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STRAIN-LIFE FATIGUE ANALYSIS

Report No. R-589

J. C. Shang

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13. ABSTRACT

The strain-life approach to characterize the fatigue behavior of materials and its applications to railroad equipment design have been investigated. A logical means of accounting for local plastic strains, which are the cause of fatigue failures, has been given, and constitutive relations between strain and fatigue life were also developed. Moveover,

- o A concept of cyclic material characteristics was introduced;
- o Mean stresses were taken into account by a simple modification of the strain-life equation;
- o A technique to include the effect of geometric notches was presented.

A combination of the above analytical tools with an adequate cycle counting technique, such as rainflow, which accrues closed hysteresis loops permits prediction of fatigue initiation life of materials under random road environments.

In addition, a general overview of current fatigue analysis methodologies using the stress-life and strain-life approaches and the fracture mechanics formulation, is presented in relation to their design philosophy, benefits and pitfalls when applied to the railroad industry.

14. SUBJECT TERMS

Railroad Cars; Fatigue Analysis; Design Methodology

15. AVAILABILITY STATEMENT

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EXECUTIVE SUMMARY

Trends in the design of mechanical or structural components in transportation equipment to prevent fatigue failures prior to their expected life have led to more sophisticated procedures, both in analysis/design and in testing. The major effort of this program focused on the appraisal of strain-life fatigue analysis methodology and its application feasibility to railroad equipment design. Moreover, research efforts to date related to fatigue design technology, including fracture mechanics, and stress-based or strain-based methods are also evaluated.

The major tasks accomplished are:

1) Appraisal of research efforts to date, and evaluation of application experience,
2) Assessment of application feasibility to railroad car design,
3) AAR specification enhancement and program implementation.

The general observations made were the following:

1) The fracture mechanics approach utilizes the crack propagation model to characterize crack growth from inherent material and manufacturing defects such as those related to welds. Unfortunately, one of the stumbling blocks to such an approach is the introduction of a concept using flawed components which have no counterpart in conventional railroad design practice; the primary interest in railroad equipment design is the fatigue life associated with the crack initiation period and the crack propagation phase becomes of secondary concern. Another problem related to the

use of this approach is that a periodic inspection program must be executed in order to monitor the growth of cracks before they reach the critical stage that would lead to a total failure. The railroad industry does not exercise such an inspection program.

2) Materials fail due to over-straining, not by over-stressing. The material characteristics (typically, stress-strain data) also exhibit a tendency for the stress not to follow the strain once plastic strains, which are the real cause of fatigue damage, are experienced. Thus, the use of stress-based techniques to predict fatigue life could be somewhat over-simplified, since the resulting stress spectra generated under a ramdom road environment will not be identical to that of the loading. This is a major drawback in terms of estimating the cumulative damage incurred by fatigue.

3) The strain-based life method allows for the presence of plastic strains through cyclic material characterization. With an appropriate cycle counting procedure, such as the rainflow counting process, the strain-based method has become a vital tool for fatigue life prediction and has been in extensive use in the automobile industry. Adoption of this technique to railroad equipment design poses no foreseeable difficulty. This has been demonstrated by a working example. A FORTRAN program to solve the complex strain-life relation has also been prepared for implementation in order to further enhance the AAR standards and specifications.

Recommendations for future enhancements in strain-life fatigue analysis are:

1) Procedures for dealing with simultaneous application of multiaxial motion,
2) Methods of analysis to assess fatigue damage due to the presence of multiaxial stress state variation resulting from random road environment input.

ACKNOWLEDGEMENTS

This report documents engineering activities involved in an investigation of application feasibility of the strain-life fatigue analysis techniques to the design of freight cars subjected to random road environment. The work described in this report was conducted under the cooperative sponsorship of the Federal Railroad Administration and the Association of American Railroads through the Track Train Dynamics (TTD) Contract No. TTD-83-104.

The author wishes to express his appreciation to all of the members in the TTD Fatigue Task Force, in particular to its chairman, Mr. Shaun Richmond, and the director of the TTD program at the AAR, Mr. Roy A. Allen, for their valuable contribution to the direction of the project.

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Section 1

INTRODUCTION

1.1 Background

It has been understood that in general, the demonstration of basic strength of appurtenances or structural components under static and/or impact loading alone is insufficient for long-life service, when cars are constantly subjected to fluctuating environments on the roads. Moreover, the assurance of adequate fatigue performance in structural components of railroad equipment is still one of the most difficult problems facing engineers. Uncertainty involved in the analysis of complex service environments, complicated structural geometry and attachment details, coupled with a lack of usable material data and inadequate design methodology, are prime contributors to this problem.

Until recently, fatigue design and service life estimates of mechanical and structural components have generally been made on the basis of the nominal (far-field) stress concept in conjunction with uniaxial test data and Goodman's diagrams. While this approach has enjoyed wide acceptance as a useful tool to deal with fatigue problems, some of the fatigue failures experienced have been quite contrary to expectation.

For instance, the life of automobile components has been shortened because of occasional impacts (bumps) which create local plastic strains. The results of recent tests conducted at the U.S. Steel Laboratory also indicate that the presence of an undercut (a cut generated at the toe of a weld during cooling) could reduce the fatigue life of welded equipment as much as 80 from that of specimens without undercuts. These differences between the actual fatigue life observed and those predicted by means of the nominal stress method are attributed to the fact that, while load and strain histories are observed to be similar in character, the stress history displays a considerably different waveform; stresses are obviously not always proportional to strains. Moreover, they may be of opposite sign for particular instances, as a result of sudden startup, occasional impacts, or hysteresis behavior of the material.

The primary reason for the continuous use of the nominal stress method stems from a lack of analytical tools required to determine the strain/stress conditions locally in the critical areas which actually control the fatigue/fracture behavior. However, with the advent of the finite element method (FEM), the differences encountered in determining the local strain/stress response have been substantially reduced, if not eliminated. Since then, the strain fatigue life analysis has gained favor over the nominal stress method, not only by the aircraft industry (dealing with high stress-low cycle fatigue) and process equipment manufacturers (concerned with low stress-high cycle fatigue), but also by automobile manufacturers (involving both aspects).

1.2 Program Definition

The program effort is mainly in supporting the Track Train Dynamics (TTD) program in the transformation of basic information related to the strain life fatigue analysis techniques into a simple, yet comprehensive, format that can be easily understood and used by engineers/designers of railroad cars. This format can then be incorporated into the AAR fatigue design specifications.

1.3 Program Objective and Scope of Work

The objective of the program is to review the strain life analysis techniques, to evaluate the feasibility of using such techniques in the fatigue design of railroad cars, and to identify its potential and its limitations for enhancement of the AAR fatigue guidelines.

The scope of work involves the following:

1) General description of the background, theory, assumptions, limitations, and procedures related to the subject method, as it applies to railroad car structural analysis,
2) Identification of data required to perform fatigue analysis using the strain-life approach, such as loadings, environment, material characteristics, and analysis tools,
3) Enhancement of the AAR fatigue guidelines with the strain-life analysis approach,
4) Documentation and reporting.

1.4 Program Tasks and Accomplishments

Four major tasks were defined as the following:

1) Appraisal of research efforts to date, and evaluation of application experience,
2) Assessment of application feasibility to railroad car design,
3) Specification enhancement and program implementation,
4) Documentation and reporting.

The program plan executed to accomplish these tasks is shown in Figure 1.1, which was developed in relation to the program objective and the scope of work established.

Program accomplishments include the following:

1) <u>Appraisal of Research Efforts to Date and Evaluation of Application Experience</u>

   Leading experts from the academic community, industrial research organizations and government agencies were invited to offer their views regarding the theoretical background and practical application experience of the strain-life fatigue analysis approach in general (Appendix A). The majority of responses received supported the use of the strain-life analysis technique. Others rejected the idea because of its apparent theoretical simplicity as compared with continuum mechanics considerations, and suggested different approaches based on the fracture mechanics formulation. An overview of comparison of these current fatigue analysis/design methods is presented and discussed in Section 2. A description of the fundamentals of the strain-life analysis method is documented and reviewed in Section 3, including discussions of the related assumptions, limitations, potential benefits and pitfalls, as applied to the fatigue life analysis and design of railroad cars in Appendix C. Furthermore, the analysis procedures for using such methods in a step-by-step fashion, in association with the basic data requirements, are also demonstrated in Section 4.

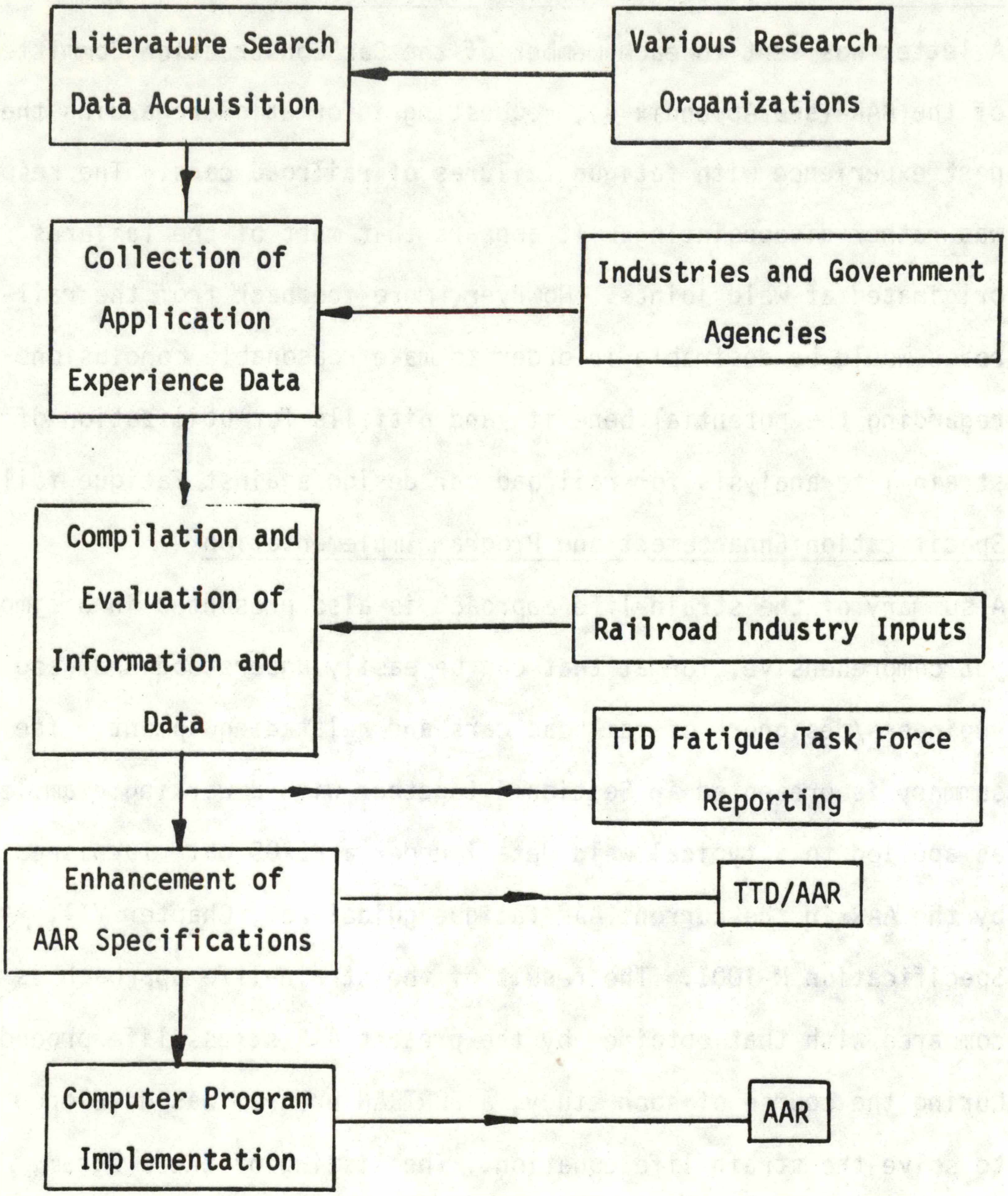


Figure 1.1 PROGRAM PLAN

2) <u>Assessment of Application Feasibility to Railroad Industry</u>

A letter was sent to each member of the Car Construction Committee of the AAR (see Appendix B), requesting information regarding their past experience with fatigue failures of railroad cars. The response was rather disappointing. It appears that most of the failures originated at weld joints. However, more feedback from the railroads would be desirable in order to make reasonable conclusions regarding the potential benefits and pitfalls for utilization of strain-life analysis for railroad car design against fatigue failure.

3) <u>Specification Enhancement and Program Implementation</u>

A summary of the strain-life approach is also presented in a simple, yet comprehensive, format that can be easily understood and used by engineers/designers of railroad cars and related equipment. The summary is presented in Section 4 together with a working example, as applied to a typical weld detail under a REPOS data furnished by the AAR in the current AAR fatigue guidelines, Chapter VII, AAR Specification M-1001. The result of the strain-life approach is compared with that obtained by the present AAR stress-life procedure. During the course of such study, a FORTRAN program was developed to solve the strain-life equation. The listing of the program, related input requirements and output format are presented in Appendix D.

4) Documentation and Reporting

In addition to the background and theoretical basis for strain-life analysis application, a sensitivity study of influential factors that affect the fatigue life prediction by both stress-life and strain-life approach was conducted. The results of the study are presented as part of the technical discussions in Section 5. Comments with regard to the current AAR fatigue guidelines, concerning cars under multiaxial random motion or multiaxial stress fatigue in particular, are given. The areas requiring future research are also identified.

Section 2

OVERVIEW OF FATIGUE DESIGN METHODOLOGY

(Continuum Mechanics Vs Fracture Mechanics)

The object of this overview is not intended to review the "state-of-the-art" or to evaluate the merit of modern methods employed in the treatment of discontinuity problems in relation to fatigue failures and their prevention, but rather to summarize the viewpoint, methodology, and scope of these analysis methods, in conjunction with their inherent niceties and uncertainties of the assumptions and approximations. Through the remarks which follow, it is hoped to present a coherent picture of what each approach sets out to accomplish and how each achieves those goals.

In addition, an evaluation of strain-life analysis methods in relation to practical applications is presented in terms of theoretical considerations, criticisms, and application justification and limitations.

2.1 Background

The assurance of adequate fatigue performance in structural components of railroad equipment is still one of the most difficult problems facing engineers. Uncertainty involved in the analysis of complex service environments, complicated structural geometry and attachment details, coupled with a lack of usable material data and inadequate design methodology, are prime contributors to this problem.

Failure of transportation vehicle structures by fatigue has received the attention of engineers for over a century. Because of the often serious consequences associated with structural failures (not only damage to equipment and to property but also the loss of life), an immense amount of research effort has been undertaken. This has contributed to present fatigue technology and the understanding of the failure processes, such as the stress control failure mode in conjunction with the nominal (far field) stress application, and to the development of design tools (such as the "safe" stress level design approach in association with S-N curves/endurance limits, Goodman's diagrams, Miner's cumulative damage rule, etc.) for dealing with them in engineering design.

Until recently, fatigue design and service life estimates of mechanical and structural components have generally been made on the basis of the nominal (far-field) stress concept in conjunction with uniaxial test data and Goodman's diagrams. While this approach has enjoyed wide acceptance as a useful tool to deal with fatigue problems, some of the fatigue failures experienced have been quite contrary to expectation.

For instance, the life of automobile components has been shortened because of occasional impacts (bumps) which create local plastic strains. The results of recent tests conducted at the U.S. Steel Laboratory also indicate that the presence of an undercut (a cut generated at the toe of a weld during cooling) could reduce the fatigue life of welds as much as 80% from that of specimens without undercuts. These differences between the actual fatigue life observed and those predicted by means of the nominal stress method are attributed to the fact that, while load and strain histories are observed to be similar in character, the stress history displays a considerably different waveform; for example, stresses are obviously not always proportional to strains. Moreover, they may be of opposite sign for particular instances, as a result of sudden startup, occasional impacts, or hysteresis behavior of the material.

The primary reason for the continuous use of the nominal stress method stems from a lack of analytical tools required to determine the strain/stress conditions locally in the critical areas containing discontinuities such as notches, weld defects, etc., which actually control the fatigue/fracture behavior. However, with the advent of the finite element method (FEM), the difficulties encountered in determining the local strain/stress response have been substantially reduced, if not eliminated. Since then, strain fatigue life analysis has gained favor over the nominal stress method, not only by the aircraft industry (dealing with high stress-low cycle fatigue) and process equipment manufacturers (concerned with low stress-high cycle fatigue), but also by automobile manufacturers (involving both aspects).

In addition, the utilization of crack propagation models to characterize fatigue crack growth from inherent discontinuities has also been presented to engineers as a fatigue design tool which is an alternative to the use of stress or strain-life methods. Basically, they distinctly differ by the approaches taken in the treatment of discontinuities; continuum mechanics for the stress/strain methods versus fracture mechanics for the crack propagation models. Details are discussed in the following subsections.

2.2 Definition of Discontinuities

"Discontinuities" are perturbations in the material that disrupt the distribution of load transmission through the body. They cause a localized intensification of strain (stress) and thus reduce the material resistance to failure.

2.3 Discontinuity Analyses - Continuum Mechanics vs. Fracture Mechanics

2.3.1 Continuum Mechanics Model - Treatment of Finite-Size Discontinuities

In the theory of continuum mechanics, the material is assumed to be continuous, even though the domain contains discontinuities, such as notches, inclusions, defects, etc., inherent to the material as the result of manufacturing or/and fabrication processes.

This can be easily illustrated by a stress analysis of a plate with an elliptical hole, shown in Figure 2.1. The elliptical hole is treated as a discontinuity with a finite width, c, and a finite length, a, such that the maximum stress found at the tip of the hole (at the point A) is given by:

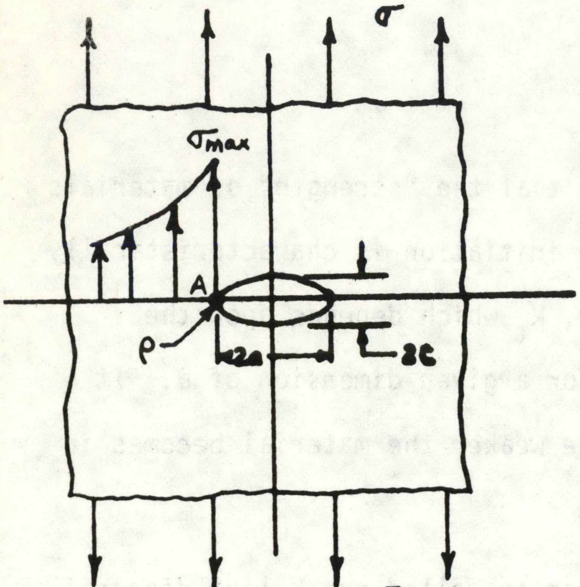
$$\sigma_{max} = \sigma\left(1 + 2\sqrt{\frac{a}{\rho}}\right) \qquad (2.1)$$

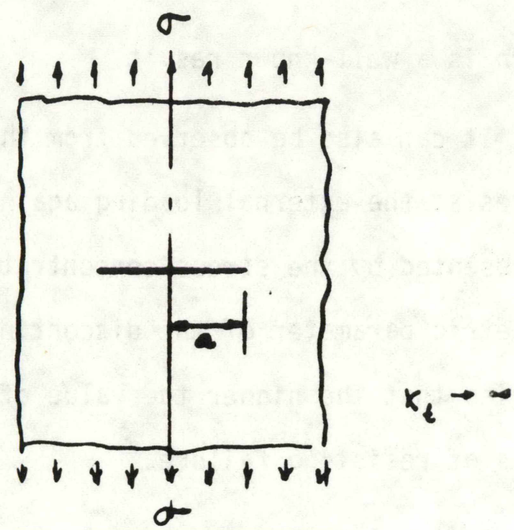
where

$$\rho \text{ (tip radius)} = \frac{c^2}{a} \qquad (2.2)$$

Furthermore, the stress concentration factor is defined such that

$$K_t = \frac{\sigma_{max}}{\sigma} = \left(1 + 2\sqrt{\frac{a}{\rho}}\right) \qquad (2.3)$$





Continuum Mechanics Model

$$K_t = \frac{\sigma_{max}}{\sigma} = \sigma \left(1 + 2\sqrt{\frac{a}{\rho}}\right)$$

For Design:

$$\sigma_{max} = K_t \sigma \le \sigma_{design} = \frac{\sigma_{yield}}{S.F.}$$

Fracture Mechanics Model

$$K = \sigma\sqrt{\pi a}$$

For design:

$$\sigma\sqrt{a} \le \frac{K}{\sqrt{\pi}}$$

Figure 2.1 TREATMENTS OF DISCONTINUITY PROBLEMS

For a circular hole, when $c = a$, $\rho = a$,

$$K_t = 3$$

which is a well-known result.

It can also be observed from this example that the "strength" of materials to resist the external loading against failure initiation is characteristically represented by the stress concentration factor, K_t which depends upon the geometric parameter of the discontinuity, ρ, for a given dimension of a. It implies that the higher the value of K_t is, the weaker the material becomes in terms of resisting failure.

