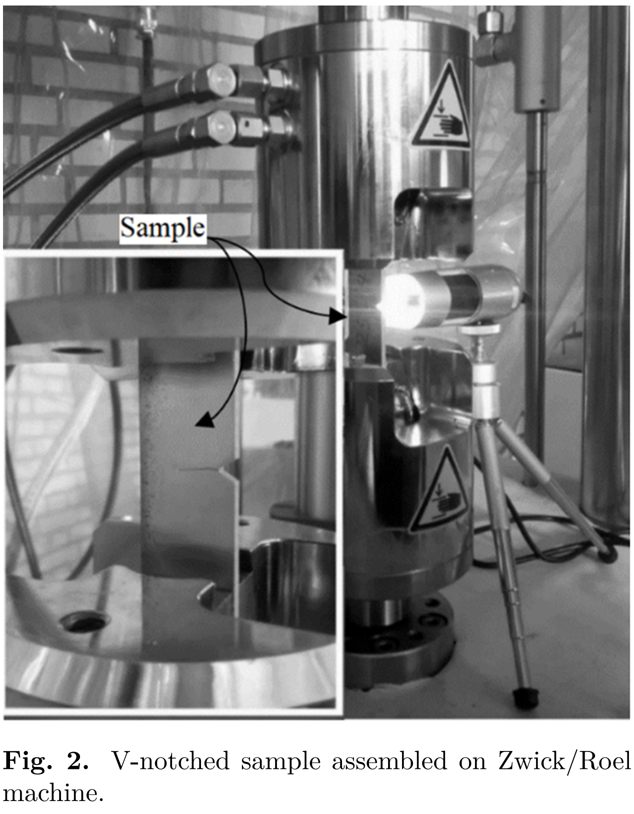


Fig. 2. V-notched sample assembled on Zwick/Roel machine.

Table 3
Details of Loading Conditions

#Code	Overload Stress (MPa)	Fatigue Stress (MPa)
U5-01	150	133.3
U5-02	150	106.7
U5-03	150	80
U5-04	200	160
U5-05	200	133.3
U5-06	200	106.7
U5-07	200	80
U5-08	250	186.7
U5-09	250	160
U5-10	250	133.3
U5-11	250	106.7
U5-12	250	80
V60R0.4-01	100	93.3
V60R0.4-02	100	80
V60R0.4-03	100	66.7
V60R0.4-04	100	53.3
V60R0.4-05	100	40
V60R0.4-06	150	106.7
V60R0.4-07	150	93.3
V60R0.4-08	150	80
V60R0.4-09	150	66.7
V60R0.4-10	150	53.3
V60R0.4-11	200	106.7
V60R0.4-12	200	93.3
V60R0.4-13	200	80
V60R0.4-14	200	66.7
V60R0.4-15	200	53.3

Quadratic elements with reduced integration points were applied to meshing the model. Fine meshing with the minimum size of 15μm was performed around the notch root to increase the computations accuracy. The convergence validation was carried out by mesh refinement process around the notch affected zone.

Using reduced integration points will basically mean that it will take less time to run the analysis but it could have a significant effect on the accuracy of the element for a given problem. Displacement-based finite element formulations usually overestimate the stiffness matrix, so the use of fewer integration points should produce a less stiff elements. Therefore, in some cases, particularly non-linear problems such as plasticity, it is actually advisable to use reduced integration instead of full integration. The slight loss of accuracy is counteracted by the improvement in approximation to real behavior.

In the current study, the normal stress distribution along the notch bisector in the loading direction can be considered as the maximum principal stress according to the loading type and geometry of the samples. Two types of notched samples, one with a 5mm U-notch and other with the V-notch with 0.4mm root radius and 60° angle were used. Each type of the samples was loaded and overloaded with various values as depicted in Table 3. Each sample was modelled with proper elements, for instance, the sample V60R0.4 had a mesh with 7130 nodes and 7058 elements. Biased mesh around the notch root and induced residual stress due to 150MPa far field tension stress are depicted in Fig. 3a. Additionally, variations of the induced maximum compressive residual stress versus element size in the notch root is depicted in Fig. 3b. As can be seen, for elements with the lengths smaller than 100μm, there are ignorable variations in the maximum compressive residual stress distribution.

The computed residual stresses in the hot spots were rectified by applying EVS approach. The EVS was applied by stress-life curves of AA 2024-T3 published for different stress ratios in MMPDS handbook [25] and depicted in Fig. 4.

