Materials Science & Engineering A 697 (2017) 95–100

Contents lists available at ScienceDirect

Materials Science & Engineering A

journal homepage: www.elsevier.com/locate/msea

On estimating axial high cycle fatigue behavior by rotating beam fatigue testing: Application to A356 aluminum alloy castings

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ARTICLE INFO

Keywords:
Basquin Law
Universal slopes method
Four points method
High-cycle fatigue

ABSTRACT

Three methods available in the literature to estimate the axial S-N curve from the rotating bending fatigue performance were summarized. Axial and rotating bending fatigue tests were conducted at various alternating stresses on specimens excised from A356 aluminum alloy castings in the high cycle fatigue range. These data were used to assess the effectiveness of the models. Among the three models, the one developed by Manson and Muralidharan did not provide good estimates of the axial fatigue performance from rotating bending data. The geometric correction factor proposed by Philipp yielded the best fit, however, did not alter the Basquin exponent whereas the one proposed by Halford and Manson provided a poor estimate. Only the model proposed by Esin provided accurate estimates of fatigue performance as well as the Basquin exponent.

1. Introduction

The prediction of fatigue performance has been of interest to engineers so that mechanical components can be design accordingly. Several attempts to estimate fatigue performance have been made [1–5], mostly relying on tensile properties either directly measured or estimated by hardness data. Although these methods provide good initial estimates, it has been recommended [6] that these methods should not be used for final design, and should always be accompanied by fatigue testing.

One of the most commonly used fatigue testing methods has been rotating bending, in which a bending moment is applied to a rotating specimen. As a result, maximum stresses are generated on the surface of the specimen, where fatigue crack is expected to be initiated. In contrast, axial fatigue tests generate uniform stresses on the cross-section of fatigue specimens, and consequently subsurface defects can also initiate a fatigue crack. Because rotating beam tests take significantly less time than the axial tests, there is an abundance of rotating beam fatigue data in the literature.

Although much progress has been made in the last several decades on how microstructural features and structural defects, such as pores and inclusions, affect fatigue performance, even a wider understanding can be gained by interpreting multiple datasets together. Currently, rotating beam fatigue data cannot be used in combination with those obtained in axial fatigue tests for joint analysis. Although there are some models presented in the literature for conversion of rotating beam fatigue data to axial fatigue data,there is no study in the literature in which those models were compared, to the authors' knowledge. The present study is intended to fill this gap by using fatigue data obtained by rotating beam and axial fatigue tests on specimens excised from A356 aluminum alloy castings.

2. Background

When the applied stress is below the yield strength, σ_y, of the material, the relationship between the applied stress amplitude, σ_a, and resultant fatigue life follows the Basquin Law [7]:

$$\sigma_a = \sigma_f' N_f^b \quad (1)$$

where σ_f' is the fatigue strength coefficient (MPa) and b is the Basquin exponent. Both Basquin parameters are strongly affected by the material [8], specimen geometry [9] as well as the type of test conducted [10]. Most models in the literature are built on the strain-based fatigue life. Eq. (1) can be rewritten in terms of strain by dividing both sides of the equation with modulus of elasticity, E:

$$\varepsilon_{ae} = \frac{\sigma_a}{E} = \frac{\sigma_f'}{E} N_f^{\,b} \quad (2)$$

where ε_{ae} is the elastic strain amplitude. The plastic strain amplitude, ε_{ap}, can be found by the Coffin-Manson relationship [11,12]:

$$\varepsilon_{ap} = \varepsilon_f' N_f^{\,c} \quad (3)$$

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http://dx.doi.org/10.1016/j.msea.2017.05.008
Received 3 March 2017; Received in revised form 14 March 2017; Accepted 2 May 2017
Available online 03 May 2017
0921-5093/

where ε_f' is fatigue ductility coefficient and c is plastic exponent. Eqs. (2) and (3) can be combined to find the total strain amplitude, ε_a:

$$\varepsilon_a = \varepsilon_{ae} + \varepsilon_{ap} = \frac{\sigma_f'}{E} N_f^{\,b} + \varepsilon_f' N_f^c \tag{4}$$

The stresses developed in axial and rotating beam fatigue tests are quite different; in axial testing, stresses are mostly homogeneous across the cross-section of the specimen. In rotating beam fatigue tests, however, stress is zero on the center of the specimen and changes linearly from center to surface, where it reaches the maximum stress, σ_{max}. To find the equivalent stress amplitude in an axial test, a correction factor, φ, between rotating beam maximum stress, $\sigma_{max\,|\,rb}$, and axial stress amplitude, $\sigma_{a\,|\,ax}$ can be introduced, such that;

$$\sigma_{a\,|\,ax} = \varphi \sigma_{max\,|\,rb} \tag{5}$$

One of the first attempts to estimate φ was made by Philipp [13] who used geometric correction:

$$\varphi = \frac{3\pi}{16} \tag{6}$$

which is approximately 0.59. The analysis of data from literature on various steels by Sors [14] showed that the endurance limits obtained from the axial fatigue tests were approximately 0.57 times those obtained by rotating beam tests, which agree with Philipp's geometric model. However, these results is not consistent with those reported by France [15] who conducted axial and rotating beam fatigue tests on ten types of steel and four types of iron. The ratio of endurance limits obtained in axial to rotating beam fatigue tests ranged between 0.74 and 1.0. Therefore there is no agreement ratio of fatigue lives obtained by the two fatigue tests near endurance limit. Halford and Manson [16] followed a similar approach and proposed a constant based on a "material homogeneity factor", which yielded $0.71 \le \varphi \le 0.85$.