Obviously, for an extremely narrow hole (or so called crack-like discontinuity), the value of ρ (or c) approaches zero. Consequently,

$$\lim_{\rho \to 0} K_t \to \text{infinity} \tag{2.4}$$

$$\lim_{\rho \to 0} \sigma_{max} \to \text{infinity} \tag{2.5}$$

These results definitely indicate that the elastic continuum mechanics model fails to yield a realistic assessment of the stress distribution around crack-like discontinuities.

2.3.2 Fracture Mechanics Model - Treatment of Crack-Like Discontinuities

The concern here is not so much about the material "strength" to resist the initiation of a crack which already exists, but with the material properties that can characterize the material resistance to further growth of the crack.

A convenient and reasonably rigorous definition of fracture mechanics is "the applied mechanics of crack growth". It supposes the pre-existance of a significant crack-like defect that will advance under loading, leading towards complete fracture. It has also helped to quantify the rather elusive concept of "toughness" which can now be usefully defined as "resistance to crack growth". For brittle fracture, it is "fracture toughness". On the other hand, for crack propagation (crack growth at slower rate), it is "stress intensity factor", which is defined as:

$$K = \lim_{\rho \to 0} \left\{ \frac{1}{2} \sigma_{max} \sqrt{\pi \rho} \right\} \tag{2.6}$$

For a fine elliptical crack, as $\rho \to 0$,

$$K = \lim_{\rho \to 0} \left\{ \frac{1}{2} \sigma \left(1 + 2 \sqrt{\frac{a}{\rho}}\right) \sqrt{\pi \rho} \right\}$$

or

$$K = \sigma \sqrt{\pi a} \tag{2.7}$$

If a small crack-like defect is acceptable, then knowledge of its growth rate (either by single loading or fatigue) will allow the <u>logical definition of a re-inspection time</u> adequate to detect the larger crack before it reaches its critical crack length, a_{cr}, after which a complete fracture occurs. The value of K at the onset of rapid fracture is referred to as the critical stress intensity factor, K_c, the material toughness against crack propagation.

2.4 Design Applications

Fracture mechanics has emerged as a new design concept, in addition to the stress/strength based concept, because all engineering materials, to a certain degree, have inherent imperfections, particularly those associated with manufacturing and fabrication, such as castings and weldments. Efforts are continuously made by fracture mechanics experts to incorporate this approach into standards and codes of design practice. In the example of (elliptical) cracks, if the material toughness given by K_c (experimentally determined) is available, then, Eq. (2.7) may be used to determine an allowable stress for safe design when the crack size is known.

However, one of the stumbling blocks to such an effort is the introduction of a concept using flawed components, which has no counterpart (in relation to the stress/strength design method) in conventional design. The only way such a relationship between stress level and resistance to crack growth can be used is to know the specific flaw sizes, and this by its very nature forms no part in the normal design process. In other words, continuity between the design method for flawed components based on fracture mechanics, and unflawed parts based on continuum mechanics, is definitely lacking.

The second major problem that has created a great deal of confusion among engineers and designers, is related to a lack of a clear definition regarding the ranges of the characteristic dimension, c, of flaw size.

Unfortunately, to many fracture mechanics analysts and designers, "a crack is a crack" with little concern about the limiting values of flaw size c (other than just being very small) within which the fracture mechanics approach is valid. For example, should sizable casting flaws or weld defects, such as shown in Figure 2.2, be simply treated as crack-like discontinuities, or are they more appropriately regarded as notch-like discontinuities? Furthermore, one may even argue that under such circumstances, both methods may have to be utilized in order to describe the total material resistance to failure.

The previous discussions are not intended to dispute that each technique, either continuum or fracture mechanics, has its own place. However, one should not overlook the fact that the proper application of these methods depends heavily upon the definition of limiting values of the flaw size dimension, c, which has not yet been clearly established.

Figure 2.2 WELD DEFECTS

2.5 Definition of Fatigue Failures

Fatigue failures are physical damage to materials as a result of excessive internal energy consumption by the materials caused by plastic strain excursions under fluctuating environments. The evidence of internal energy dissipation can be observed in the generation of hysteresis loops in the stress-strain responses and the changes in the microstructure of the materials as the excursion cycle increases.

In fact, the fatigue life of metallic materials generally exhibits an inverse proportionality to the range of strain (or stress) excursion, but in a nonlinear fashion (often in the form of a power function), since the amount of energy dissipation per hysteresis loop is not linearly proportional to the range of the strain (stress) cycle.

In general, fatigue failures involve the following physical processes:

1) Localized deformation either in the form of localized plastic flow or constrained plastic flow,
2) Continuous deformation and cumulative damage,
3) Crack initiation/early crack growth - irreversible damage,
4) Progressive damage towards final fracture.

Consequently, based on load and life requirements, it is the function and responsibility of engineers to establish the analysis/design methodology to provide for safety and durability in structural components.

2.6 Fatigue Design Philosophy

Structural fatigue design philosophy has been developed along two major lines:

1) Safe-life concept,

2) Damage tolerant approach

2.6.1 Safe-Life Fatigue Design Concept in Line with Continuum Mechanics

Safe-life fatigue design is based on the assumption that the materials are essentially free of flaws and defects or at least operated at a stress (or strain) level that is too low to propagate cracks from any crack-like discontinuities if they do exist. In other words, safe-life design indicates that no catastrophic fracture will occur prior to the design lifetime.

It is noted that the safe-life procedures have been developed in line with continuum mechanics considerations. The safe-life concept implies that at a given stress or strain range of operation, a component will not fail during its intended life cycles. However, it must be realized at the outset of any fatigue design, that safe-life or safety are relative terms that must be quantified. A designer may find that the finite safe-life approach may be used to obtain the desired performance goal. However, if infinite-life is desirable, then the use of fatigue endurance limit (in terms of stress or strain) may be preferable.

It should be noted that a very limited number of materials have a true fatigue stress limit designated for infinite life cycles of operation. Even among those, the fatigue stress limit could be eradicated by repeated, even though occasional, overloads, such as start-ups and impacts.

Therefore, when the safe-life design concept is used, it generally becomes necessary to rigorously analyze and test the component to establish that the probability of failure is extremely remote for the intended life of the structure. In the end, it is prudent to conduct fatigue tests of sample specimens. Nevertheless, both the stress-life and strain-life methods adopt such an approach.

2.6.2 Damage Tolerant Approach in Conjunction with Fracture Mechanics

The damage tolerant design approach is based on the assumption that the structure is considered "damaged", characteristically represented by inherent flaws and defects, by some means, and it will continue to function with the "damage" present. Thus, a structure that is designed to be damage tolerant must sustain its load-carrying capability until the damage is detected and remedial action is completed prior to reaching a complete fracture.

Generally, the damage tolerant design approach is based on the application of fracture mechanics, assuming the existence of crack-like discontinuities in the structure prior to load application. Then, during exposure to service environments, the inherent discontinuity may either grow or not grow depending upon the threshold condition defined for the discontinuity in question. If growth occurs, then the rate of propagation is of interest. Therefore, in addition to the stress (or strain), the following information is needed:

1) Description of inherent discontinuity, size, type and location,
2) Threshold values for fatigue crack growth,
3) Fracture toughness,
4) Crack propagation properties.

Obviously, these relate to the importance of the fatigue crack growth threshold. In order to employ a damage tolerant design philosophy, a rational design method must exist and a fatigue life estimation capability must be an integral part of this procedure. This is where the application of the fatigue crack growth threshold concept fits into the overall fatigue design challenge.

In practice, analysts provide the threshold (or critical) values related to material properties, often determined by experimental means, and designers attempt to size the components to prevent the occurrence of prescribed failure modes within a stated risk. In a sense, this approach is similar to the safe-life concept and requires a great deal of laboratory testing to generate the characteristic material property data and this data must still be subjected to statistical scrutiny.

2.7 Measures of Material Resistance to Fatigue

Despite disagreements among researchers regarding the methodology applicable to fatigue life prediction, the macro-fatigue failure process generally includes three periods (or regions), depicted in Figure 2.3:

1) Crack initiation in conjunction with early micro-crack growth,
2) Crack propagation after the formation of a finite crack-like discontinuity,
3) Final rapid growth toward complete fracture.

Consequently, in fatigue analysis of structures, it is important to distinguish between fatigue life governed by crack initiation, and that controlled by crack propagation. In general, the components that do not contain severe stress risers such as crack-like imperfections, can be analyzed in terms of the material strength to resist fatigue crack formation, and a continuum mechanics based approach, such as the strain-life method, is considered satisfactory for the determination of the fatigue life associated with crack initiation (first period). On the other hand, for components containing severe stress risers or inherent crack-like imperfections, fatigue life is governed by the toughness against propagation, and a fracture mechanics approach is appropriate. In fact, the number of cycles required to sharpen and initiate a fatigue crack from an existing crack discontinuity is generally negligible in comparison to the total number of cycles required for the crack to grow to a critical size and fail the member (second period).

However, it is noted that the current state-of-the-art (fracture mechanics) still has some difficulties dealing with problems involving the effects of mean

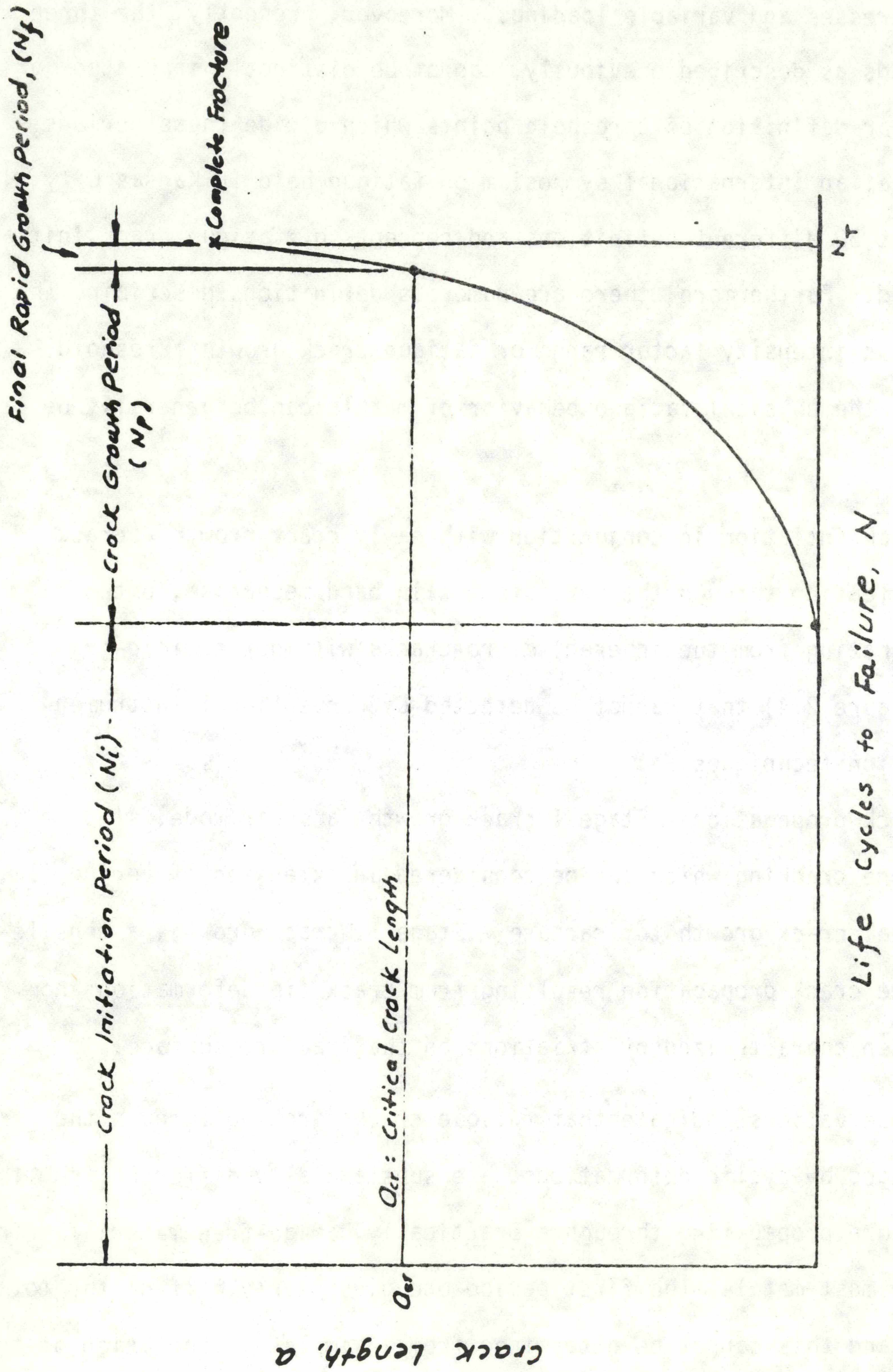
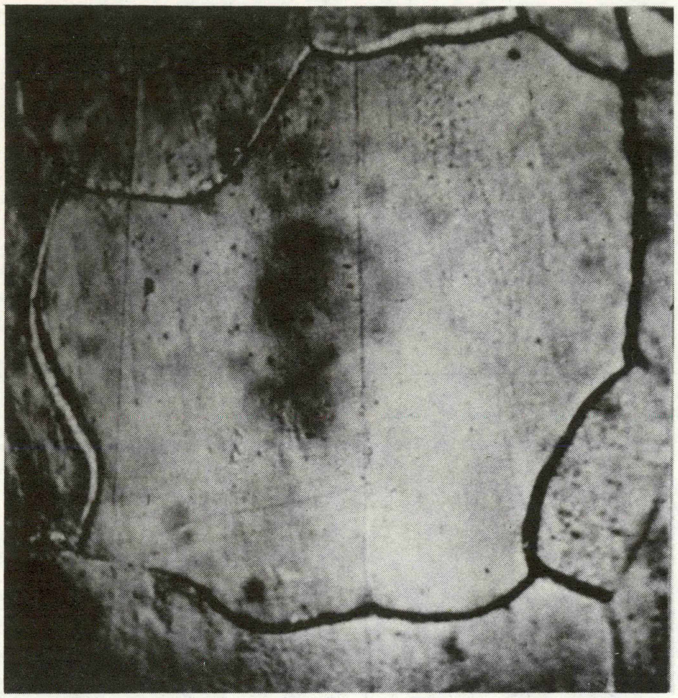


Figure 2.3 SCHEMATIC ILLUSTRATION OF MACRO-FATIGUE FAILURE PROCESS

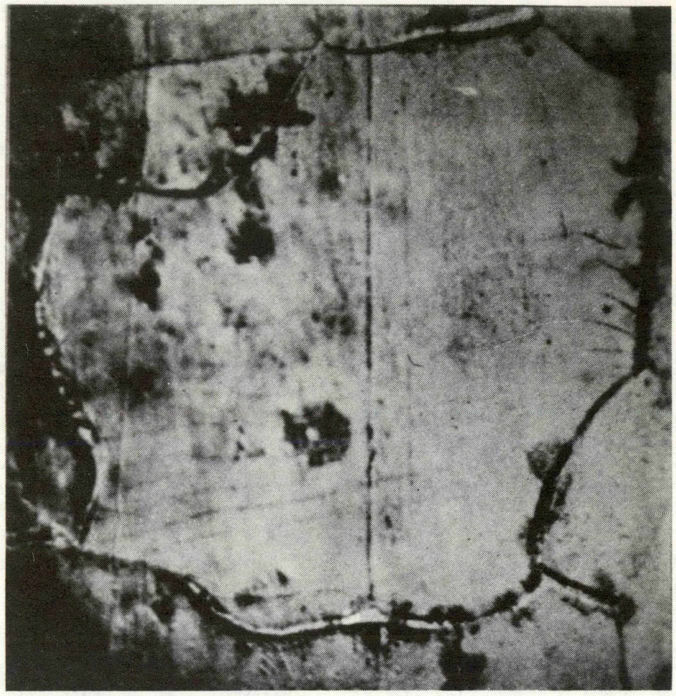
(residual) stresses and variable loadings. Moreover, presently, the three fatigue periods as described previously, cannot be distinctly seperated for a lack of clear definition of threshold points which divide these periods. For example, at an international symposium on fatigue held in Kansas City, in 1978, at least 30 different definitions and concepts of fatigue crack initiation were presented. Furthermore, there are numerous definitions describing the critical stress-intensity factor range or fatigue crack growth threshold. Nevertheless, the physical fatigue behavior of metals can be generally described as follows:

1) Crack intiation in conjunction with early crack growth - crack nucleation through the persistent slip band mechanism, often emanating from the inherent micro-cracks within the micro-grains (Figure 2.4) that cannot be detected by conventional instrumentation techniques,
2) Crack propagation - Stage I crack growth, a shear mode, slip-plane cracking which can be considered an extension of period (1),
3) Final crack growth to fracture - Stage II crack growth, a tensile mode crack propagation resulting from crack tip deformation and often characterized by striations on the fracture surface.

These observations indicate that fatigue cracks growing through the material damaged by cyclic deformations, are substantially different from the brittle fracture propagating through a practically damage-free material. In addition, for most metals, the first period occupies nearly half of the total fatigue life and this cannot be determined from a simple extrapolation of the results obtained for the crack propagation (second) period, shown in Figure 2.5

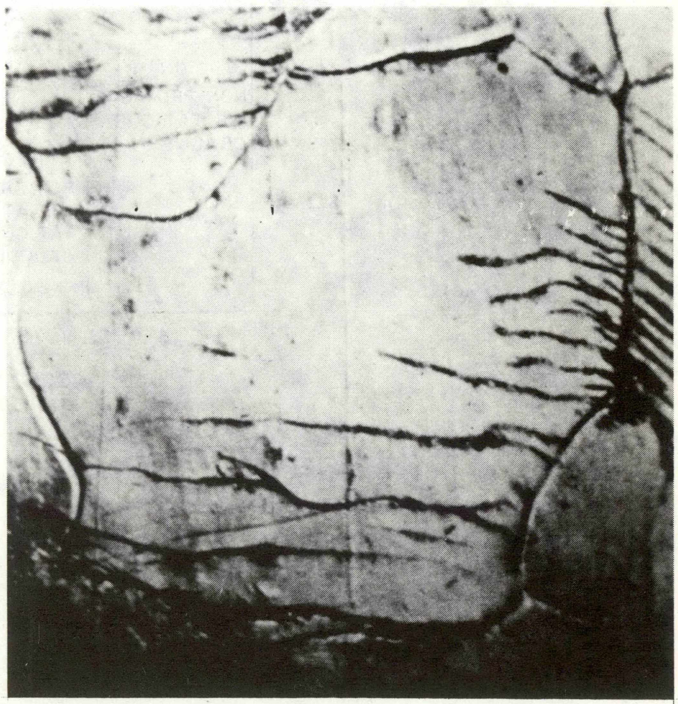


Specimen after 1000 reversals of a stress of 24,800 psi. x 1000.



Same after 2000 reversals. x 1000.

Same after 10,000 reversals. x 1000.



Same after 40,000 reversals. x 1000.

Figure 2.4 EARLY CRACK GROWTH IN CRACK INITIATION PERIOD

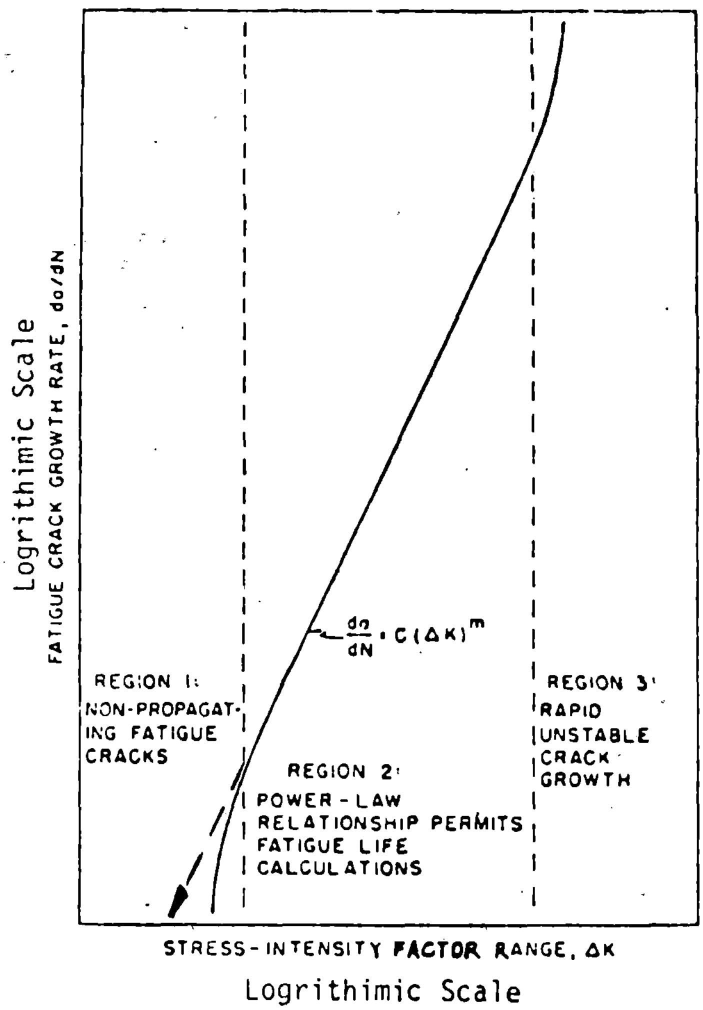


Figure 2.5 SCHEMATIC OF TYPICAL CRACK GROWTH DATA

as suggested by fracture mechanics experts. These observations are also true for components containing sizable defects and notches, such as those found in weldments, for which the crack initiation/early growth phase represents a substantial portion of the material fatigue resistance.