In this paper, the completely reversed loading condition ($R = -1$) was chosen as the EVS base stress ratio. In the evaluation of the EVS data from fatigue life curves at any stress ratio, an accurate multiplier function was derived, $G(R)$ by curve-fiitting process on the known curves of fatigue lives for AA 2024-T3 from Fig. 4. This function was validated on the experimental data in $-1 < R < 0.52$ for finite life ranges of the used alumium alloy.

$$G(R) = (a + bR^2) \text{ for AA 2024-T3 and } -1 < R < 0.52 \quad (8)$$

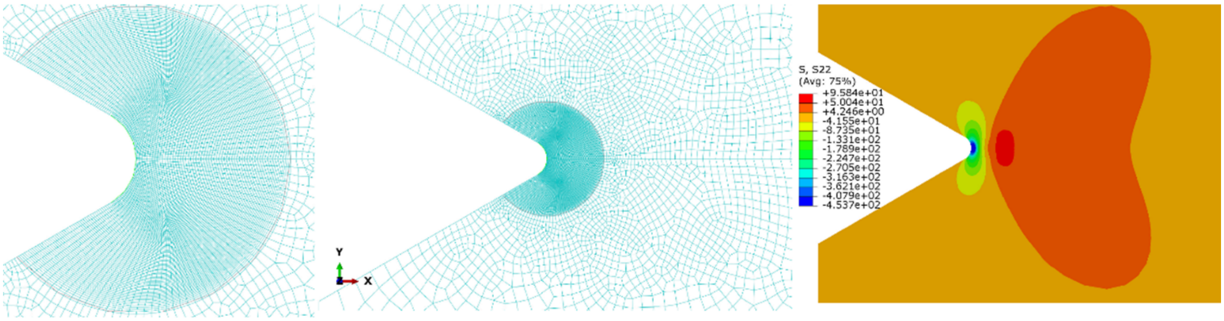
$$a = 0.457, b = -0.530, \text{ and } c = -0.265$$

To find the accurate failure cycles at a any stress ratio, one can directly multiply the value of this function by the failure cycles that is expected at the specified maximum stress in completely reversed loading ($R = -1$) condition. For other metals, similar funtions can be derived based on their fatigue curves for some stress ratios.

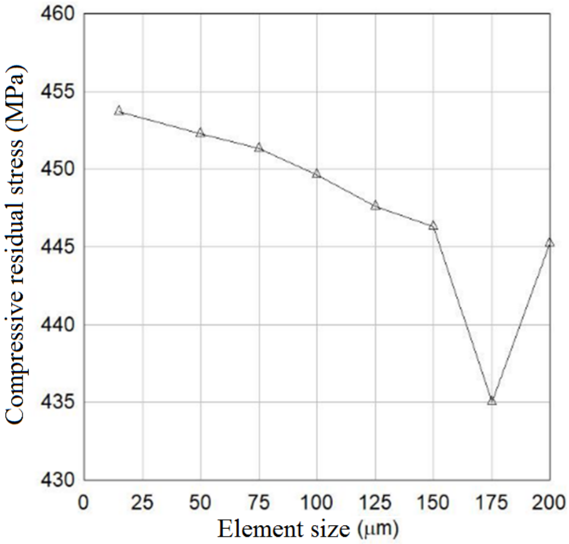
The first step in the TCD is defining the critical distance. $L_{PM} = L_{LM} = 0.146$mm were computed by employing Eq. (6) according to the material properties presented in Table 4.

Table 4
Experimental results of AA 2024 characteristic length [18, 26].

R	ΔK_{th} (MP.mm$^{1/2}$)	$\Delta\sigma_0$ (MPa)	L(mm)
-1	203	300	0.146



(a)



(b)

Fig. 3. a) Biased mesh around the notch root and normal residual stress along loading direction under 150MPa overloading in V60R0.4, b) Effects of the element size on the induced maximum compressive residual stress.