Manson and Muralidharan [17] developed a methodology by taking the plastic flow into account and based on some material constants, suggested closed form equations to identify the true bending stresses in terms of axial stresses. For Eq. (4), fatigue life at the intersection point of the two lines, N_T (also known as transition life) as indicated in Fig. 1, is given by:

$$N_T = \left(\frac{E\varepsilon_f'}{\sigma_f'}\right)^{1/(b-c)} \tag{7}$$

Instead of using the known bending moment to calculate the strain, Manson and Muralidharan used strain to calculate the required bending moment. Consequently, they provided a correction factor between rotating beam maximum stress, $\sigma_{max\,|\,rb}$, and axial stress amplitude, $\sigma_{a\,|\,ax}$:

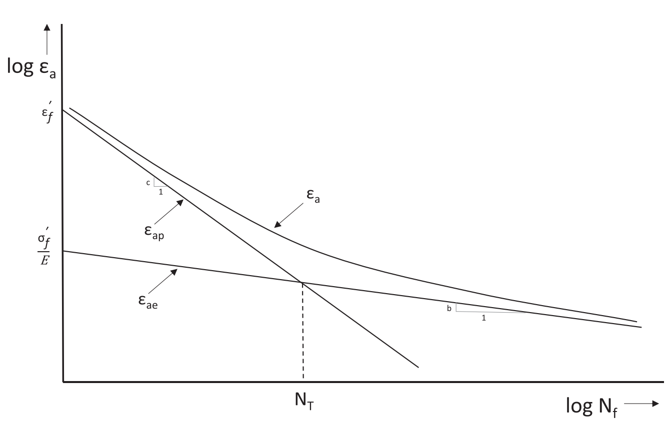

Fig. 1. The two components of the total strain based fatigue life curve shown schematically (adapted from [1]).

$$\varphi = \frac{1 + 2\left(\frac{N_f}{N_T}\right)^{(c-b)} + \left(\frac{N_f}{N_T}\right)^{2(c-b)}}{1 + f_1\left(\frac{N_f}{N_T}\right)^{(c-b)} + f_2\left(\frac{N_f}{N_T}\right)^{2(c-b)}} \tag{8}$$

where,

$$f_1 = \frac{1 + 0.843\frac{c}{b}}{0.685 + 0.255\frac{c}{b}} \tag{9}$$

$$f_2 = \frac{2.619\frac{c}{b}}{1.0 + 1.541\frac{c}{b}} \tag{10}$$

Note that f_1 and f_2 depend on the geometry of the cross-section and are given here only for circular cross-sections. To estimate the four parameters of Eq. (4), they used the "universal slopes" method introduced by Manson [1] in which c and b are taken as −0.6 and −0.12 for all alloys, respectively. Their results showed a good agreement with the calculations made in the low cycle region. For high-cycle fatigue where volumetric effect plays an important role, Manson and Muralidharan noticed some discrepancy between bending and axial type fatigue stresses. It should be noted that Manson [1] also introduced the "four point" method to estimate the parameters of Eq. (4). The equations for the "universal slopes" and "four points" methods are outlined in Table 1, where R_A is reduction in area.

Esin [18] studied micro-plasticity taking place due to stress distributions across the cross-section of specimens. By taking the volumetric effect into account, and using the macro-micro element concept, Esin provided a model that could successfully correlate axial and rotating beam fatigue testing results in the high-cyclic region in steels. According to Esin, the correction factor is found by:

$$\varphi = \frac{2}{3}\left(\frac{1 - k^3}{1 - k^2}\right) \tag{11}$$

where,

$$k = \frac{d}{D} \tag{12}$$

D is the specimen diameter, and d is the diameter of the section of the specimen inside which stresses are lower than the endurance limit. Esin also provided a correction factor for fatigue life:

$$\lambda = \frac{D^2}{(D^2 - d^2)} \tag{13}$$

$$N_{f\,|\,rb} = \lambda . N_{f\,|\,ax} \tag{14}$$

Even though the true elastic limit of steels was found to be below endurance limit [19], Esin suggested simply using the endurance limit as the true elastic limit for a practical approach. It should be noted that aluminum alloys do not show a distinct endurance limit. Because Esin's model requires an endurance limit, the alternating stress level corresponding to a fatigue life of 10^8 was taken as the endurance limit in the present study, which is consistent with the results of Yi et al. [20] for 319 aluminum alloy castings.

Table 1
The two estimation methods for fatigue properties by Manson [1].

	Universal Slopes	Four Points
ε_f'	$0.758[-\ln(1 - R_A)]^{0.6}$	$\frac{0.125}{20^c}[-\ln(1 - R_A)]^{0.75}$
c	−0.6	$\frac{1}{3}\log\frac{0.0066 - \sigma_f'(2\times10^4)^b / E}{0.239[-\ln(1 - R_A)]^{0.75}}$
σ_f'	$1.902S_T$	$1.25S_T(1 + \ln(1 + R_A))2^b$
b	−0.12	$\log\left(\frac{0.36}{(1 + \ln(1 + R_A))}\right)/5.6$