Therefore, in these and similar cases, measurements of the fatigue resistance of materials must be determined for each period, characterized by parameters applicable to material performance and geometric configuration of structural components, and the use of a combined initiation-propagation (I-P) model may become desirable for the determination of material total fatigue resistance.

2.8 Strain-Life Fatigue Analysis Methods

2.8.1 Theoretical Considerations

2.8.1.1 Theoretical Basis and Assumptions

- Strain-life approaches assume the continuum concept.
- Materials are assumed to be homogeneous and isotropic.
- The material is treated on the basis of macroscopic observation, and less attention is paid to the behavior of molecules and/or crystals of which it is composed.
- Cyclic material properties differ from those under monotonic loadings.
- The failure behavior at a notch is identical to that of a smooth specimen when subjected to the same strain history. Hence, the fatigue behavior of a notched specimen is closely correlated to that of unnotched component by a transfer function, designated as the "fatigue notch factor, K_f", which depends upon the stress concentration factor, K_t, and the notch geometry.
- The strain at notches follows the extension of Neuber's rule which states that the fatigue notch factor, K_f, is equal to the geometric mean of the actual stress and strain concentration factors.
- The linear cumulative damage concept holds.

2.8.1.2 <u>Criticisms</u>

- Materials have inherent geometric, material and manufacturing discontinuities. This will shorten the crack initiation period to a minimum, such that the propagation period can represent practically the total fatigue life of the materials. Thus, the fracture mechanics approach should be used.

- The strain-life analysis essentially lumps all influential variables, such as changes in microstructure, relief of stress concentration caused by plastic flow at discontinuities, etc., into one single quantity at a given strain excursion level. That quantity is often represented by life or cycles to failure or other related criteria. While offering simplicity, this procedure has difficulties associated with the physics of fatigue (or mechanism of failure).

- The strain-life analysis, similar to the stress-life approach, is empirically based with little theoretical continuum mechanics considerations.

- The methods may be useful, if the required data are available. However, it is extremely difficult to generate strain history data, either directly from on-road tests, or converted from on-road load data.

- The local strain spectrum is independent of loading spectrums obtained individually for the vertical, lateral and longitudinal directions.

- The application of this technique in environments much removed from conditions under which life data are obtained (often in the laboratory), is questionable.

- The formulation of the transfer function, K_f, is rather complex and often inaccurate, either analytically or experimentally, particularly under the conditions of complex 3-D stress states. The strain-life analysis establishes such a transfer function based on idealized continuum mechanics with very restrictive boundary conditions, such as a uniform nominal stress field assumption at the boundary, which can hardly be realized in most applications.

- Strain at notches usually does not obey the Neuber relation. Instead the Smith-Topper-Watson equation is recommended, which implies that there can be no fatigue failures without alternating tensile stresses and thus takes care of many non-propagating cracks.

- A fundamental drawback of all present fatigue models, including the strain-life method, lies in their inability to translate the one-dimensional test specimen data to nonhomogenous material and/or load situations at the site of damage such as the crack front where a multiaxial stress state is experienced.

- Simultaneous development of the equally important technologies related to load determination and structural analysis techniques are required for the needed load-strain conversion.

- A very costly and long-term development effort and commitment is envisioned, if the strain-life approach is to be used as a standard tool for structural fatigue design, since there is still a considerable gap between the theory and practical applications.

2.8.1.3 Basis for Justification

- Cyclic fatigue material properties, instead of monotonic, are used for the characterization of material responses to fluctuating environments.

- Plastic strains experienced at notches result mostly from a high degree of stress (or strain) concentration. However, the behavior of the local plastic zone is somewhat dictated by the elastic deformation of the surrounding materials. Hence, it is the strain controlled deformation at notches to which the strain-life assumptions apply.

- In general, most inherent discontinuities in structures are not Griffith's type of crack-like discontinuities, to which the theory of fracture mechanics applies, even though they may grow to become crack-like discontinuities later under cyclic loading. Hence, the initiation-propagation model should be employed for most inherent flaws and defects.

- Most fatigue failures have been observed to start on material surfaces, particularly at the stress raisers such as the toes of welds or notches, where the maximum strain excursions due to cyclic loading would be experienced. Hence, a mere application of the fracture mechanics model appears inappropriate.

- The primary interest to railroad equipment design is the fatigue life associated with the crack initiation period. Thus the crack propagation phase becomes of secondary concern.

- The use of Neuber's rule is justifiable.
- Fatigue life prediction is more art than science. It is an engineering - scientific art. Despite its imperfect basis for formulating the required methodology, the strain-life analysis approach has been accepted by many industries as a viable tool for fatigue design. It has been especially welcomed by the automobile industry in which periodic scheduled inspections are deemed impossible to enforce.

2.8.2 Benefits and Limitations

The advantages of using strain-life analysis, especially over stress-life analysis, include:

- Use of physical quantity (strain) rather than non-measurable quantity (stress), and cyclic material properties in fatigue analysis.
- Accounting of notch effects, especially local plasticity, high stress (strain) gradient and strain-controlled deformations.
- Correct assessment of mean stress influences, if a sequential analysis is performed.
- No problems with eradication of endurance limit by occasional overloads.
- More accurate fatigue life estimates particularly for ductile metals.

On the other hand, the disadvantages are:

- Requirement of on-road local strain data, or
- Accurate estimation of the "K_f" factor to be used in conjunction with on-road nominal strain data, or
- Transfer function needed for the load-strain conversion from on-road load data.

Moreover, pitfalls are related to the predictions of fatigue life, which are only as accurate as the laboratory tests are in duplicating field conditions.

Finally, the applications of strain-life analysis at this point exclude:

- Composite materials,
- Long-life fatigue range, in which the effects of surface finish appear important.
- Environmental impacts, such as corrosion, temperature.
- Multiaxial fatigue.
- Contact fatigue.

Section 3

BASIC STRAIN-LIFE ANALYSIS FORMULATION

Four major steps are required to perform strain-life analysis:

1) Understanding of material behavior under cyclic loading,
2) Development of basic strain-fatigue life relationships,
3) Determination of the relationship between the applied loads and resulting strains at the local level,
4) Evaluation of fatigue damage incurred and prediction of fatigue life.

At first, a simple, yet comprehensive, summary of the strain-life fatigue analysis formulation is presented in this section. The summary is prepared in a format which can be easily understood and used by engineers/designers who are unfamiliar with the strain-life analysis techniques. This format can also be incorporated into the current AAR specification to enhance the fatigue design guidelines.

The summary includes:

1) Basic cyclic stress-strain relation of material,
2) Strain-life fatigue analysis methods.

In addition, the fundamentals of strain-life analysis are presented in Appendix C, which contains the theoretical basis, in great detail, for the development of strain-life analysis.

In the subsequent subsection, data required to perform fatigue analysis using the strain-life approach are identified. A step-by-step precedure for performing strain-life analysis, depending on the availability of basic data in order to yield local strain history, is also given.

Various cyclic counting techniques, that can be utilized to assess fatigue damage associated with random loading input, are discussed.

Finally, the methods for estimating fatigue life are presented.

3.1 Steady-State Cyclic Stress-Strain Relation

$$\varepsilon = \varepsilon_e + \varepsilon_p = \frac{\sigma}{E} + \left(\frac{\sigma}{K'}\right)^{1/n'} \qquad (3.1)$$

where ε = True strain

ε_e = Elastic strain component

ε_p = Plastic strain component

σ = True stress (psi)

K' = Cyclic strength coefficient (psi)

n' = Cyclic strain hardening exponent

= 0.10 ~ 0.20 (0.15 average)

E = Elastic modulus (psi)

3.2 Strain-Life Analysis Methods

3.2.1 Local Strain-Fatigue Life Approach

$$\frac{\Delta\varepsilon}{2} = \underbrace{\frac{\sigma}{E}(2\,N_f)^b}_{\text{Elastic Region}} + \underbrace{\varepsilon_f'\left(\frac{\sigma}{\sigma_f'}\right)^{c/b}(2\,N_f)^c}_{\text{Plastic Region}} \qquad (3.2)$$

where $\Delta\varepsilon$ = Strain excursion

ε_f' = Fatigue ductility coefficient $\cong \varepsilon_f \cong \ln \frac{1}{1 - RA}$

RA = Area reduction percent (%)

σ = $\sigma_f' - \sigma_o$

σ_f' = Fatigue strength coefficient (psi)

$\cong \sigma_f$ (True fracture strength) $\cong (S_u + 50{,}000)$ (psi)

S_u = Ultimate strength of material (psi)

σ_o = Mean stress (psi)

b = Fatigue strength exponent

$$\cong -\frac{1}{6} \text{Log} \frac{2\sigma_f}{S_u} = -0.05 \sim -0.12 \ (-0.085 \text{ average})$$

c = Fatigue ductility exponent = - 0.5 ~ - 0.7

$$= \begin{cases} -0.6 \text{ for } \varepsilon_f \cong 1.0 & \text{(ductile materials)} \\ -0.5 \text{ for } \varepsilon_f \cong 0.5 & \text{(strong materials)} \end{cases}$$

N_f = No. of cycles to fatigue failure.

Moreover, the following relationship between the material data generally hold:

$$n' = \frac{b}{c}$$

$$K' = \frac{\sigma_f{}'}{(\varepsilon_f{}')^{n'}}$$

It should be noted that this method requires the local strain history data.

3.2.2 Nominal Strain (or Stress) - Fatigue Life Approach in Conjunction with Neuber's Rule

a) Neuber's Rule

$$(K_f \Delta S)^2 = (\Delta\sigma \Delta\varepsilon E) \tag{3.3}$$

where

K_f = Fatigue notch factor

$$= 1 + \frac{K_t - 1}{1 + \frac{a}{r}} \quad \text{(by Peterson)}$$

K_t = Theoretical stress concentration factor

$$a \cong \left(\frac{300}{S_u\,(ksi)}\right)^{1.8} \times 10^{-3} \quad \text{(in.)}$$

$$\cong \begin{cases} 0.01 & \text{for normalized or annealed steels} \\ 0.001 & \text{for highly hardened steels} \\ 0.0025 & \text{for quenched and tempered steels} \end{cases}$$

r = Notch radius

ΔS = Nominal stress range

$\Delta\sigma$ = Notch stress range

$\Delta\varepsilon$ = Notch strain range

b) Cyclic Stress-Strain Relation

$$\frac{\Delta\varepsilon}{2} = \frac{\Delta\sigma}{2E} + \left(\frac{\Delta\sigma}{2K'}\right)^{1/n'} \tag{3.4}$$

c) <u>Nominal Stress - Notch Stress Relation</u>

$$\frac{(K_f \Delta S)^2}{4E} = \frac{(\Delta\sigma)^2}{4E} + \frac{\Delta\sigma}{2}\left(\frac{\Delta\sigma}{2K'}\right)^{1/n'} \quad (3.5)$$

d) <u>Notch Stress - Fatigue Life Relation</u>

$$\frac{\Delta\sigma}{2} = \sigma(2\ N_f)^b \quad (3.6)$$

e) <u>Notch Strain - Fatigue Life Relation</u>

$$\frac{\Delta\varepsilon}{2} = \frac{\sigma}{E}(2\ N_f)^b + \varepsilon_f'\left(\frac{\sigma}{\sigma_f'}\right)^{c/b}(2\ N_f)^c \quad (3.7)$$

f) <u>Neuber's Rule - Fatigue Life Equation</u>

$$(K_f \Delta S)^2 = (K_f \Delta e E)^2$$

$$= \underbrace{4\sigma^2(2\ N_f)^{2b}}_{\text{Elastic Region}} + \underbrace{4\sigma E\varepsilon_f'\left(\frac{\sigma}{\sigma_f'}\right)^{c/b}(2\ N_f)^{c+b}}_{\text{Plastic Region}} \quad (3.8)$$

where:

$\Delta e = \gamma_e(\Delta P)$ = Nominal strain range

$\Delta S = \gamma_s(\Delta P)$ = Nominal stress range

γ_e = Nominal strain-load conversion factor

γ_s = Nominal stress-load conversion faction

ΔP = Load excursion history data

and $\sigma = \sigma_{f'} - \sigma_o$

For small nominal strain (stress) excursions,

$$N_f \cong \frac{1}{2}\left[\frac{K_f \Delta S}{2\,(\sigma_f{}' - \sigma_o)}\right]^{\frac{1}{b}} \tag{3.9}$$

It should be noted that this method can only be applied when

- ΔP and the appropriate values for γ_e or γ_s are available.
- ΔP is a single loading.
- The sequence effect on ΔS (Δe) is identical to that on ΔP, during the cycle counting process.

3.3 Data Requirements and Strain-Life Analysis Step-by-Step Procedures

In general, the following steps are required for performing strain-life fatigue analysis:

1) Generation of cyclic material properties, including:

   a) Fatigue strength coefficient and exponent, σ_f' and b

   b) Fatigue ductility coefficient and exponent, ε_f' and c

   c) Cyclic hysteresis loops and stress-strain curve

   d) Cyclic strength coefficient, K'

   e) Cyclic strain hardening exponent, n'

   f) Cyclic yielding strength, σ_y

   g) Elastic modulus, E

   If these material properties are unknown, they have to be determined by a series of fatigue tests or estimated from monotonic material data, such as hardness, ultimate strength, percentage area reduction, fracture strength and ductility, etc.

2) Determination of stress concentration factor, K_t

   The value of K_t can be determined either from Peterson's stress concentration tables for simple stress raisers, finite-element analysis for complex configurations, or from experimental stress analysis methods applied to actual components.

3) Determination of fatigue notch factor, K_f

   The value of K_f can be found either experimentally or analytically with Peterson's formula. For weldments, it is recommended that Peterson's formula be exercised, in conjunction with a series of finite-element computations, for various notch sizes, r, to determine (K_f) max.

4) Construction of strain-life relation of Eq. (3.2), or curve as shown in Figure C.9 , or Figure C.11 which includes the effects of mean stress.

5) Establishment of Neuber's parameter-fatigue life relationship of Eq. (3.8) or curve as depicted in Figure D.12.

6) Creation of records of nominal loading or nominal strain history.

7) Rainflow counting of loading, nominal strains, or notch strains, either including or excluding the mean stress determination.

8) Estimation of cumulative damage and fatigue life prediction.

Of course, data requirements depend upon the method of approach employed. The most popular approaches are described as follows:

3.3.1 Smooth Specimen Nominal Strain-Life Approach

The nominal strain-life analysis is generally used to predict the crack initiation life, which is particularly suitable for low-stress/long-life prediction. This type of analysis uses smooth specimen material property data. The analysis procedure involves:

1) Smooth specimen material property data,
2) Theoretical strain/stress concentration factor, K_t,
3) Nominal strain (e) history, recorded or obtained from field loading data if nominal strain data are not available,
4) Conversion of nominal strains to notch strains through

$$\varepsilon = K_t \times e$$

It is noted that the elastic behavior (the elastic branch of strain-life curve) is assumed in steps 2, 3, and 4, for the nominal far-field and also at the notch root,

5) Rainflow counting of notch strains on a cycle-to-cycle basis or from a histogram of rainflow counted ranges,

6) Estimation of fatigue damage for each strain cycle, from the strain-life relation of Eq. (3.2).

7) Calculation of fatigue life from Miner's rule.

It is noted that the effects of mean stress are excluded in this approach. By assuming elastic behavior, the mean stress could be calculated from mean strain. However, it was found that the inclusion of a mean stress effect could lead to considerable error in some cases. Sequence effects are taken into account only in the rainflow counting procedure.

3.3.2 Direct Component Nominal Strain-Life Approach

An alternate approach to the smooth specimen approach is to determine the nominal strain-life curve directly for the individual component through a series of constant amplitude strain-controlled tests. Then this strain-life curve is used in place of the actual material properties, and the notch strains (stress concentrations) need not be determined. This method of approach is sometimes useful in evaluating the fatigue life of welded structures in which the stress concentration factor (the fatigue notch factor) and mean stresses are difficult to obtain.

3.3.3 Component Calibration (Load-Strain) Approach

Component calibration refers to a method in which applied loads are converted to notch root strains retaining all sequence effects. This method is applicable for both elastic and plastic strains and requires a relationship between applied load and notch root strain. Such a relationship can be obtained experimentally by mounting a strain gage over the notch root, or theoretically through a finite element model involving plastic analysis of notched plates. A piecewise linear solution is obtained for each load increment. Cyclically stable properties for the cyclic strength coefficient and strain hardening exponent are used in the analysis.

The advantage of the component calibration technique is that it accounts for notch root plasticity if it exists, and it does not require the determination of the stress concentration factor. The steps, as outlined below, do not include the effect of mean stress, which is discussed in the next method of approach (Section 3.3.4).

1) Material properties required are the same as those used in nominal strain-life analysis (Section 3.3.2),
2) Generation of cyclically steady-state load-strain relations (curves),
3) Loading history,
4) Rainflow counting of loading history to obtain the load ranges,
5) Conversion of load ranges to strain amplitudes, first dividing the load range by two to obtain load amplitude, then determining the corresponding notch root strain amplitude,
6) Calculation of fatigue damage.

It is noted that this approach is almost identical to the nominal strain-life approach (Section 3.3.2), with the exception of the manner in which the notch strains are obtained, directly from applied loads, and that rainflow counting is exercised on loads, not strains.

3.3.4 Load-Strain-Stress Conversion Method

This analysis is a refinement of the previous one by accounting for the mean stress of each load reversal. The strain-stress response of each reversal must be determined on a reversal-by-reversal basis retaining all sequence effects. The strain range and mean stress of each reversal in the load history is obtained from the material stress-strain model.

Computational requirements of the analysis are increased because of the material response calculation needed for mean stress determination. However, once the procedure is programmed for computer usage, the analysis can easily be performed. Fairly accurate life prediction can be obtained when notch root strain and mean stress effects are properly accounted for.

3.4 Complex Loading History and Effects of Loading Sequence

3.4.1 Treatments of Complex Loading History

In order to assess damage associated with random loading, it is necessary to reduce the random history to a series of discrete events by employing some kind of cycle counting procedure. A great number of such procedures have been devised, for example:

- o Peak counting
- o Mean crossing peak counting
- o Level crossing
- o Time at level
- o Range counting
- o Rainflow counting (Ref. C.32)

To appreciate in more detail the problems involved in cycle counting, these counting methods are illustrated in Figure 3.1 through Figure 3.6 respectively. Furthermore, the following discussions will demonstrate the adoptability of rainflow counting to fatigue damage assessment.

Again, it is noted that the stress and strain spectra are quite different without a clear functional relationship. Thus, the nonlinear response evident in the strain-stress record reflects the importance of sequence effects in complex history analysis. This can be demonstrated with a simple example shown in Figure 3.7 with the following noteworthy observations of material stress-strain response behavior:

1) Following the elastic unloading b-c, the material exhibits a discontinuous accumulation of plastic strain upon deforming from c to d. Such behavior is unique to complex histories.