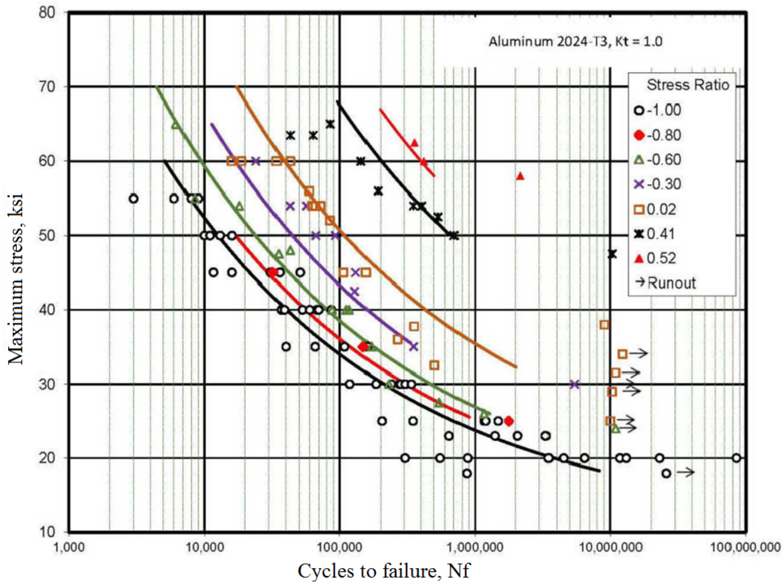
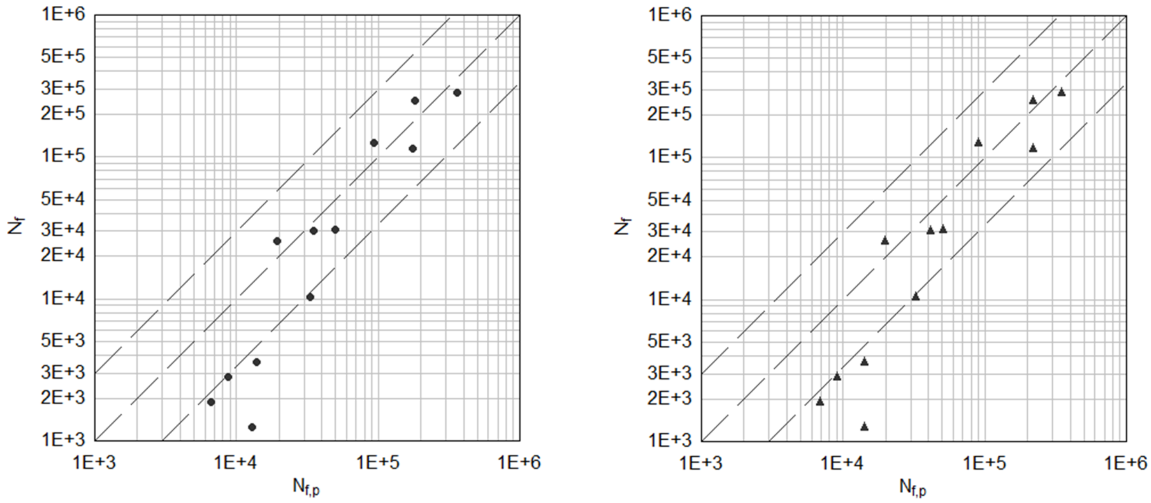


Fig. 4. Fatigue life curves for AA 2024-T3 in different stress ratios [25].

It was initially attempted to predict the fatigue failure of U-notched specimens due to the relatively smaller stress gradient (more stabilized stress curve) that is expected in notched components with small stress concentration factors. The PM and LM with the EVS were applied to stress distribution computed for each notched specimen. The experimental number of cycles to failure (N_f) versus the predicted number of cycles by both PM and LM ($N_{f,p}$) are shown in Fig. 5. One can compare the estimated and measured life cycles' variations with data scattering in obtaining the fatigue curves in Fig. 4. It is evident that the estimations results of PM and LM are within the accepted accuracy. It seems that both PM and LM showed almost similar behavior, although PM is slightly more conservative. The accuracy of both PM and LM increased by increasing fatigue failure cycles.