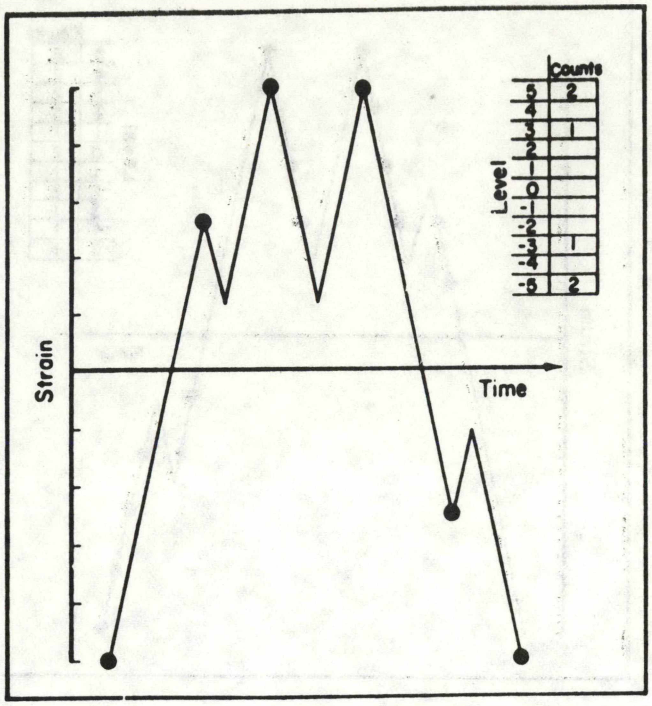


Figure 3. 1 PEAK COUNTING EXAMPLE

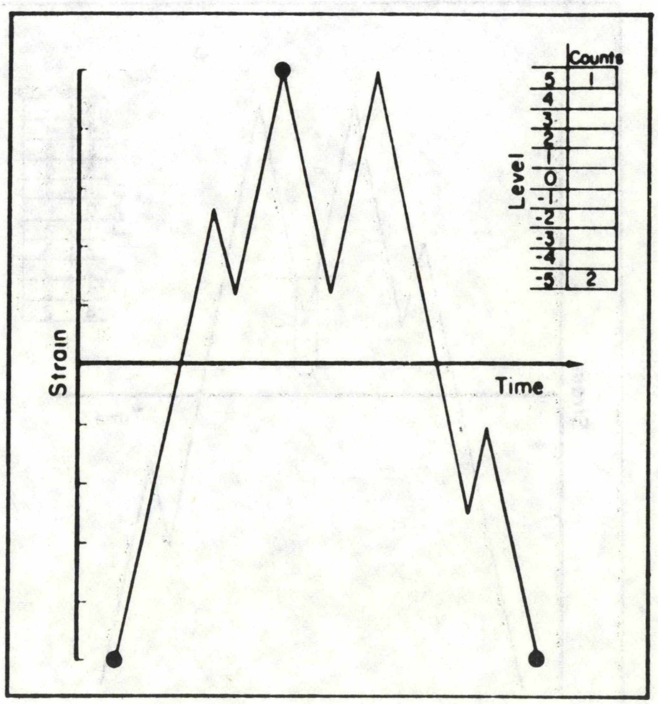


Figure 3.2 MEAN CROSSING PEAK COUNTING EXAMPLE

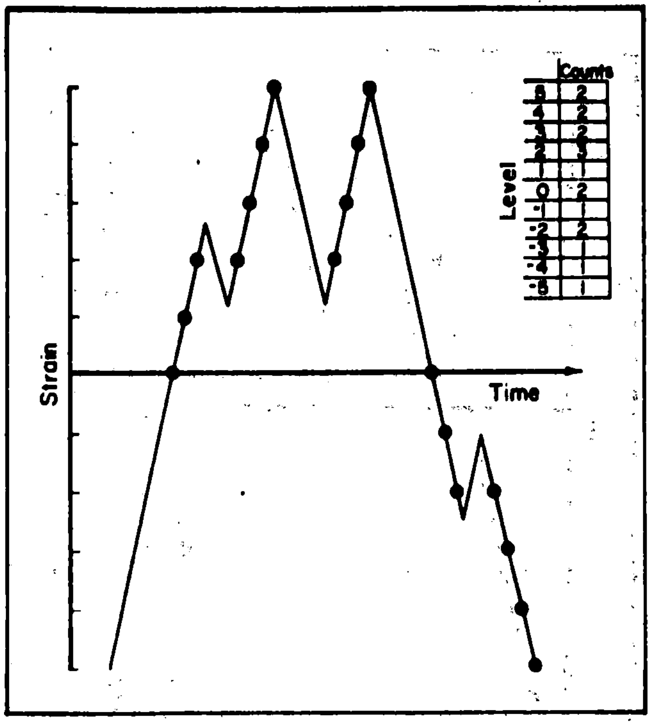


Figure 3.3 LEVEL CROSSING EXAMPLE

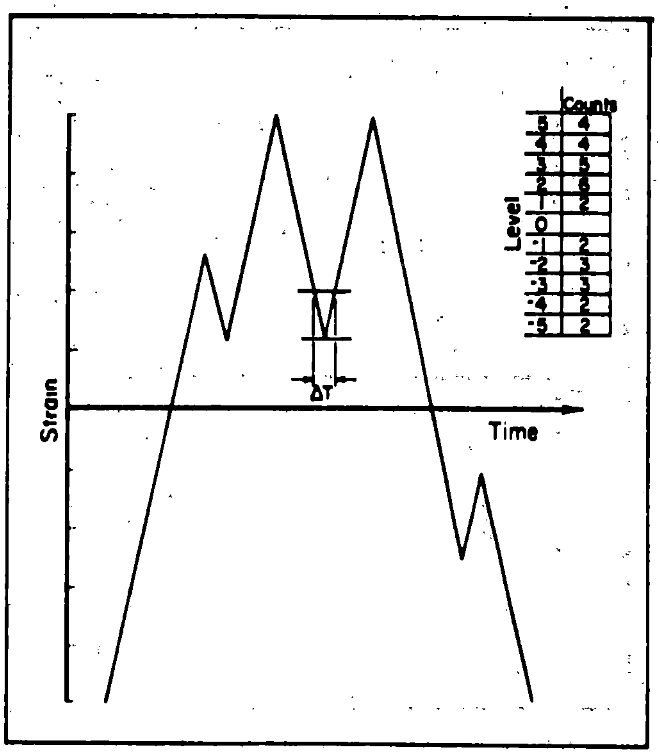


Figure 3.4 TIME AT LEVEL EXAMPLE

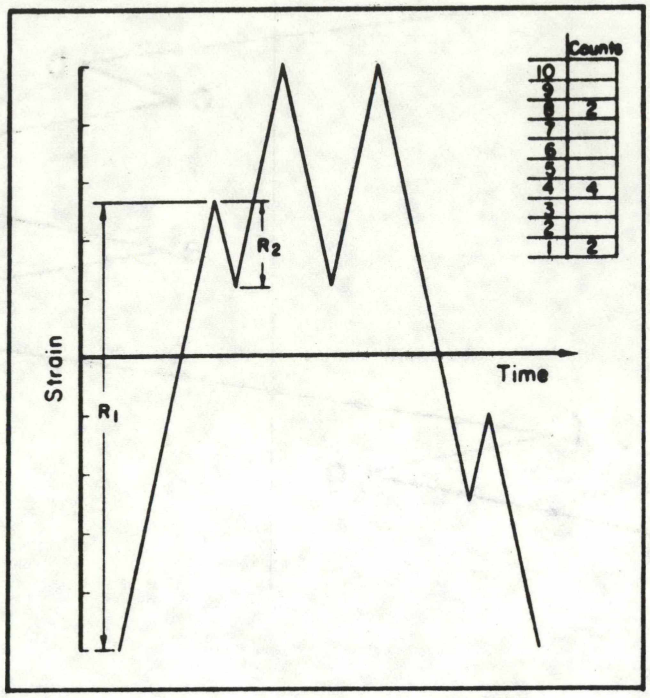


Figure 3.5 RANGE COUNTING EXAMPLE

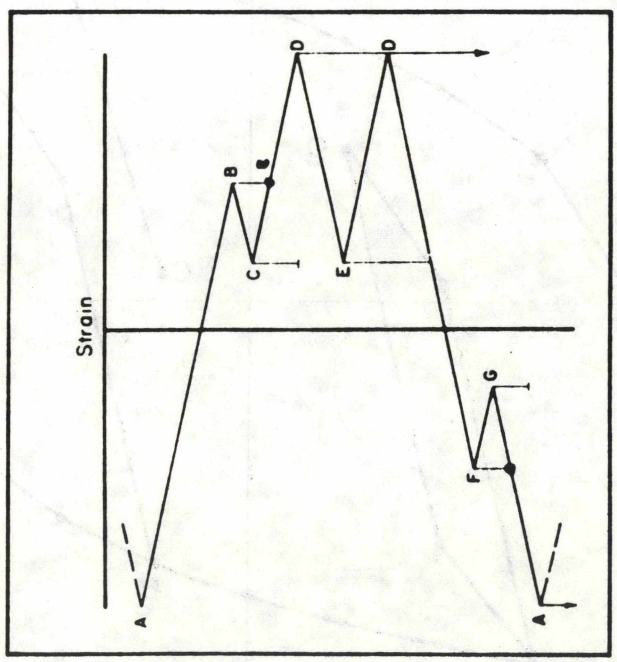


Figure 3.6 RAINFLOW COUNTING EXAMPLE

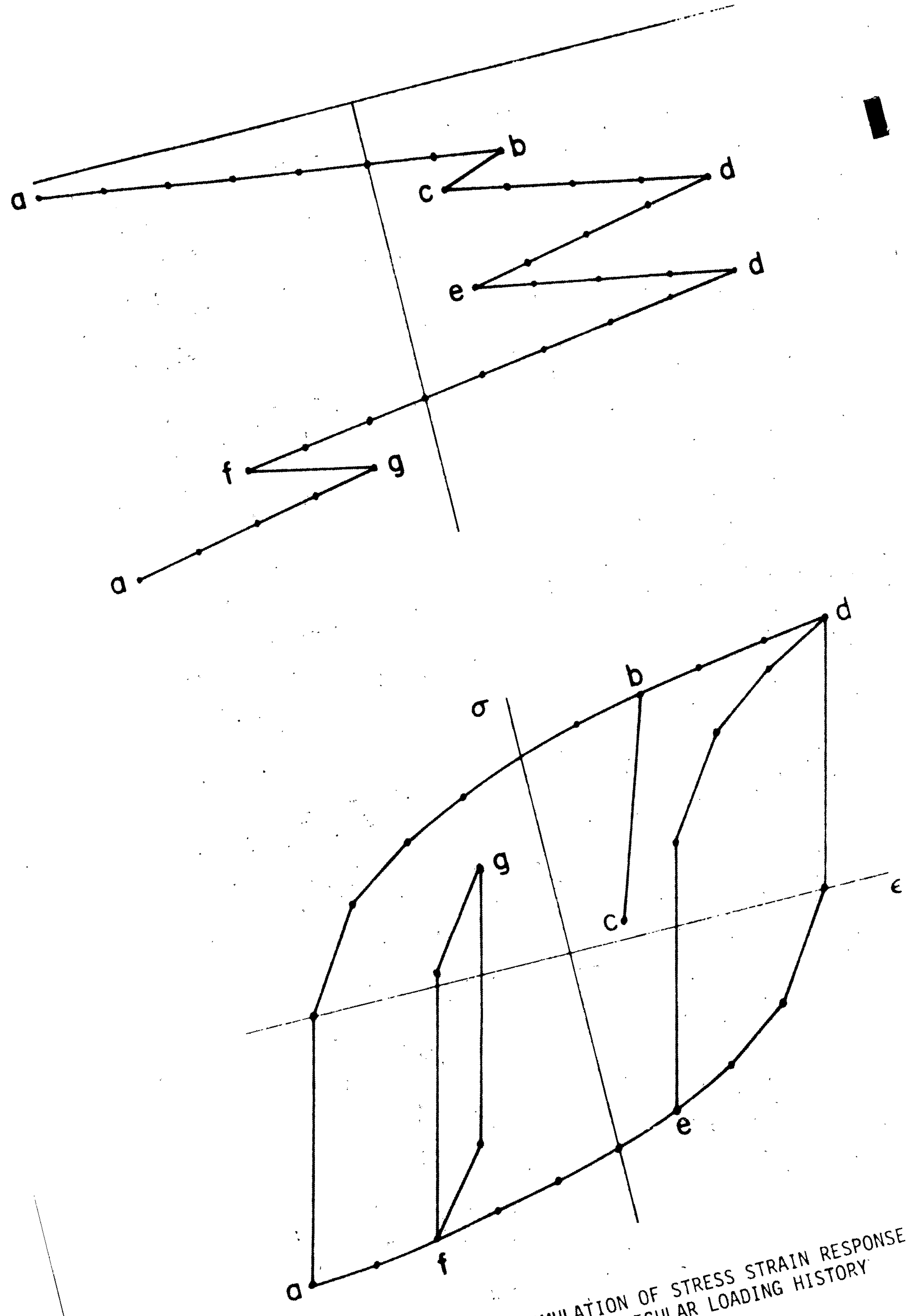


Figure 3.7 SIMULATION OF STRESS STRAIN RESPONSE FOR AN IRREGULAR LOADING HISTORY

2) A comparison of events c-d and e-d reveals that, although they appear identical on the strain-time record, they are not contributing equally to damaging of the material because of different amounts of plastic strain involved.

3) It is easy to recognize events similar in character to those observed in constant amplitude cycling; namely those defined by closed hysteresis loops, such as a-d-a, e-d-e, b-c-b, and f-g-f.

Indeed these observations provide the basis for a counting procedure which has proven superior to others for a wide range of complex histories, and the rainflow counting method is one technique that accomplishes such reduction, as demonstrated in Figure 3.8 The flow lines, as determined in accordance with a set of simple rules, serve to identify strain ranges associated with closed hysteresis loops. The inevitable value of this technique is, then, that it identifies events in a complex sequence which are compatible with constant-amplitude fatigue data.

It should be noted that the precise mean stress associated with specific events cannot be directly computable from the histogram, the corresponding mean strain and the strain range. However, the stress corresponding to the largest (absolute magnitude) strain will always be on the cyclic stress-strain curve. A hysteresis loop is formed from the largest to smallest strain in the history, from which the mean stress for that particular event may be determined, as shown in Figure 3.9 .

3.4.2 <u>Loading Sequence and Memory Effects</u>

It has been shown in the previous example (Figures 3.7 and 3.8) that the stress response is dependent on the path of strain excursions. Additional illustrations will further reinforce this observation.

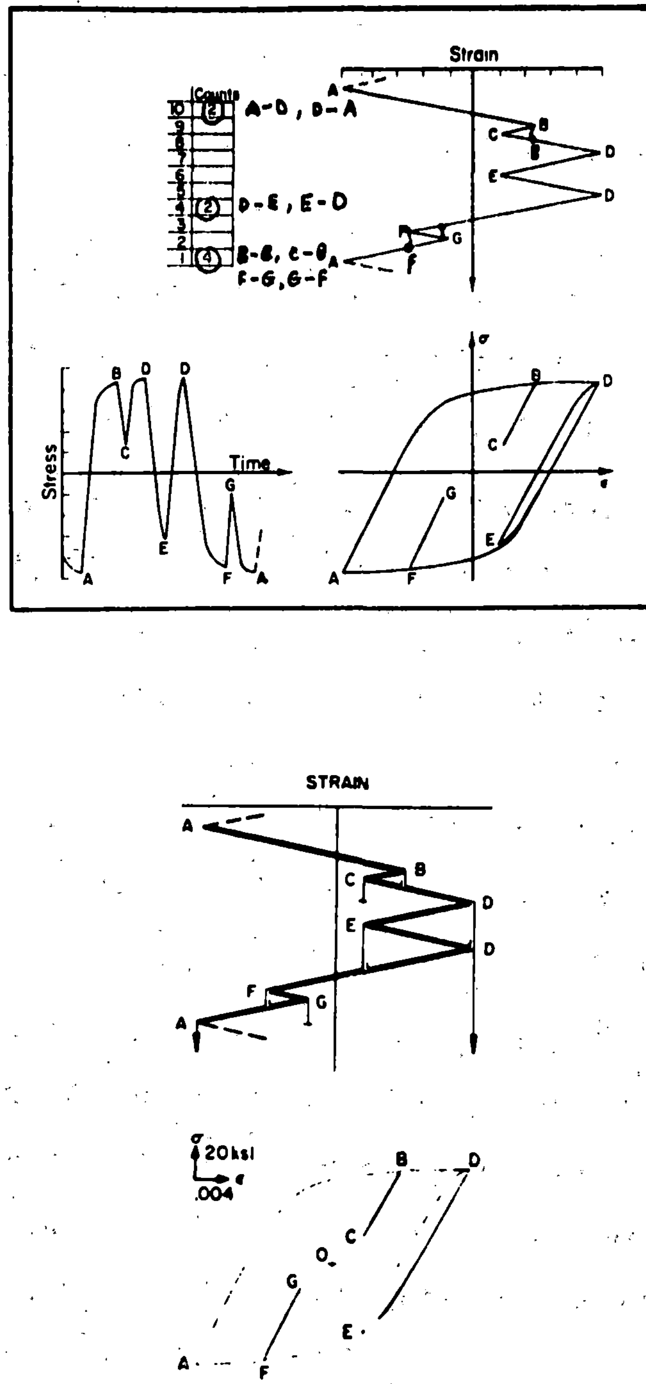


Figure 3.8 STRESS-STRAIN RESPONSE VS. RAINFLOW CYCLE COUNTING PROCEDURE

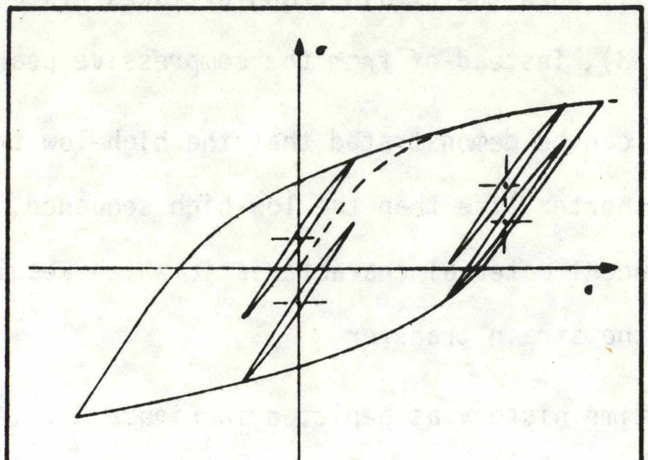


Figure 3.9 MEAN STRESS EXAMPLE

For example, as shown in Figure 3.10, it is postulated that a large strain amplitude is imposed first and after several reversals is transferred to a smaller strain amplitude, from the compressive peak (at point no. 4). As a result, a self-imposed tensile mean stress is developed, as indicated in Figure 3.10. On the other hand, a self-imposed compressive mean stress is created (Figure 3.11) when the strain transfer takes place from the tensile peak (at point no. 3), instead of from the compressive peak.

Similarly, it can be demonstrated that the high-low transfer sequence would result in a shorter life than the low-high sequence, because of the cyclic (path) dependent material characteristic which results in a tensile mean stress after the strain transfer.

For a strain-time history as depicted in Figure 3.12, it is evident that the specimen is cyclically stressed in tension whereas the strains applied are compressive. Moreover, the magnitude of tensile stress increases at every start-up because of the memory effect associated with load sequencing. As a result, the specimen may experience fatigue failure even with application of cyclic compressive strains.

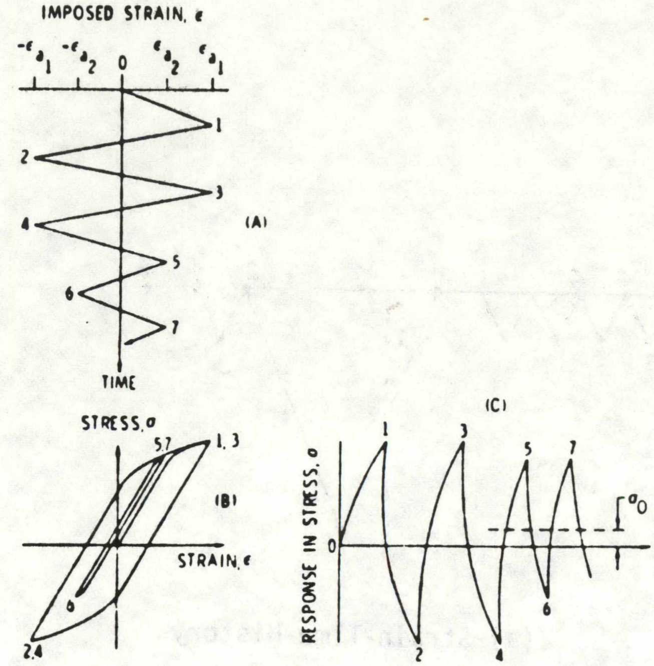


Figure 3.10 DEVELOPMENT OF TENSILE MEAN STRESS BECAUSE OF SEQUENCE EFFECT

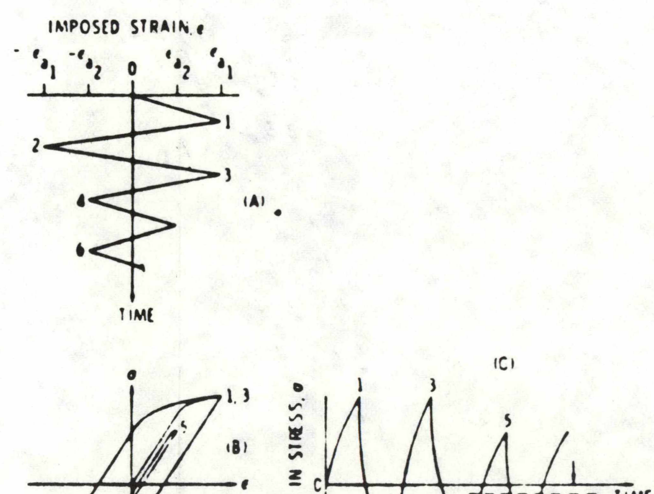
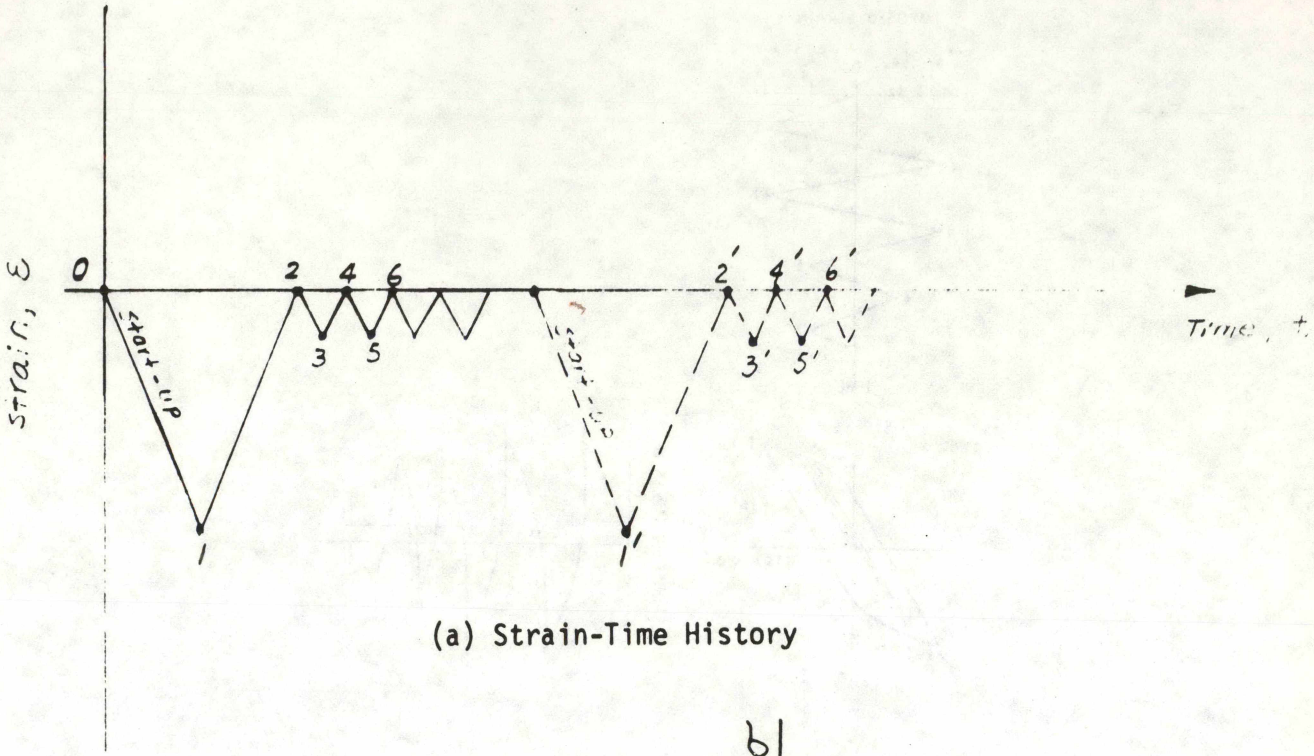
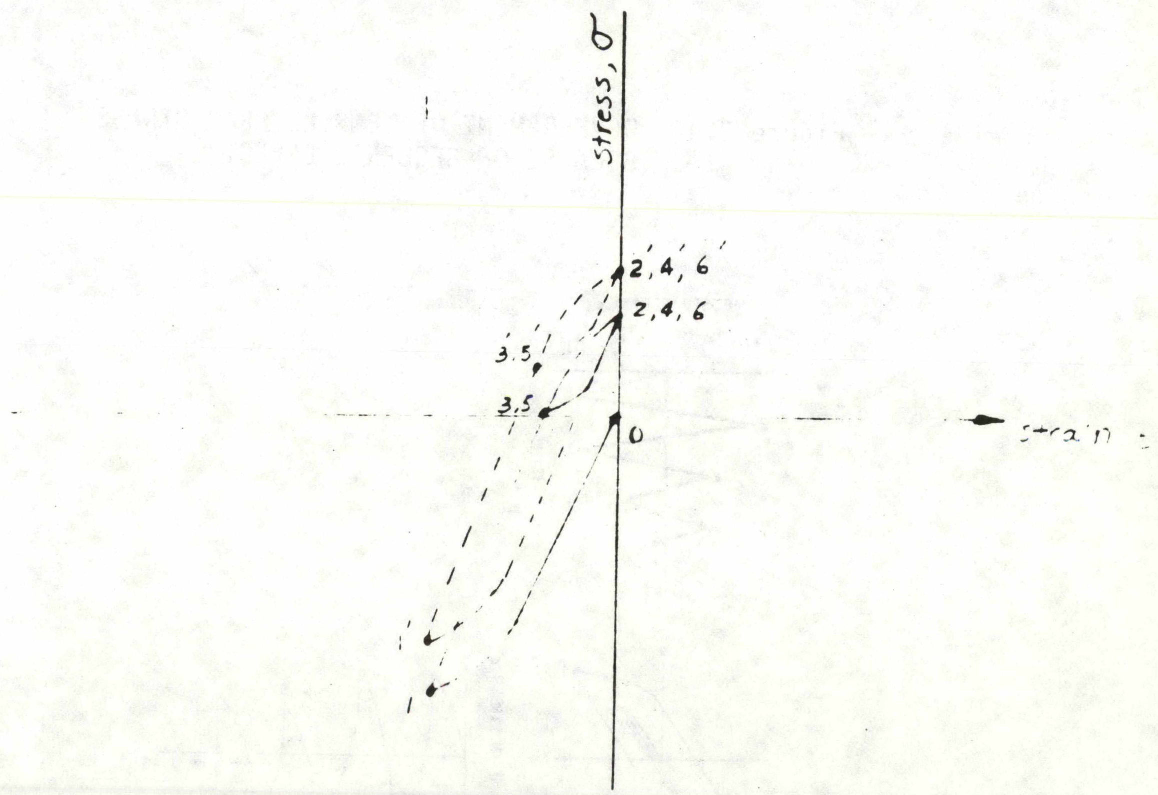


Figure 3.11 DEVELOPMENT OF COMPRESSIVE MEAN STRESS BECAUSE OF SEQUENCE EFFECT



(a) Strain-Time History



(b) Stress-Response History

Figure 3.12 MEMORY EFFECT ASSOCIATED WITH REPEATED START-UPS

3.5 Cumulative Damage/Fatigue Life Estimation

The final steps in arriving at a fatigue life estimate involve assigning damage to individual events in a history, accumulating this damage in some fashion, and calculating how much of the available life has been consumed. A large number of cumulative damage estimating methods have been suggested; however, the linear cumulative damage rule set forth by Palmgren (Ref. C.33) and Miner (Ref. C.34) remains the most widely used.

In the strain-life approach, the linear damage rule is defined in terms of the cycles to failure for each strain level, obtained from the material strain-life relation (or curve). For example, the damage incurred per cycle at a given strain range, $\Delta\varepsilon_i$, can be obtained directly from the strain-life relation of Eq. (3.2) for reciprocal life:

$$\frac{(\Delta\varepsilon)_i}{(\Delta\varepsilon_p)_i} = \frac{(\sigma_f')_i}{(E\,\varepsilon_f')_i} \; \frac{(2\,N_f)_i^b}{(2\,N_f)_i^c} + 1$$

and

$$\frac{(\Delta\varepsilon)_i}{(\Delta\varepsilon_p)_i} - 1 = \frac{(\Delta\varepsilon_e)_i}{(\Delta\varepsilon_p)_i}$$

which yields,

$$\text{Damage/cycle} = \frac{1}{(2\,N_f)_i} = \left[\frac{\sigma_f'}{\varepsilon_f' E} \; \frac{(\Delta\varepsilon_p)_i}{(\Delta\varepsilon_e)_i}\right]^{1/(b-c)} \qquad (3.10)$$

By including a mean stress correction, Eq. (3.62) can be rewritten as

$$\text{Damage/cycle} = \frac{1}{(2\,N_f)_i} = \left[\frac{\sigma_f'}{\varepsilon_f' E} \; \frac{(\Delta\varepsilon_p)_i}{(\Delta\varepsilon_e)_i} \; \frac{\sigma_f'}{\sigma_f' - \sigma_o}\right]^{1/(b-c)} \qquad (3.11)$$

The corresponding damage incurred at each level is then taken as the ratio of the accrued cycles to the total cycles to failure. Failure (often under random loading) is assumed to occur when the sum of the cycle ratios equals unity. These rules can be mathematically stated as

$$d_i = \frac{(2\ N)_i}{(2\ N_f)_i} = \frac{\text{Reversals applied as } (\Delta\epsilon)_i}{\text{Reversals to failure at } (\Delta\epsilon)_i} \qquad (3.12)$$

where d_i = damage ratio corresponding to $\Delta\epsilon_i$

and failure will occur when

$$\sum d_i = \sum \frac{(2\ N)_i}{(2\ N_f)_i} = 1 \qquad (3.13)$$

Section 4

APPLICATIONS OF STRAIN-LIFE ANALYSIS PROCEDURES

This section provides information needed for practical railroad engineers to make use of stain-life techniques in the design of railroad cars for long term service. An example problem is worked out to demonstrate a typical application of strain-life methods, through which the fundamentals and application of strain-life analysis can be easily understood, received and used by the railroad industry.

It has been clearly stated in Section 3 that the strain-life procedures deal with situations in which the local strain-time history is directly available for either smooth specimens or notched components at locations of great strain. However in practice, the local strain-time history is extremely difficult to obtain, particularly at notches (welds), under field conditions and may not be so readily available. Under such circumstances, the nominal (remote field) strain data should be used in conjunction with corrections for notch stress concentration and fatigue notch factor, before entering the strain-life computation.

Considering the fact that the current AAR manual of Standards and Recommended Practices furnishes the REPOS data (Road Environment Percent Occurence Spectrum) for the fatigue analysis, the use of such load data in conjunction with the nominal strain (stress) approach, associated with the fatigue notch factor, can be a feasible combination to complete the strain-life analysis. An outline of such an approach is described in the following:

1) Selection of appropriate load spectrum.

2) Determination of the load-nominal strain (stress) relationship, most likely by means of finite element methods.

3) Determination of the fatigue notch factor, either analytically or by tests.

4) Selection of proper fatigue properties of materials.

5) Computation of fatigue life, utilizing the FORTRAN program developed to obtain solutions for the strain-life equation.

A sample problem of strain-life procedures is worked out by using the basic data provided for the example problem presented on p. C-II-146 of the current AAR manual. This not only demonstrates the application feasibility of the strain-life analysis approach, but also allows a direct comparison of strain-life analysis with the stress-life approach which is currently in use by the AAR.

4.1 FEEST/REPOS Road Data

Over the past several years, the AAR has collected over-the-road load data for various categories of freight cars through the FEEST (Freight Equipment Environmental Sampling Test) program. This field data has been processed by means of the rainflow cycle counting procedures, to yield the REPOS (Road Environment Percent Spectra) data. These data are collectively presented in Section 7.3, Chapter VII Fatigue Guidelines, AAR Specification M-1001, Part C which became effective in July, 1983.

One of these REPOS data sets will be used in a demonstrative example of strain-life analysis in a subsequent section.

4.2 Working Example

An application example of strain-life analysis is demonstrated in this subsection. The result of the strain-life analysis is compared with that obtained by the AAR stress-life method. Before entering into actual computation of the fatigue life prediction, the problem definition requires clarification:

1) This example problem involves only the case of a single loading condition. This is the vertical motion of the bolster for a 100 ton auto part box car. The REPOS data for the example is shown in Figure 4.1, which is taken from p. C-II-150 of the AAR manual used for the stress-life analysis example (see Section 7.2.4 on p. C-II-144 in the AAR manual). No attempt is made to investigate combined effects with other loading modes.
2) No change in the sequence effect is expected during the conversion process from the load REPOS data to the strain REPOS data (which is not always true).
3) The base static stress due to dead and live loads is taken as 10,000 psi, as in the case of the stress-life example.
4) The stress/load conversion factor is assumed to be 10 psi/kips, which is chosen to yield the same total static and dynamic stress obtained for the stress-life analysis example.
5) The material is A-411 with a yield strength of 50,000 psi. The associated fatigue material properties are estimated to be:
   - o Fatigue strength coefficient, σ_f' = 120,000 psi
   - o Fatigue ductility coefficient, ϵ_f' = 0.5
   - o Fatigue strength exponent, b = -0.89
   - o Fatigue ductility exponent, c = -0.6
   - o Elastic modulus, E = 29,000,000 psi

REPOS FOR EXAMPLE 7.2.4.1.

C-II-150

ROAD ENVIRONMENT PERCENT OCCURRENCE SPECTRUM

VERTICAL G'S @ BOLSTER
100 TON AUTO PARTS BOX CAR

UP — DOWN | COLUMN NUMBER | UP — UP

MAXIMUM PEAK VERTICAL ACCELERATION G'S: 1.0, 0.9, 0.8, 0.7, 0.6, 0.5, 0.4, 0.3, 0.2, 0.1, 0, -0.1, -0.2, -0.3, -0.4, -0.5, -0.6, -0.7, -0.8, -0.9, -1.0

ROW NUMBER	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
1																				
2																				
3																				
4																				
5					0 0020															
6				0 0020	0 0040	0 0020	0 0040	0 0020												
7				0 0010	0 0030	0 0190	0 0260	0 0140	0 0060	0 0010										
8					0 0020	0 0170	0 0670	0 1740	0 1160	0 0180	0 0010									
9				0 0010	0 0010	0 0030	0 0470	0 4530	1 6290	0 7750	0 0130									
10						0 0010	0 0090	0 2520	5 4630	[illegible]	1 4150									
11							0 0010	0 0070	1 3630	49 8420										
12							0 0010		0 0010											
13				0 0010			0 0010													
14																				
15																				
16																				
17																				
18																				
19																				
20																				

ROAD TEST OF: 100 TON AUTO PART BOX CAR (CUSHION UNDERFRAME)
CROSS RAIL WT.: 235,500 lbs. TEST DATE: ______
FROM-TO ______
ON ______ RR
SIZE TRAIN: ______
TEST CAR LOCATED: ______
MAX MPH 60-70 AVG. MPH 32 3
RECORDED MILEAGE 157 7
RECORDING TIME 5 0 Hrs

MINIMUM PEAK VERTICAL ACCELERATION G'S: -1.0, -0.9, -0.8, -0.7, -0.6, -0.5, -0.4, -0.3, -0.2, -0.1, 0, 0.1, 0.2, 0.3, 0.4, 0.5, 0.6, 0.7, 0.8, 0.9, 1.0

DOWN — DOWN

SPECTRUM TOTAL 99 9489
POSITIVE FORCE IS UP
NEGATIVE FORCE IS DOWN

TOTAL OCCURRENCES 111 856
AVG OCCURRENCES/MILE 709 322
AVG CYCLES/MILE 354 661

Figure 4.1 REPOS DATA FOR EXAMPLE PROBLEM

6) The weld detail selected for the analysis example is a beam reinforced with a partial cover plate applied with longitudinal and transverse fillet welds, as shown in case number 3.1.5 of Section 7.4, p. C-II-160. It is assumed to have Fatigue notch factor, $k_f = 3.0$.
7) The residual stress due to the welding process is assumed to be at the yield level of 50,000 psi.

Based on these above data, the computer program, which will be described in detail later, solves the strain-life equation to determine the estimated fatigue life for the component in question. The input and output formats are presented in Appendix D. The result shown in p. D-6 indicates that:

Fatigue life by means of the strain-life method is $\underline{49.72 \times 10^8}$ cycles.

The result of the AAR example, shown on p. C-II-148 (or p. D-8) shows that:

Fatigue life by means of stress-life method is $\underline{13.68 \times 10^8}$ cycles.

The above comparison of results suggests that the strain-life analysis predicts a fatigue life longer than that obtained by the stress-life method. However, it should be noted that such an outcome is not generally true, but depends on the set of input data used. The sensitivity of input data to the fatigue life output will be discussed later, in Section 4.4.

4.3 Brief Description of Strain-Life Equation FORTRAN Program

The strain-life equation FORTRAN program solves the strain-life equation (3.8), for a series of input minimum and maximum loads and frequencies of occurrence, such as AAR REPOS data, to evaluate the fatigue life in cycles.

The maximum and minimum stresses in a cycle are calculated from the maximum and minimum loads using

$$S_{max} = S_s + P_{max} \left(\frac{S}{P}\right) + S_{res} \tag{4.1}$$

$$S_{min} = S_s + P_{min} \left(\frac{S}{P}\right) + S_{res}$$

where

P_{max}, P_{min} = maximum and minimum loads

S_s = static base stress

S_{res} = residual stress

$\frac{S}{P}$ = stress-to-load ratio

$= \frac{S}{P}+$ for load $P > 0$

$= \frac{S}{P}-$ for load $P < 0$

S_{max}, S_{min} = maximum and minimum stresses

For each P_{max} - P_{min} pair, S_{max} and S_{min} are evaluated, and the minimal stress range ΔS and mean stress σ_o are calculated

$$\Delta S = S_{max} - S_{min}$$

$$\sigma_o = \tfrac{1}{2} (S_{max} + S_{min}) \tag{4.2}$$

Equation (4.8) is used to determine the corresponding life in cycles N_f. The damage per cycle due to a particular $P_{max} - P_{min}$ pair is

$$D = \left(\frac{f}{100}\right) / N_f \tag{4.3}$$

Where f is the frequency of the pair in percent. Total damage for all pairs is

$$D_{tot} = \sum_i D_i \tag{4.4}$$

and the overall effective life in cycles is

$$L = \frac{1}{D_{tot}} \tag{4.5}$$

4.3.1 Input Format

Input for the program is prompted by titles in the input file. An example is shown on page D-3. All input numbers are in free format and thus they need not be lined up to particular columns. The input lines are as follows:

Prompt line

One line with the number of cases, N - i.e., the number of $P_{max} - P_{min}$ pairs.

Prompt line

N lines, each containing an integer (for convenience in counting cases), the maximum load P_{max} in kips, the minimum load P_{min} in kips, and the frequency of occurrence in percent (i.e., total for all cases should be 100).

Prompt line

One line with

Static base stress, S_s, in psi

Residual stress, S_{res}, in psi

S/p ratio for positive load, in psi/kip

S/p ratio for negative load, in psi/kip

Fatigue strength coefficient, σ_f' , in psi

Fatigue ductility coefficient, ε_f'

Prompt line

One line with

Fatigue strength exponent, b

Fatigue ductility exponent, c

Young's modulus, E, in psi

Fatigue notch factor, K_f

4.3.2 Solution Procedures

For solution, equation (3.8) is recast into the following form

$$A = (2N_f)^{2b} [1 + B (2N_f)^{c-b}]$$

where

$$A = \frac{1}{4} \left(\frac{K_f \Delta S}{\sigma_f' - \sigma_o} \right)^2$$

$$B = \frac{E\varepsilon_f'}{\sigma_f'} \left(\frac{\sigma_f' - \sigma_o}{\sigma_f'} \right)^{\frac{c}{b} - 1}$$

The first estimate of $2N_f$ is

$$(2N_f)_1 = A^{\frac{1}{2b}}$$

The second estimate is

$$(2N_f)_2 = \left[\frac{A}{1+B\ (2N_f)_1^{\ c-b}} \right]^{\frac{1}{2b}}$$

and succeeding estimates are computed from

$$(2N_f)_{i+1} = \left[\frac{A}{1+B\ (2N_f)_i^{\ c-b}} \right]^{\frac{1}{2b}}$$

This iterative procedure has converged when

$$[\ (2N_f)_{i+1} - (2N_f)_i]\ /\ (2N_f)_{i+1}$$

is less than the acceptable error which has been fixed at 0.000001. If a limit number of iterations (set at 100) has been made without satisfactory convergence, an error message

NO CONVERGENCE FOR ITEM NUMBER ________

ERROR FRACTION = __________

is printed. The 'error fraction' is the convergence test value stated above.

4.3.3 Output Format

Sample program output is shown on pages D-4 and D-5. The input is echoed, followed by a listing of the P_{max} - P_{min} pairs together with the cycle fraction (the input frequency divided by 100), the mean stress σ_o, the failure life in cycles N_f for the input pair, and the damage D for the input pair. Following this list is the total damage D_{tot} and the overall effective life in cycles L.

The program, as listed here, is limited to 100 input P_{max} - P_{min} pairs. This limit can readily be extended by obvious means. Similar the acceptable error and the iteration limit can be altered as desired.

4.4 Parametric Sensitivity Study

The results of the foregoing example could have misled someone to believe that the strain-life methods would yield a fatigue life longer than that which could have been obtained from the stress-life method. To assist in understanding the sensitivity of material characteristics to the strain-life results, a parametric study of various influential factors on both the stress-life and strain-life results was conducted. In general, the fatigue life/material parameter relationship can be expressed in the following forms:

1) Stress-Life Analysis

$$N_f = f\,(B, m, K, S_s, \Delta S) \qquad (4.6)$$

where
- B = MGD Y-Intercept
- m = MGD slope
- K = S-N curve slope
- S_s = Static base stress
- ΔS = Cyclic stress range

2) <u>Strain-Life Analysis</u>

$$N_f = f\,(\sigma_f', \varepsilon_f', b, c, S_s, S_R, \Delta e, K_f, E) \qquad (4.7)$$

where
- σ_f' = Fatigue strength coefficient
- ε_f' = Fatigue ductility coefficients
- b = Fatigue strength exponent
- c = Fatigue ductility exponent
- S_s = Static base stress
- S_R = Residual stress
- Δe = Cyclic strain range
- K_f = Fatigue notch factor
- E = Modules of elasticity

Obviously, the influence of some material parameters are extremely important, while others are not. To assess their relative significance, this sensitivity analysis was conducted with reference to the base data used for the previous working example.

It should be noted that the results of this analysis, as shown in the following figures, must not be regarded as indicative of the superiority of one method over the other. Rather, the comparisons of two methods should be strictly applied to this particular setting, e.g., the welding detail investigated, the material properties variations used and the loading condition selected. Nevertheless, the following general observations can be made:

1) The strain-life procedures are more sensitive, than the stress-life approach, to changes in the material fatigue properties.
2) With the exception of the fatigue ductility coefficient, ε_f', and the fatigue ductility exponent, c, a slight variation in

any other parameters could result in a great difference in the fatigue life predicted by means of the strain-life method.

3) The residual stresses play an extremely important role in the fatigue life of materials and should be kept as minimal as possible.

4) Poor workmanship, represented by the high values for the fatigue notch factor, would shorten the fatigue life. A quality control program must be required to assure that high standards of workmanship are attained.