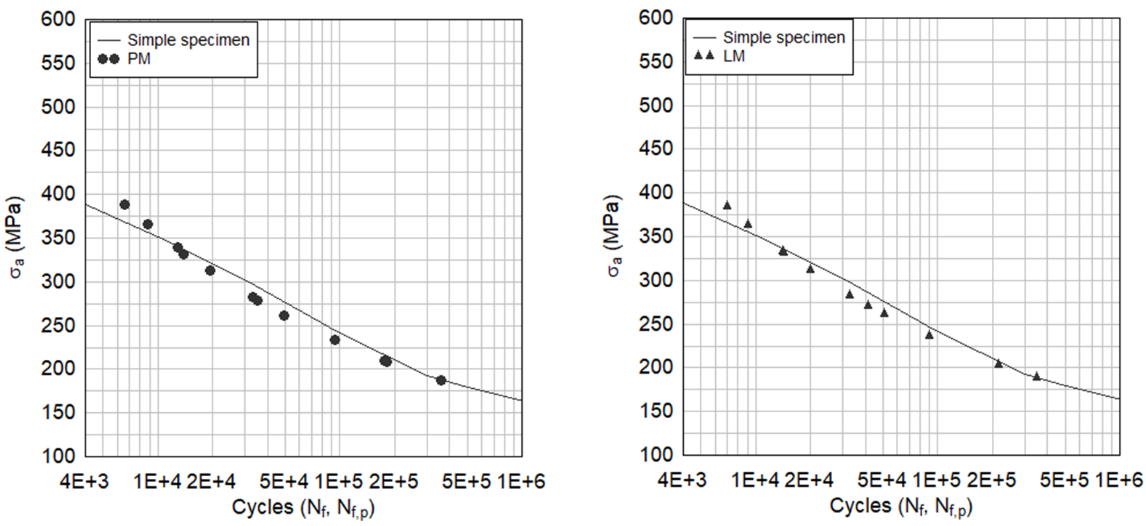
For checking the reliability of the employed stress parameter, EVS, the results are summarized in Fig. 6 based on the effective stress parameter. It can be observed that the effective stress estimation in the hot spot for both PM and LM was considerably accurate. This strongly supports the idea of the EVS about computing effective stress parameter in the hot spot.

Subsequently, the V-notched specimens were investigated. The EVS proposed based on experimental data of MMPDS is not capable of predicting failure life cycles lower than 4000 cycles. This makes some obstacles for analyzing notches with high stress concentration factors due to the high stress values that are expected in regions near to the notch root and may exceed the applied EVS limits.



(a) (b)

Fig. 5. Variations of predicted fatigue lives of U-notched specimens versus experimental data with a) PM b) LM .



(a) (b)

Fig. 6. The accuracy of effective stress parameter of U-notched specimens computed by EVS combined with a) PM and b) LM.

In a few cases of V60R0.4 samples, the stresses values of the very small area close to the notch root exceeded the explained limits. Due to the highly compressive residual stresses at the notch root, the small unpredictable area hardly infiltrated through the notch depth. According to the fact that this area was small in comparison with the notch affected zone, the accuracy is not altered considerably and theses variations can be ignored. On the other hand, this assumption would increase the conservation of the prediction method because effective stress ratio is smaller than zero and larger than −1 due to high compressive residual stress in all cases which faced the explained difficulty. For estimating the fatigue failures occurred in 4000 cycles and smaller, the stress-life curve of AA 2024-T3 at complete reversal loading was employed [27]. This fatigue life curve was chosen due to its high accuracy in low cycle fatigue (LCF) ranges.

The number of failure cycles versus number of estimated failure cycles for V-notched specimens is shown in Fig. 7. The results confirmed that both PM and LM applied with EVS had an acceptable accuracy and the methods are validated in the case of V-notched specimens with high stress concentration factors. As expected, the PM was found more conservative compared to LM, especially in the LCF ranges while LM was found to be more accurate in higher ranges.

For evaluating the accuracy of estimating the effective stress parameter of the V-notched specimens, a similar strategy to the U-notched specimens was taken and the fatigue failure results are re-plotted versus effective stress parameter in Fig. 6. Although the prediction of accuracy was slightly altered in MCF-LCF range (similar to U-notched specimens), both methods were found almost accurate and proved the EVS capability. Fig. 8 shows that the obtained results by PM for LCF regime are not reliable in comparison with MCF and HCF regimes. The reason is the increased sensitivity of failure estimation accuracy to effective stress in LCF and PM due to the reduced slope of the stress-life curve. On the other hand, PM as a method that is based on the point stress close to the notch root seemed not to have the adequate capability when it enters the area with failure cycles in LCF range. Therefore, further investigations are needed to apply PM in the LCF in the mentioned cases.