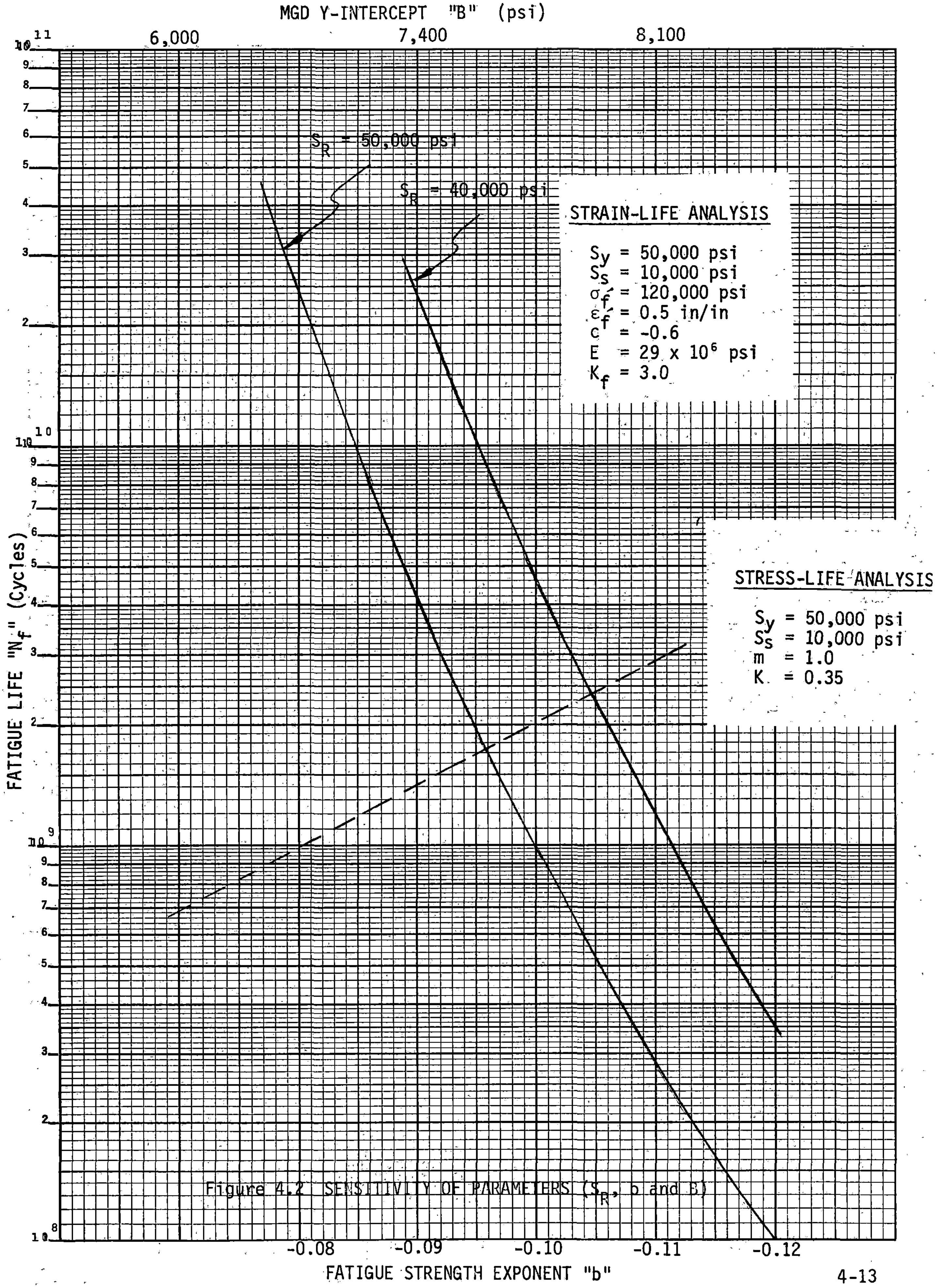


Figure 4.2 SENSITIVITY OF PARAMETERS (S_R, b and B)

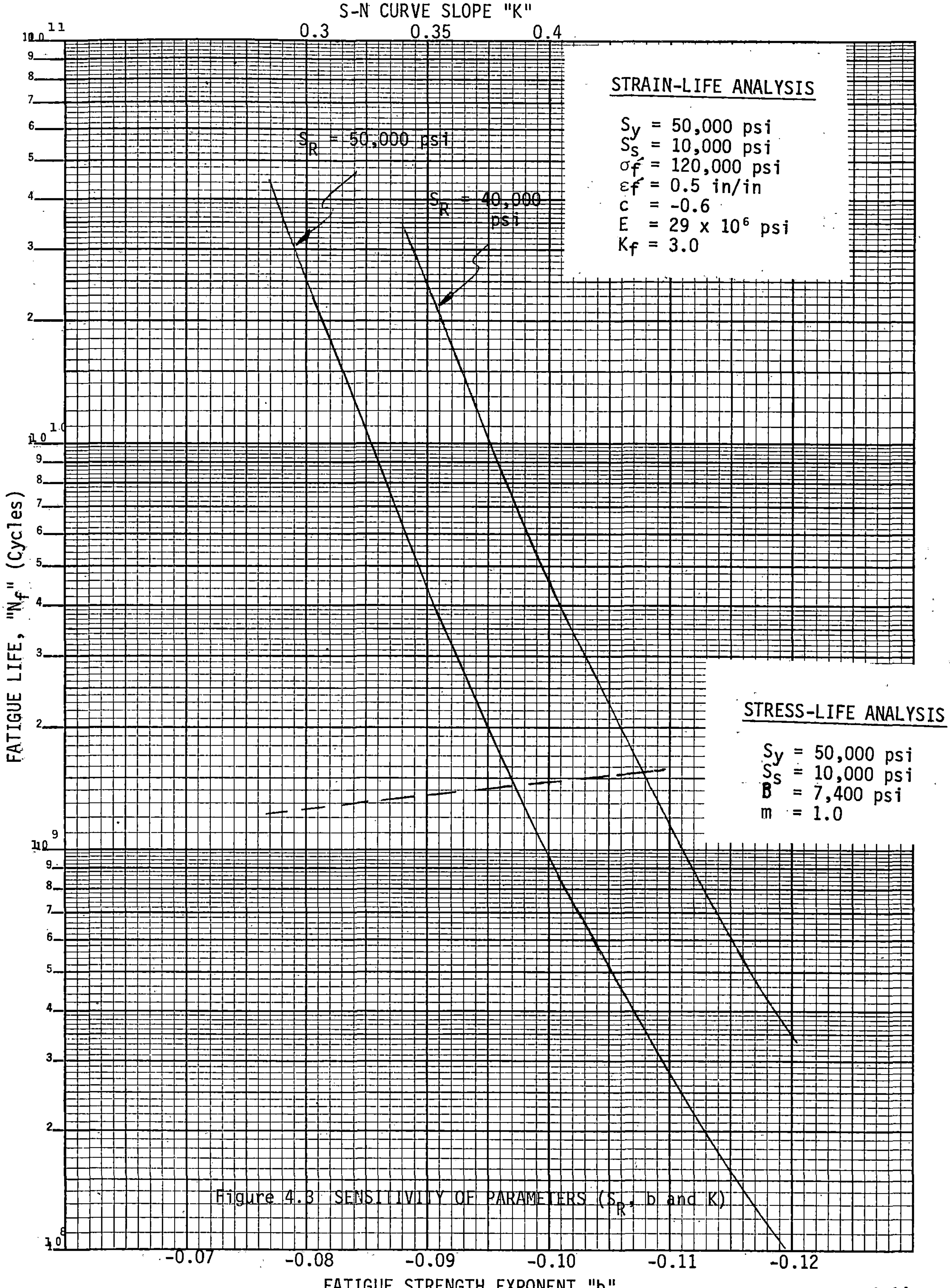


Figure 4.3 SENSITIVITY OF PARAMETERS (S_R, b and K)

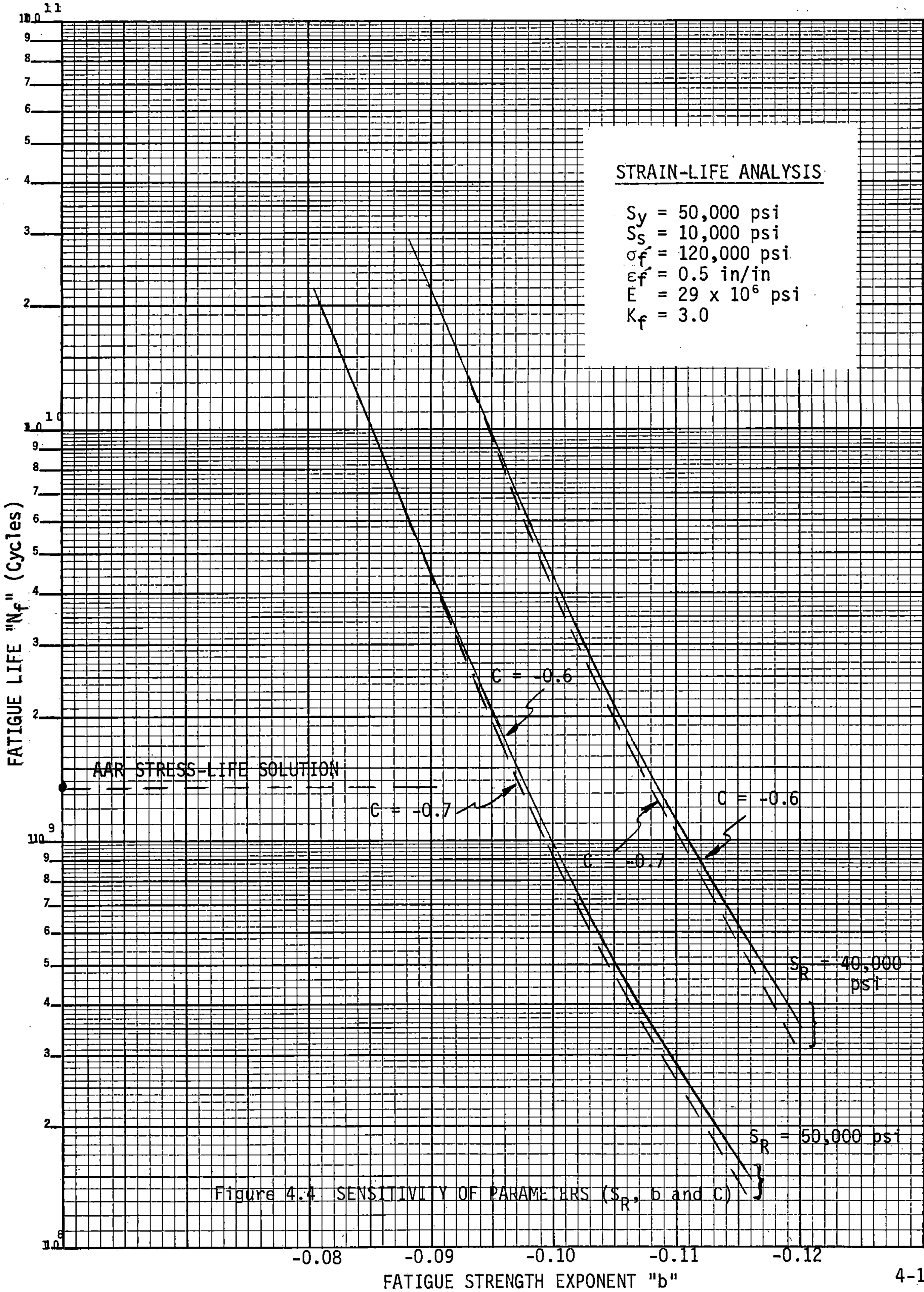


Figure 4.4 SENSITIVITY OF PARAMETERS (S_R, b and C)

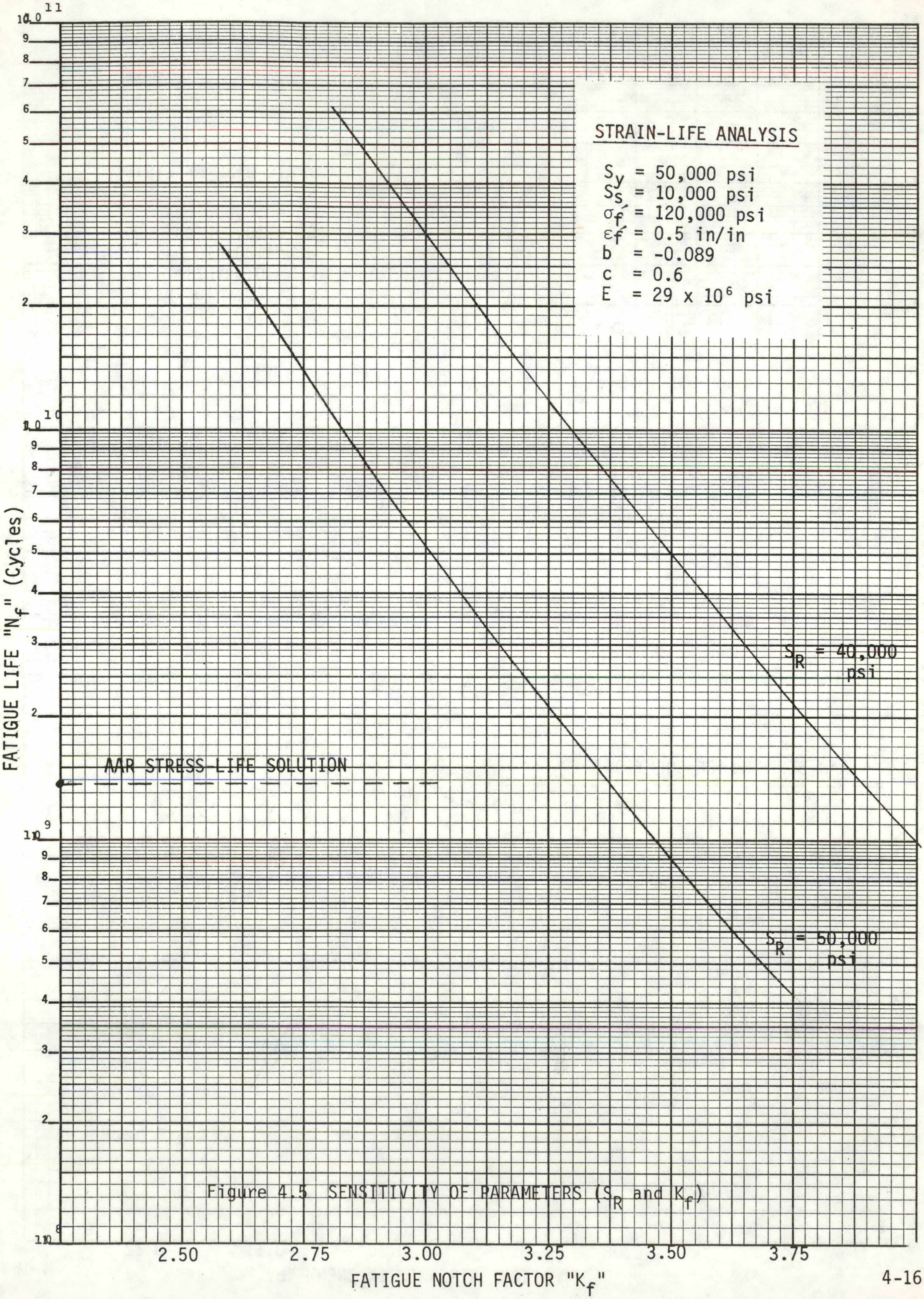


Figure 4.5 SENSITIVITY OF PARAMETERS (S_R and K_f)

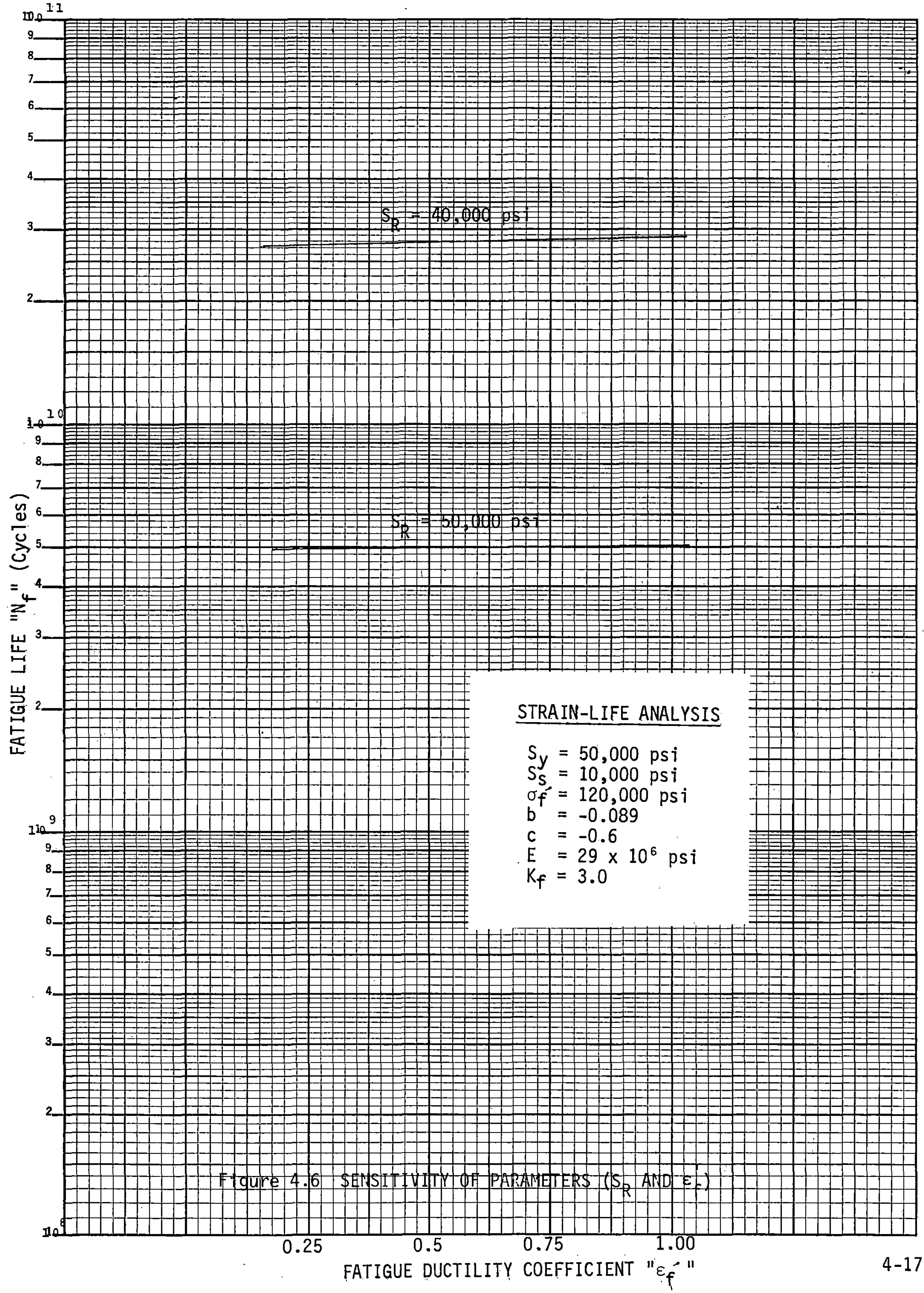


Figure 4.6 SENSITIVITY OF PARAMETERS (S_R AND ε_f')

Section 5

DISCUSSION AND RECOMMENDATIONS

5.1 Discussion

5.1.1 Application of Modern Fatigue Technology to the Railroad Industry

As presented in the previous discussion, trends in the design of mechanical or structural components in transportation equipment to prevent fatigue failure prior to their expected service life have led to more sophisticated procedures, both in analysis/design and in testing. Therefore, it is not surprising to observe different emphasis expressed and pursued by various groups of researchers and engineers, as indicated in Section 2, who perform design/analysis and testing of components or full-scale structures.

It was also indicated in Section 2 that there are some problems associated with the use of fracture mechanics criteria in the design process for railroad cars. One problem deals with the definition of crack size, for example, applied to actual material defects, such as bad welds. Another problem is that a periodic inspection program is definitely required in order to monitor the growth of cracks before they reach the critical stage that would lead to a total failure.

On the other hand, the continuum mechanics considerations, such as stress-life or strain-life approaches, necesitate the collection of strain-time history data at the local level so that a proper account of the cyclic damage due to plastic deformation can be registered through a reasonable cycle counting process, such as the rainflow counting method.

However, most engineers/designers in the railroad industry express their preference for the component calibration approach (see the AAR Manual), because the strain data at critical locations, such as at the tips of welds, are not easily obtainable. This method permits a conversion of applied loads to local notch strains or stresses if all sequence effects are identical and can be retained. A typical application example of such an approach is given in Appendix C. It is noted that one should not attempt to perform the load/stress conversion (the current AAR stress-life methods) when the material experiences plastic strains at the notch under certain load conditions that alter the sequence effect. The resulting stress spectra will not be identical to that of the loading even though both are subjected to the same cycle counting process. On the other hand, the strain-load conversion used by the strain-life methods allows for the presence of plastic strains through the cyclic material characterization.

5.1.2 Random Variable Effects

In general, the method of approach employed in the analysis and design process of structures depends upon what kind of information is available to the designer about the environment input and the material data. The analysis requires the environment input including the force (load) definition, the geometric configuration, and the physical constraints, whereas the design process is primarily dictated by the material characteristics including various strengths of materials and their degradation, failure behavior, and moreover by the predetermined safe performance requirement such as the service life.

It is generally recognized that most engineering problems, including those in structural design, are non-deterministic. Structures are often designed in spite of insufficient information with regard to the environment and the response characteristics of the materials selected for use, and an inadequate methodology for analysis and design. Structures are, furthermore, (unintentionally) built with a certain degree of initial imperfection in shape and with flaws. The material strength deteriorates randomly with time as well in the case of corrosion and fatigue.

In short, problems of structural design must be resolved within the context of uncertainties. As a consequence, risk consideration involving the probability of adverse events (failure or malfunction) becomes virtually inevitable. Indeed, it is because of such uncertainties that factors of safety (or equivalent load factors) are traditionally required during the design process.

Moeover, there is considerable controversy over whether the deterministic safety factor should include adequate allowances for errors in the method of analysis, for insufficient knowledge of the environment expected, and for incomplete data about the material used.

5.1.2.1 Variations of Environmental Input

In general, environmental load spectra are given for specific types of cars by the AAR in relation to car motions (for instance, in terms of the longitudinal coupler force, or vertical acceleration, in g's), as if they were acting independently of each other. The stress or strain spectra resulting from the combination of such independent load spectra will definitely be substantially different from the spectra obtained from direct conversion of the load-time history data. In fact, the AAR approach completely ignores the time-phase relationship to local stress or strain effects. Consequently, the fatigue life prediction formula given by the AAR (p. C-II-141) under multiaxial motion is a questionable approach. As a matter of fact, serious consideration should be given to efforts seeking to look into design practices employed by construction engineers for treating comparable problems related to the design of buildings and nuclear power plants for multiaxial seismic input.

5.1.2.2 Variations of Material Characteristics

For the case of design involving long life service consideration, the fatigue performance of structural components may be significantly altered due to the fabrication process, such as welding, even if the properties of materials could be properly characterized by carefully conducted laboratory study. For instance, no laboratory welds can be repeatedly produced in field operations. Therefore, it is expected that some variations in the material characteristics will be observed from one car to another.

In order to determine the impact of material variations on the service life estimation, a sensitivity study of material parameters, influential to fatigue life prediction by means of both stress-life and strain-life approaches, was conducted. The results are presented in Appendix D. It can generally be observed from these results that:

1) The fatigue life prediction by both analysis methods is very sensitive to changes in the material parameters.
2) The strain-life approach is more prone to the scatter of the material data than its counter part, the stress-life approach.
3) The stress-life method gives no consideration to the quality control of car construction, such as the quality of welded

joints, whereas the strain-life analysis uses the fatigue notch factor, K_f, to characterize the weld detail, including defects.

5.1.3 Multiaxial Stress Fatigue

It is a well known fact that the results of uniaxial tests have been utilized to predict the failure of materials under monotonic static loading. Some useful failure criteria include:

1) Maximum stress theory,
2) Maximum strain theory,
3) Maximum shear stress theory,
4) Maximum strain energy theory,
5) Distortion energy theory, etc.

However, the application of such criteria to the prediction of fatigue failure of railroad equipment under multiaxial states of stress has not been established to be reasonable. The reason is that railroad cars experience a state of multiaxial stress which constantly changes with time on the roads. Of course, both the magnitude and direction of the principal stresses change with time as well.

Therefore, the real issue confronting engineers is the use of test data obtained with the application of uniaxial constant amplitude loading to predict the real life situation on the roads, when the cars are in fact continuously subjected to random multiaxial stress fatigue. This is another area, in addition to multiaxial motion fatigue, which requires a major effort to reach some reasonable resolution.

In summary, reliability is the probability that a given product will perform as anticipated for a certain service life. Therefore, the real and significant role of probability in structural engineering lies in its logical framework for uncertainty analysis and its provision of a quantitative basis for risk and safety assessment. Unfortunately, at present, railroad engineers cannot effectively deal with such design problems, due to the lack of fundamental information.

Undoubtedly, there are many very important areas of research to develop the needed basic fatigue design technology, including proper treatment of environmental data, sufficient collection of material information related to fatigue behavior, and adequate development of fatigue analysis/design methodology.

In conclusion, the strain-life analysis procedure has been reviewed and evaluated. It has been found to be a viable tool that can be used by the railroad industry without foreseeable difficulty. The method should yield results more reasonable than those obtained by means of the stress-based approach which does not account for local plastic strain, the real cause of fatigue damage. The potential pitfall is an attempt to oversimplify the procedure when treating problems involving multiaxial motion as well as multiaxial stress fatigue.

5.2 Recommendations

The efforts to improve the reliability and safety of railroad cars and associated equipment, especially related to fatigue performance, should be focused on one or several of the following areas:

1) Improvement in fatigue (not monotonic) properties of materials,
2) Modification of component parts and fabrication processes, including the methods of their attachment, to reduce stress (strain) concentration,
3) Reduction or relief of residual stresses due to fabricating processes,
4) Improvement of the fatigue analysis methodology, including variable-amplitude spectra representative of actual environment conditions, especially the technique to be used for load/strain (or load/stress) conversions under simultaneous application of multiaxial motion,
5) Development of logical fatigue design procedures, particularly for equipment subjected to multiaxial stress fatigue.

Combination of efforts in these areas is definitely expected to yield products that are more reliable to the railroads and ultimately satisfy public safety requirements.

5.2 Recommendations

Strain-life analysis has been in use in many industries, including the railroad industry, as a tool to deal with fatigue problems. While this approach has enjoyed wide acceptance with engineers and designers, a major drawback continues to persist with the method, when used for design purposes. The resulting stress spectra differ from the loading spectra, after being subjected to the same cycle counting procedure (typically the rainflow counting method). This is especially evident, for instance, when railroad cars are subjected to start-up loads, impacts in switch yards, etc., that generate local plastic cyclic strains at critical notches, such as at welds. These local plastic strains are the real cause of fatigue damage.

The preceding appraisal, review and discussion concerning strain-life fatigue analysis indicates that strain-life methods overcome the above difficulties and can be readily adopted for use by the railroad industry. It is recommended that engineers be encouraged to utilize these methods in designing railroad cars and associated equipment to prevent premature fatigue failure.

For further enhancement in strain-life fatigue analysis, research in the following areas is strongly recommended:

1. Procedures for dealing with simultaneous application of multiaxial motion.
2. Methods of analysis to assess fatigue damage due to the presence of multiaxial stress state variations resulting from random road environment input.

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Section 6

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APPENDIX A

Letter to Academic Community, Industrial Research Organizations and Government Agencies

Note: * Indicates response received.

GATX

GARD, INC.

7449 NORTH NATCHEZ AVE
NILES, IL 60648
312 647-9000

July 21, 1983

Attention:

Subject: Theoretical Background & Application Experience Related to Strain-Life Fatigue Analysis

Dear

Currently GARD is working for the Association of American Railroads (AAR) in conjunction with the Track/Train Dynamics (TTD) Fatigue Task Force to investigate the application feasibility of the strain-life analysis approach to the fatigue design of railroad cars and related equipment.

One of our major tasks is to invite the leading experts in the field of fatigue technology to offer their views regarding its theoretical background and practical applications. Your comments are regarded as one of the major factors in determining the utilization feasibility of the strain-life methodology by the railroad industry.

Some specific examples of pertinent commentary needed include, but are not limited to, the following:

1) Your opinion concerning the use of strain-life approach, with pertinent comments regarding its theoretical validity, assumptions, limitations, potential benefits and pitfalls.
2) Comparison of strain-life vs. stress-life analysis approach in terms of material characteristics, loading definition, analysis methodology, and design objectives including service life requirements.
3) Application experience, if any.
4) Information related to test design, including test procedures, load definition, material selection, failure observation, instrumentation, environmental description, and results.

We, GARD and AAR, hope that you will respond to this request by furnishing answers to the above questions, as much as possible. Your cooperation will be greatly appreciated and your contribution will be acknowledged.

Sincerely yours,

J. C. Shang

J. C. Shang, Ph.D.

cc: Mr. R. A. Allen (AAR); K. L. Hawthorne (AAR)
Dr. V. K. Garg (AAR)
Mr. R. D. Sims (TTD Fatigue Task Force)
Mr. S. Richmond (TTD Fatigue Task Force)

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APPENDIX B

Letter to Members of AAR Car Construction Committee

Note: * Indicates reponse received.

GATX

GARD, INC.

7449 NORTH NATCHEZ AVE
NILES, IL60648
312 647-9000

July 21, 1983

Attention:

Subject: Fatigue Failures Experienced by Railroad Industry

Dear

Presently GARD is working for the Association of American Railroads (AAR) in conjunction with the Track/Train Dynamics Fatigue Task Force to investigate the application feasibility of strain-life techniques to the fatigue design of railroad cars and equipment.

There are several factors which will ultimately determine the utilization feasibility of the strain-life methodology to railroad industry applications. One of these influential factors is past experience of the industry as a whole, with fatigue failures. Some specific examples of pertinent information are:

a) General construction and detailed description (photograph if possible) of fatigue failures experienced, repeated failure if any.

b) Vehicle road data and service life prior to failure.

c) Material characteristics (monotonic, cyclic or both) data.

d) Outline of fatigue design approach used.

e) Methods of stress analysis (including finite element methods).

It will be greatly appreciated if you can share your experience by responding to this request and providing answers to the above questions as much as possible. This information will be most helpful to determine the future use of such analysis methods in the railroad industry. Of course, information you provide will be treated with the utmost confidentiality.

Sincerely yours,

J. C. Shang

J. C. Shang, Ph.D

cc: Mr. R. A. Allen (AAR); K. L. Hawthorne (AAR)
Dr. V. K. Garg (AAR)
Mr. R. D. Sims (TTD Fatigue Task Force)
Mr. S. Richmond (TTD Fatigue Task Force)

JCS/cb

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APPENDIX C

Fundamentals of Strain-Life Analysis

Fundamentals of Strain-Life Analysis

C.1 Theoretical Justification

Beacuse the nominal stresses in most structures are elastic, the zone of plasticly deformed metal in the vicinity of a stress concentration is surrounded by an elastic stress field. In other words, the material behavior of the plastic zone is constrained by the elastic displacements of the surrounding elastic stress field. Hence, even though the structure is stress-controlled, the localized plastic zone behavior is strain-controlled. Consequently, to predict the effects of stress concentration on the fatigue crack initiation behavior of structures, it is considered appropriate that the fatigue behavior of the localized plastic zone should be simulated by smooth specimen tests under strain-controlled conditions (Figure C.1).

Four major steps are required to perform strain-life analysis:

1) Understanding of material behavior under cyclic loading,
2) Development of basic strain-fatigue life relationships,
3) Determination of the relationship between the applied loads and resulting strains at the local level,
4) Evaluation of fatigue damage incurred and prediction of fatigue life.

C.2 Material Behavior Considerations

The lack of suitable material information under cyclic loading is one of the most serious deficiencies in fatigue analysis. The normally reported material fatigue (endurance) limits are often of little value in modern structural design when the geometric complexity and long-term random loading

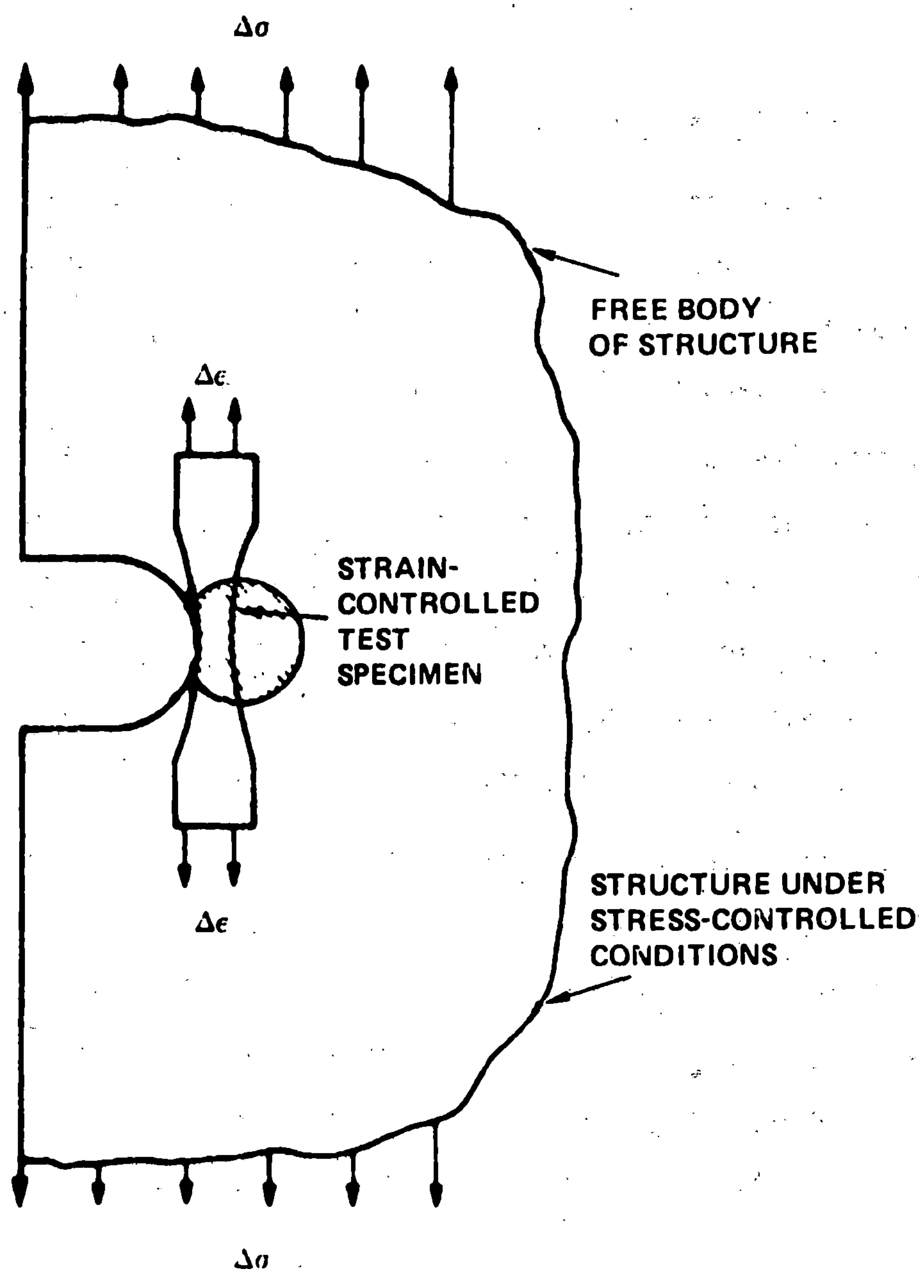


Figure C.1 STRAIN-CONTROLLED TEST SPECIMEN SIMULATION FOR STRESS CONCENTRATIONS IN STRUCTURES are of the major concern. Furthermore, it has been reported that such endurance limits may be greatly altered or eliminated by periodic overstrains (start-up peaks or overloads) of the kind that often occur in service (Refs. C.1 and C.2).

It has been observed that the mechanical properties of metals characterized by stress-strain response, when subjected to repeated plastic deformation occurring at a microscopic level, can be drastically different from the monotonic behavior.

Many steels subjected to constant amplitude cyclic loading usually exhibit an initial transient behavior, but reach an essentially cyclically stabilized stage that corresponds to a constant hysteresis loop.

C.2.1 Transient Stress-Strain Behavior

Depending upon the initial state (quenched-and-tempered, normalized, annealed, cold-worked) and the test condition (load or displacement control), metals may undergo a process of

- o Cyclic creeping,
- o Cyclic hardening,
- o Cyclic softening,
- o Cyclic relaxation,
- o Relatively stable behavior, or
- o Mixed behavior.

Schematic illustrations of typical transient phenomena are shown in Figure C.2. It is noted that the cases (a), (b), and (c) display the material behavior which could be observed under strain (displacement) control tests, whereas case (d) could be seen under stress (load) control tests. The

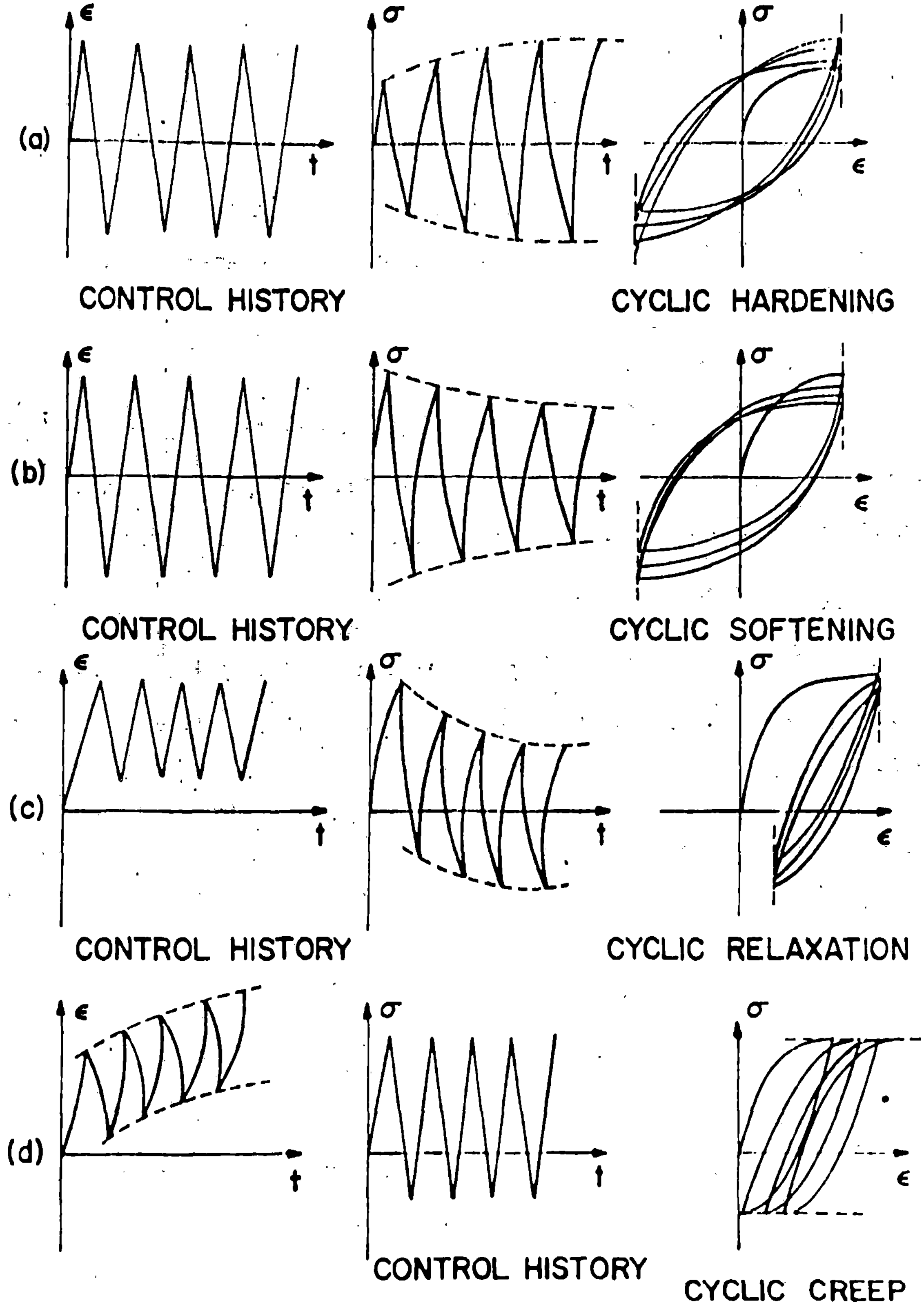


Figure C.2 SCHEMATIC ILLUSTRATIONS OF CYCLIC TRANSIENT PHENOMENA cyclically stable behavior implies that the cyclic stress-strain response is identical to that of the monotonic stress-strain process. In the mixed mode, either cyclic softening or hardening could occur depending on stress (or strain) amplitude.

C.2.2 Steady-State Stress-Strain Behavior

After the transient period of adjustment, essentially the same size and shape of hyteresis loop will be produced cycle after cycle. Such a steady-state condition, represented by a steady-state stress-strain hysteresis loop, is usually achieved in about 20 to 40% of the total fatigue life in most hardening or softening materials. A typical steady-state hysteresis loop is depicted in Figure C.3. These characteristics are established by subjecting a smooth specimen to fully reversed cyclic strain of constant amplitude.