5. Discussion

According to the obtained results, the proposed method has been validated properly. The fatigue failure of the notched components in the presence of the large scale residual stress fields around the notch root can be successfully predicted by TCD when EVS is applied. Therefore, the critical distance can be considered as a constant for a material which is independent from geometry and life cycles, similar to the general form of TCD. This method can be used to predict the fatigue failure in the presence of the residual stress. The reason for employing constant length parameter has been discussed by some researchers [3, 18, 28]. They introduced the critical distance as a damage zone size estimation (full or fraction) that appears due to the microstructural phenomena. Therefore, it seems that the microstructure phenomenon that is responsible for the fatigue mechanism does not change even in the presence of the residual stress. This was only concluded from the macroscopic studies because the relation between the TCD and microstructure is not yet clear [11], and further investigations in this field are needed.

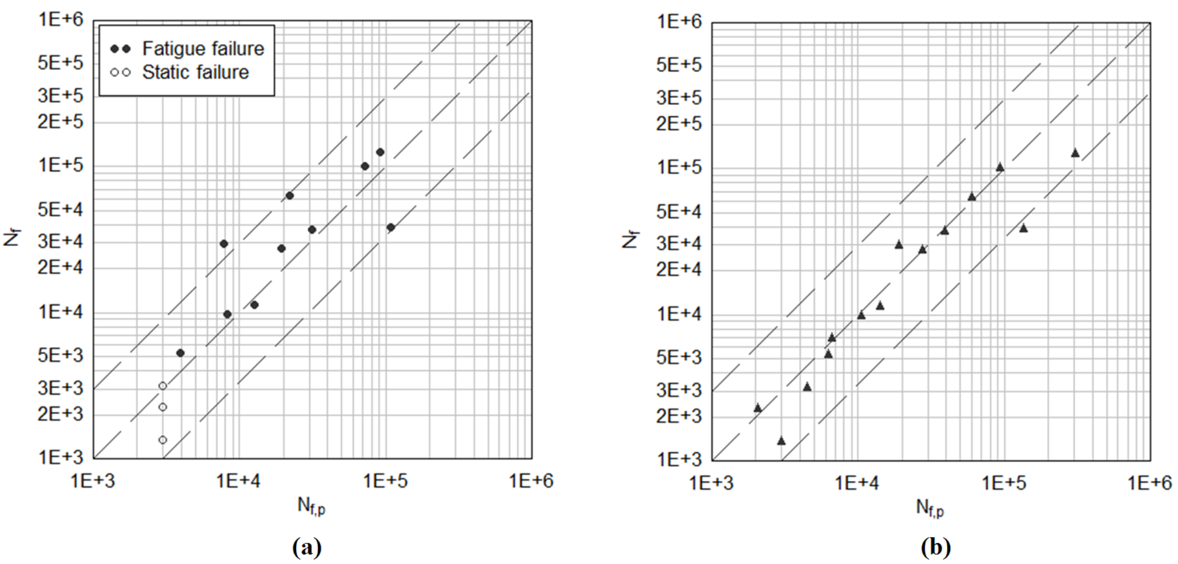
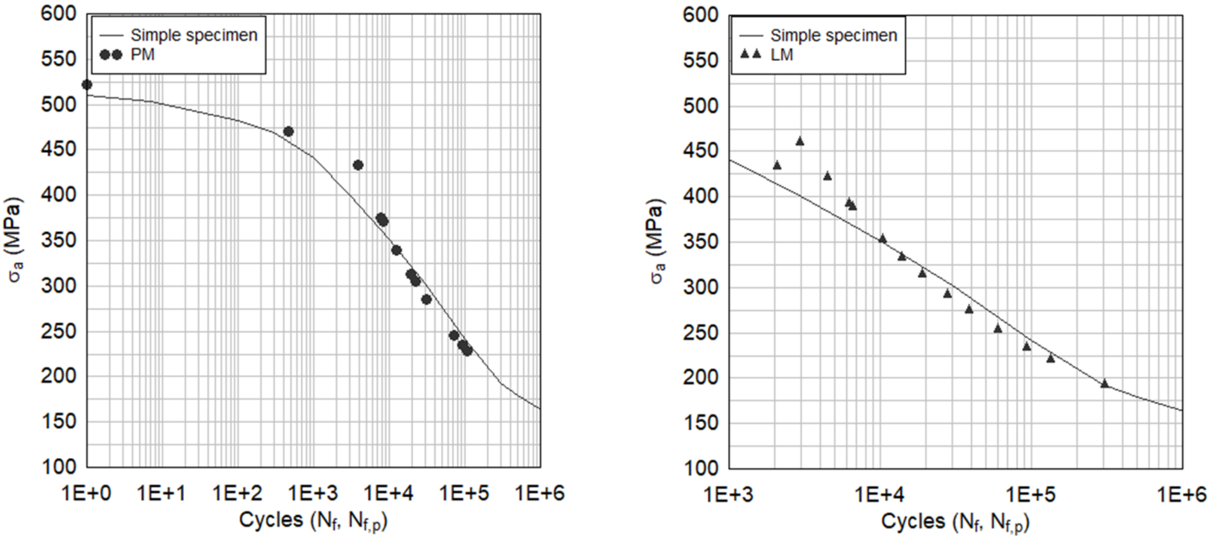


Fig. 7. The accuracy of EVS combined with a) PM b) LM in predicting fatigue failure of V-notched specimens.



(a) (b)

Fig. 8. Effective stress parameter of U-notched specimens computed by EVS combined with a) PM and b) LM.

The virtual stress distribution was applied directly to predict the fatigue failure in the notched components. This method brings considerable ease in the use of TCD to estimate the fatigue life. Therefore, the proposed method can be employed as a powerful engineering tool when residual stress fields are effective on the fatigue life.

Both PM and LM were found reliable for fatigue life calculations in the presence of the residual stress, according to the results of the current study. For the V-notched specimens with high stress concentration factor, LM seems to be more accurate in LCF range while PM was more conservative. However, both methods showed almost similar accuracy for the U-notched specimens with lower stress concentration factor in comparison with V-notched specimens. LM can be more practical in engineering design. Due to the concentration of LM, it results in a narrow band, which seems that finding a more accurate LLM for a chosen material that is larger than the employed one and gives more conservative and accurate results. However, PM results guide to the proper design because conservation is always one of the most critical concerns of the engineers when fatigue is an obstacle to the safe design.

6. Conclusions

This paper tried to present an efficient method for calculating the fatigue life of the notched samples in existence of the residual stress induced by overloading. Effective maximum stress and stress ratio in especial locations determined by theory of critical distance (TCD) beside the fatigue life curves of the simple fatigue specimens were used for life estimations.
Based on the obtained results, the following conclusions can be drawn:

1. A new method was proposed to estimate the fatigue life of the notched components in the presence of the residual stress around the notch root. This method is based on determination of the maximum stress and stress ratio at a known point or line as a hot spot. The method is found successful in both point and line methods of TCD.

2. The critical distance in the presence of the residual stresses was found to have a similar value when the specimen is stress-free. Since the critical distance is usually known as the length parameter that presents the damage zone size in the fatigue mechanism, it was concluded that the damage zone does not change in the presence of the residual stresses.

3. PM was found more conservative, while LM showed more stable behavior in the fatigue prediction. According to the results, it seems that more accurate critical distance can be employed to increase both the accuracy and conservation of the LM which is more efficient.

4. PM and LM showed almost similar accuracy and conservation in case of the U-notched specimens while their behavior was remarkably different in the V-notched specimens with higher stress concentration factors.

5. Both PM and LM showed good accuracy in estimating the effective stress parameter in high cycle fatigue, while their accuracy altered for low cycle ranges.

References

[1] R.G. Budynas, J.K. Nisbett, Shigley's Mechanical Engineering Design, 11$^{\text{th}}$ Edition, McGraw-Hill, USA, (2020).

[2] E. Haibach, Analytical Strength Assessment of Components in Mechanical Engineering. FKM-Guideline, VDMA, (2003).

[3] D. Taylor, The Theory of Critical Distances: A New Perspective in Fracture Mechanics, Elsevier, (2010).

[4] Y. Chang, C. Sun, Y. Qiu, Effective notch stress method for fatigue assessment of sheet alloy material and bi-material welded joints, Thin-Walled Struct., 151 (2020) 106745.

[5] N. Habibi, M. Amoorezayi, A 3D simulation of bolted joint and fatigue life estimation using critical distance technique, J. Stress Anal., 4(1) (2019) 53-63.

[6] M. Kamal, M. Rahman, Advances in fatigue life modeling: A review, Renew. Sustain. Energy Rev., 82 (2018) 940-949.

[7] I. Al Zamzamia, S.M.J. Razavi, F. Berto, L. Susmel, The critical distance method to estimate the fatigue strength of notched additively manufactured titanium alloys, Procedia Struct. Integrity, 28 (2020) 994-1001.

[8] M.D. Chapetti, A.O. Guerrero, Estimation of notch sensitivity and size effect on fatigue resistance, Procedia Eng., 66 (2013) 323-333.

[9] L. Susmel, D. Taylor, A simplified approach to apply the theory of critical distances to notched components under torsional fatigue loading, Int. J. Fatigue, 28(4) (2006) 417-430.

[10] D. Taylor, Geometrical effects in fatigue: a unifying theoretical model, Int. J. Fatigue, 21(5) (1999) 413-420.

[11] L. Susmel, D. Taylor, An elasto-plastic reformulation of the theory of critical distances to estimate lifetime of notched components failing in the low/medium-cycle fatigue regime, J. Eng. Mater. Technol., 132(2) (2010) 021002.

[12] S. Bentachfine, G. Pluvinage, J. Gilgert, Z. Azari, D. Bouami, Notch effect in low cycle fatigue, Int. J. Fatigue, 21(5)(1999) 421-430.

[13] G. Qilafku, N. Kadi, J. Dobranski, Z. Azari, M. Gjonaj, G. Pluvinage, Fatigue of specimens subjected to combined loading, Role of hydrostatic pressure, Int. J. Fatigue, 23(8) (2001) 689-701.

[14] G. Qylafku, Z. Azari, N. Kadi, M. Gjonaj, G. Pluvinage, Application of a new model proposal for fatigue life prediction on notches and key-seats, Int. J. Fatigue, 21(8) (1999) 753-760.

[15] R. Seifi, M.R. Mohammadi, Fatigue life estimation of the overloaded notched components, J. Brazilian Society Mech. Sciences Eng., 42 (2020) 51.

[16] A. Spaggiari, D. Castagnetti, E. Dragoni, S. Bulleri, Fatigue life prediction of notched components: a comparison between the theory of critical distance and the classical stress-gradient approach, Procedia Eng., 10 (2011) 2755-2767.

[17] D. Taylor, G. Wang, The validation of some methods of notch fatigue analysis, Fatigue Fract. Eng. Mater. Struct., 23(5) (2000) 387-394.

[18] L. Susmel, Multiaxial Notch Fatigue, Woodhead Publishing, Elsevier, (2009).

[19] M. Leitner, M. Vormwald, H. Remes, Statistical size effect on multiaxial fatigue strength of notched steel components, Int. J. Fatigue, 104 (2017) 322-333.

[20] M. Braun, A.M. Müller, A.S. Milaković, W. Fricke, S. Ehlers, Requirements for stress gradient-based fatigue assessment of notched structures according to theory of critical distance, Fatigue Fract. Eng. Mater. Struct., 43(7) (2020) 1541-1554.

[21] ASTM Standard E8/E8M-16a. Standard Test Methods for Tension Testing of Metallic Materials, vol 03.01., Annual Book of ASTM Standards, (2016).

[22] Alumium Association. International Alloy Designations and Chemical Composition Limits for Wrought Aluminum and Wrought Aluminum Alloys, Arlington, (2009).

[23] ASTM Standard E466-15: Standard Practice for Conduction Force Controlled Constant Amplitude Axial Fatigue Test of Metallic Materials, vol. 03.01. Annual Book of ASTM Standards, (2015).

[24] Dassult Simulia. Analysis User Manual. Abaqus/Standard, (2021).

[25] Battelle Memorial Institute. Metallic Materials Properties Development and Standardization (MMPDS-11), 11 (2016).

[26] Y. Fujimoto, K. Hamada, E. Shintaku, G. Pirker. Inherent damage zone model for strength evaluation of small fatigue cracks, Eng. Fract. Mech., 68(4) (2001) 455-473.

[27] W. Illg, Fatigue tests on notched and unnotched sheet specimens of 2024-T3 and 7075-T6 aluminum alloys and of SAE 4130 steel with special consideration of the life range from 2 to 10,000 cycles, NACA Technical Notes, (1956).

[28] D. Taylor, S. Kasiri, A comparison of critical distance methods for fracture prediction, Int. J. Mech. Sci., 50(6) (2008) 1075-1081.