For a completely reversed hyteresis loop, under a strain-controlled condition with zero mean strain,

$$\Delta\varepsilon = 2\,\varepsilon_a \tag{C.1}$$

and

$$\Delta\varepsilon = 2\,\sigma_a \tag{C.2}$$

where $\Delta\varepsilon$ = Total strain range

$\Delta\sigma$ = Total stress range

ε_a = Strain amplitude

σ_a = Stress amplitude

Furthermore,

$$\varepsilon_a = \frac{\Delta\varepsilon}{2} = \frac{\Delta\varepsilon_e}{2} + \frac{\Delta\varepsilon_p}{2} = \frac{\Delta\sigma}{2E} + \frac{\Delta\varepsilon_p}{2} \tag{C.3}$$

or

$$\varepsilon_a = \frac{\sigma_a}{E} + \varepsilon_p \tag{C.4}$$

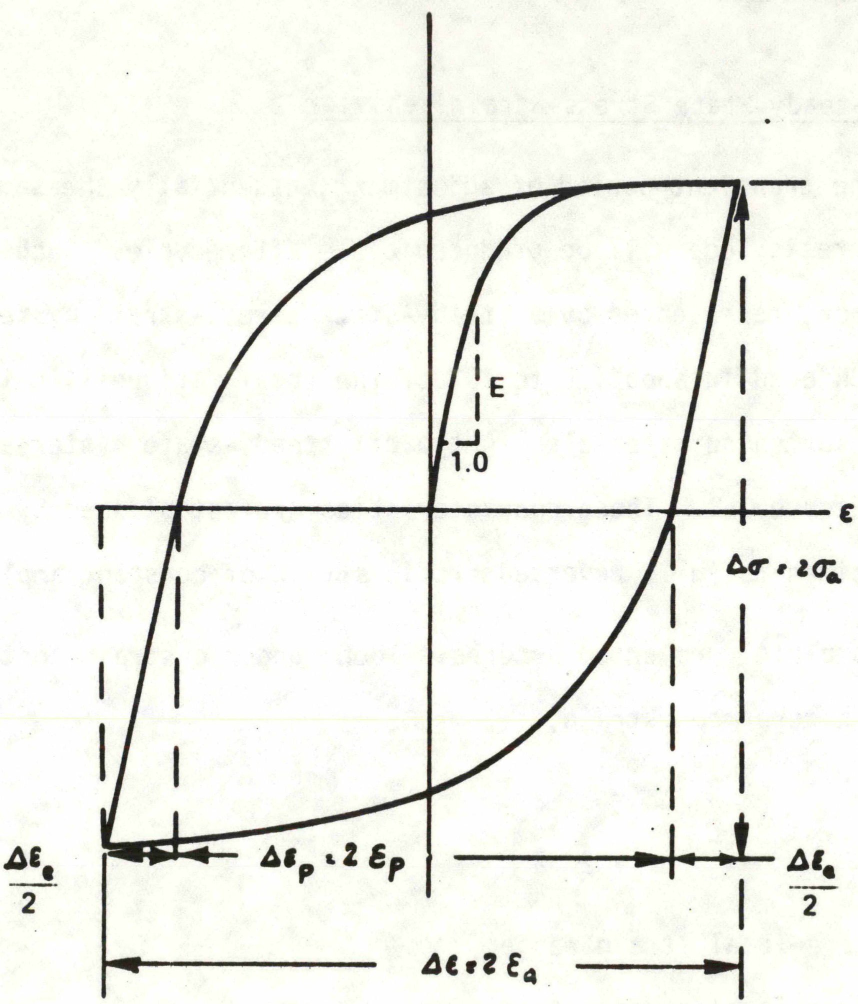


Figure C.3 SCHEMATIC OF A STEADY-STATE STRESS-STRAIN HYSTERESIS LOOP

C.2.3 Cyclic Stress-Strain Curves

The stabilized steady-state loops can be obtained for various strain ranges. Enough specimens can be cycled at different strain ranges to provide stabilized stress-strain behavior over a sufficient range to fit a function in the form of Eq. (C.3). Subsequently, the cyclic stress-strain curve can be constructed by connecting the tips of these loops, as shown in Figure C.4.

In a manner similar to the monotonic stress-strain relation, a power-law function between the true stress and the plastic strain may be established (Figure C.5) in the form:

$$\sigma_a = K' (\varepsilon_p)^{n'} \tag{C.5}$$

where σ_a = Steady-state stress amplitude

ε_p = Plastic strain component of strain amplitude

K' = Cyclic strength coefficient, measured at $\varepsilon_p = 1$

and n' = Cyclic strain hardening exponent (slope)

By combining Eq. (C.4) with Eq. (C.5) one obtains

$$\varepsilon_a = \frac{\sigma_a}{E} + \left(\frac{\sigma_a}{K'}\right)^{1/n'} \tag{C.6}$$

Eq. (C.6) can also be rewritten in terms of ranges:

$$\frac{\Delta\varepsilon}{2} = \frac{\Delta\sigma}{2E} + \left(\frac{\Delta\sigma}{2K'}\right)^{1/n'} \tag{C.7}$$

where n' = 0.10 - 0.20, with an average value very close to 0.15. (C.8)

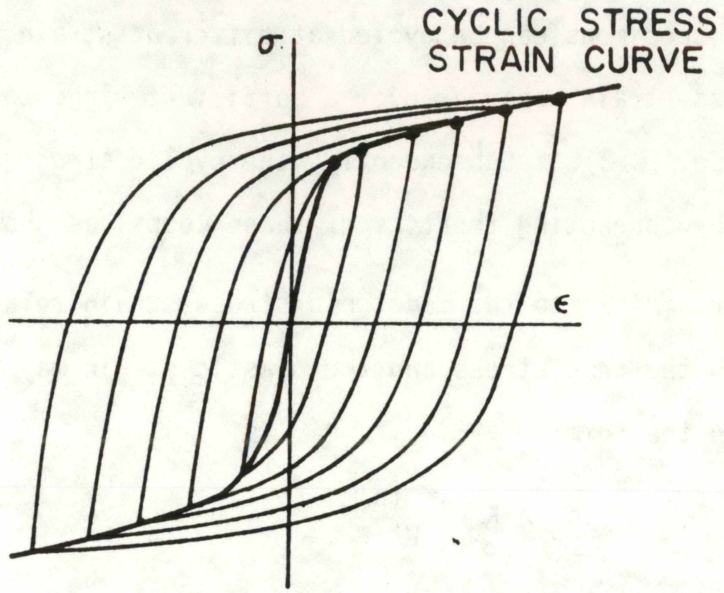


Figure C.4 STEADY-STATE STRESS-STRAIN LOOPS AND CONSTRUCTION OF CYCLIC STRESS-STRAIN CURVE

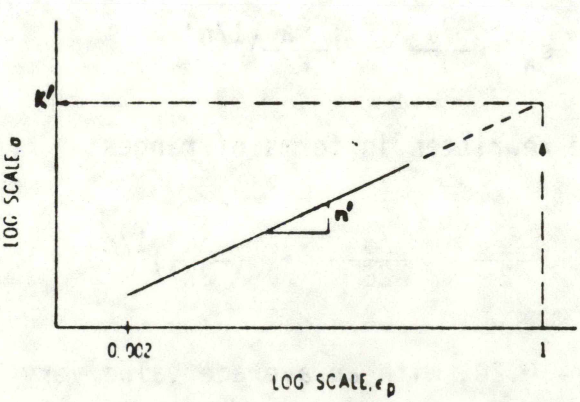


Figure C.5 TRUE STRESS VERSUS PLASTIC STRAIN FOR CYCLIC RESPONSE (LOG-LOG COORDINATES)

The material cyclic properties can often be compared with the monotonic properties in order to qualitatively assess the cyclically induced changes in material behavior. As displayed in Figure C.6 for example, lower strength alloys are found to exhibit either stable or cyclic hardening behavior while a trend towards cyclic softening develops with increasing strength level. Aluminum alloys show a stable to hardening tendency while the titanium alloys all soften. Copper and nickel alloys may harden or soften depending on condition. Steels display a range of responses from cyclic stability or hardening in low carbon grades to marked softening in the higher strength, heat treatable grades. Of significance is the observation that, at the highest strength levels, cyclic stability or even cyclic hardening can be achieved in ferrous systems through appropriate processing. Finally, austenitic stainless steels are seen to harden dramatically as a result of a deformation-induced phase transformation.

Figure C.6 also illustrates the fallacy of utilizing the monotonic yield strength, rather than the cyclic stress-strain curve, as the reference in designating structures to resist fatigue loading.

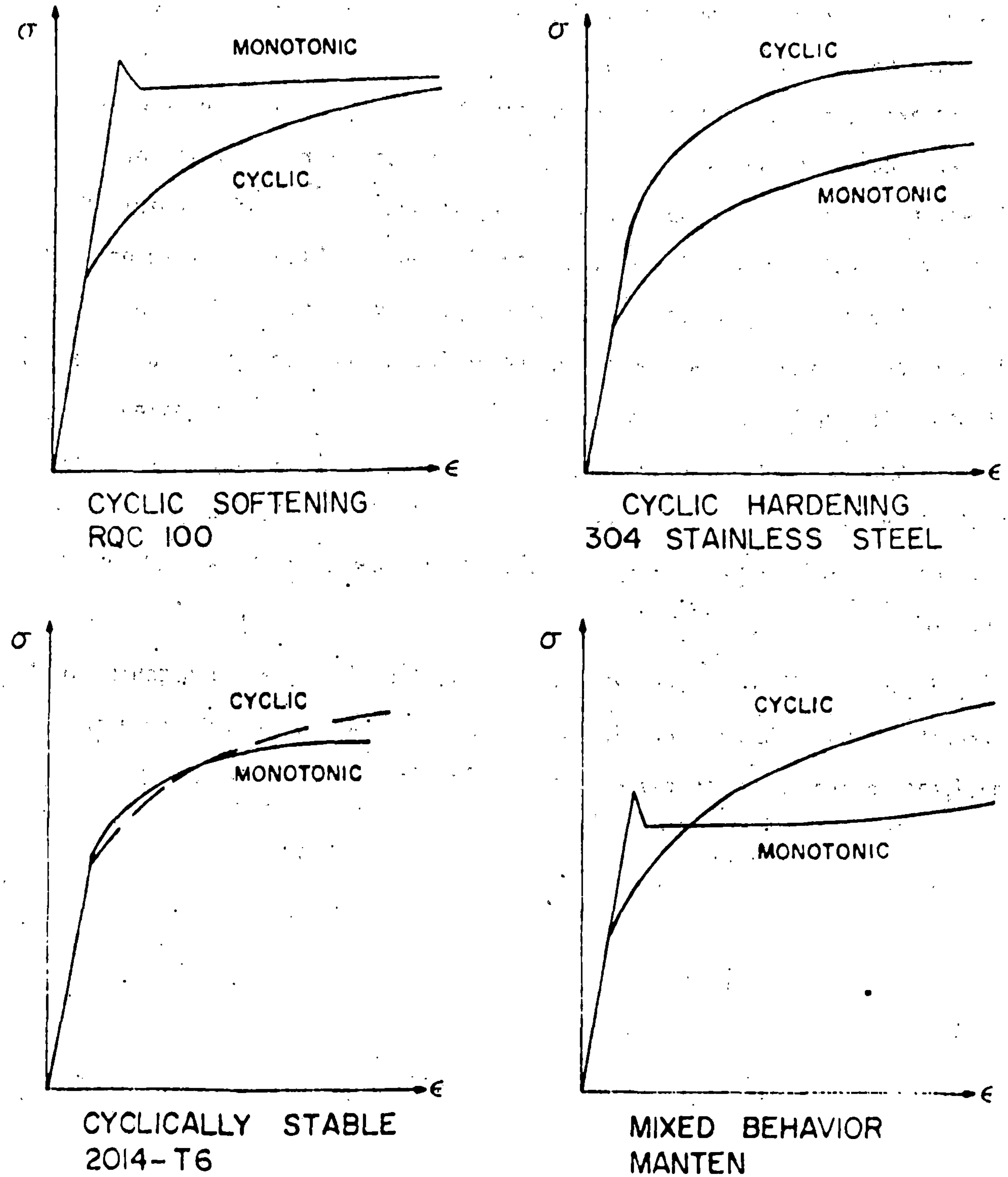


Figure C.6 CYCLIC AND MONOTONIC STRESS-STRAIN BEHAVIOR OF METALS

C.3 Basic Strain-Fatigue Life Relationships

C.3.1 Development of Strain-Life Equations

Ever since Wohler's work on railroad axles subjected to rotating-bending stresses, fatigue data have been presented in the form of S-log(N) curves. Around 1900, Basquin showed that the S-log(N) plot could be linearized with log-log coordinates (Figure C.7) and thereby established the exponential law of fatigue life, which holds if true stress amplitudes are used instead of engineering stress. Thus, the fatigue life can be related to true stresses by:

$$\sigma_a = \frac{\Delta\sigma}{2} = \sigma_f' \, (2\,N_f)^b \tag{C.9}$$

where $\sigma_a = \frac{\Delta\sigma}{2}$ = True stress amplitude in zero mean constant amplitude test.

σ_f' = Fatigue strength (Basquin's) coefficient, intercept at $(2\,N_f) = 1$

$(2\,N_f)$ = Reversals to failure (1 cycle = 2 reversals)

b = Fatigue strength (Basquin's) exponent (slope)

σ_f' and b are fatigue properties of materials. For many metals, σ_f' is approximately equal to σ_f (monotonic true fracture strength). The value for b varies between approximately -0.05 and -0.12.

About 1955, Coffin (Ref. C.3) and Manson (Ref. C.4), who were working independently on thermal fatigue problems, established that plastic strain-life data could also be linearized with log-log coordinates (Figure C.8) and can be presented by the power-law function

$$\varepsilon_p = \frac{\Delta\varepsilon_p}{2} = \varepsilon_f' \, (2\,N_f)^c \tag{C.10}$$

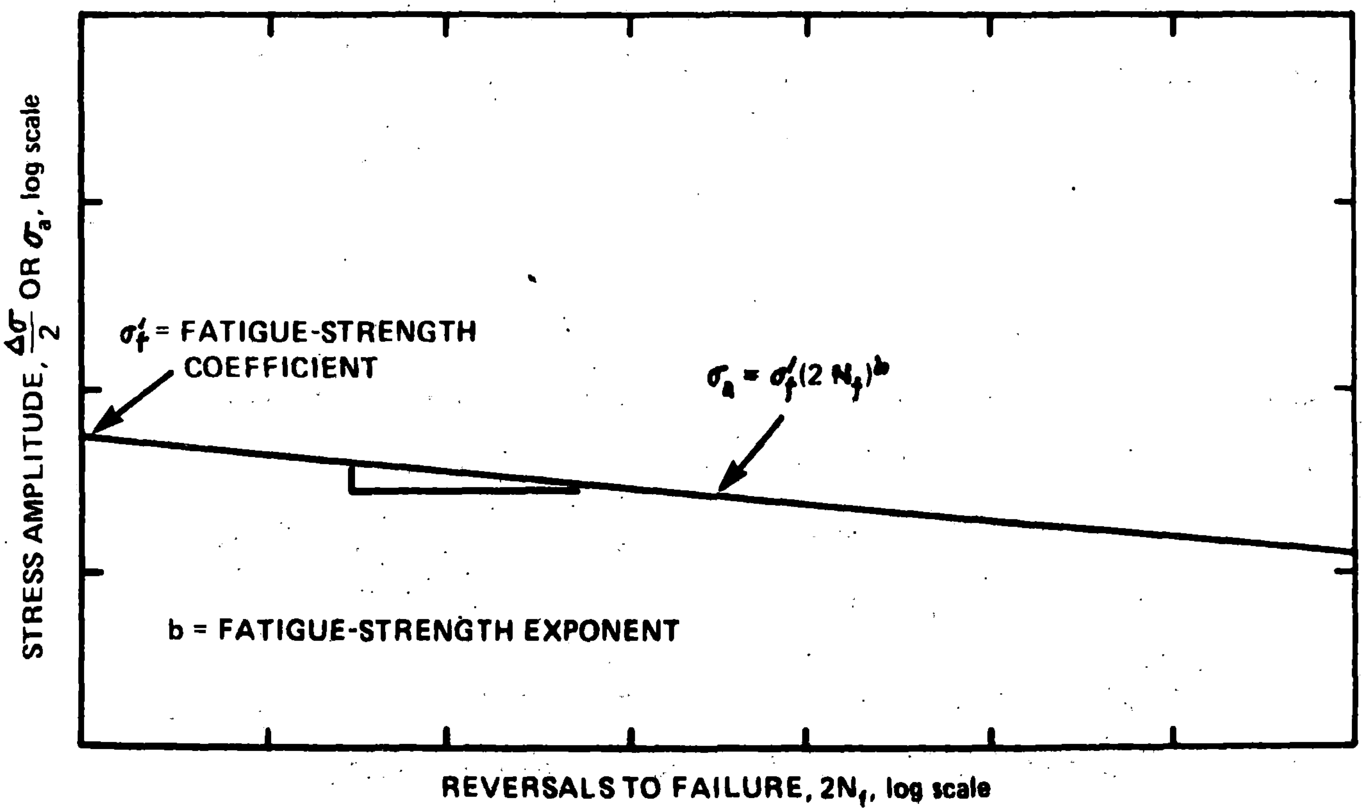


Figure C.7 FATIGUE-STRENGTH - LIFE RELATIONSHIP

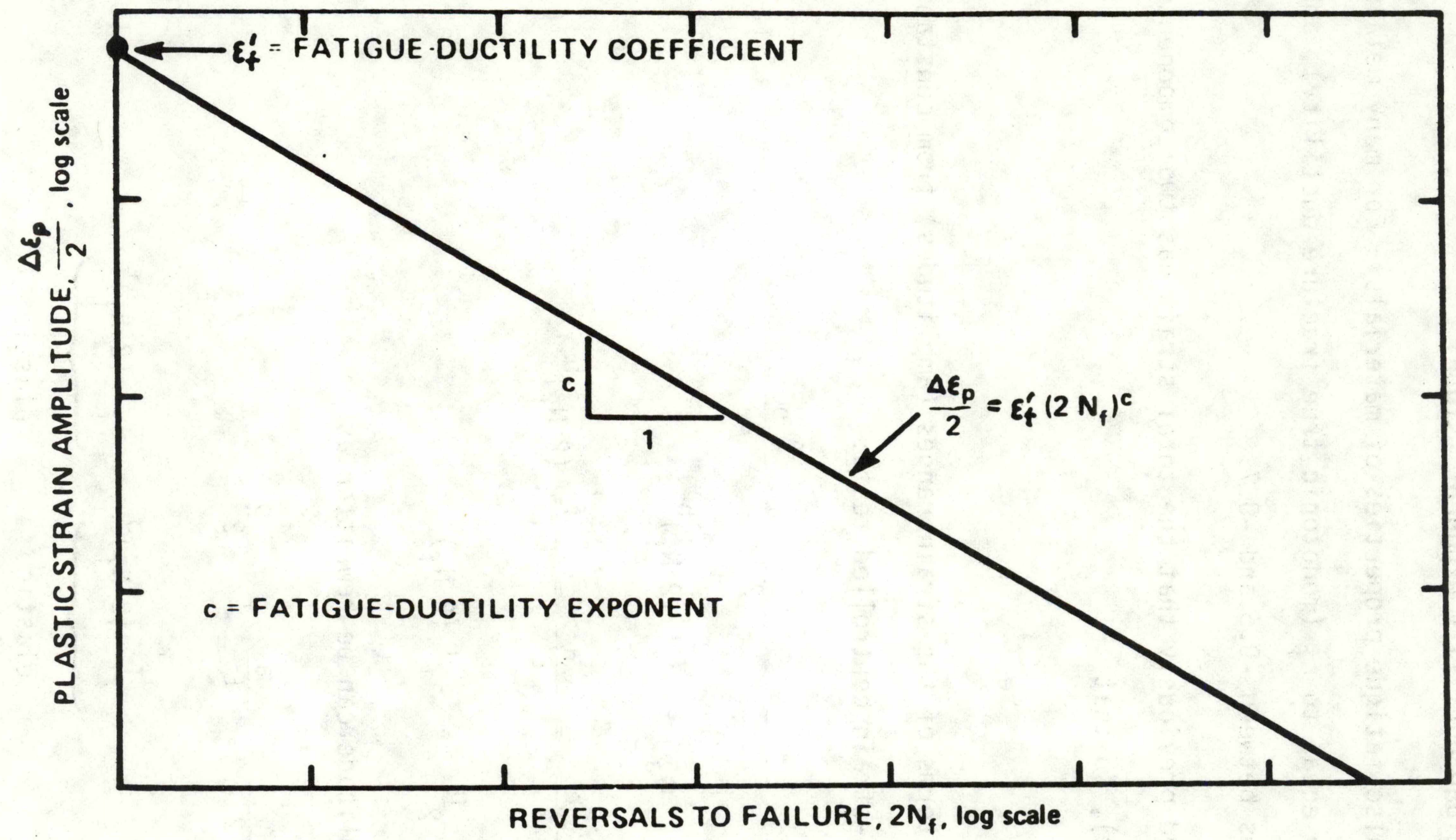


Figure C.8 FATIGUE-DUCTILITY - LIFE RELATIONSHIP where $\varepsilon_p = \frac{\Delta\varepsilon_p}{2}$ = True plastic strain amplitude

ε_f' = Fatigue ductility coefficient, intercept at $(2\,N_f) = 1$

c = Fatigue ductility exponent (slope)

ε_f' and c are also fatigue properties of materials. For many metals ε_f' is approximately equal to ε_f (monotonic true fracture ductility), and the value of c varies between -0.5 and -0.7.

It was mentioned previously that the total strain has two components (elastic and plastic), so that

$$\varepsilon = \varepsilon_e + \varepsilon_p$$

or, as expressed in terms of the strain ranges (amplitudes) from constant-amplitude, zero-mean-strain controlled tests:

$$\frac{\Delta\varepsilon}{2} = \frac{\Delta\varepsilon_e}{2} + \frac{\Delta\varepsilon_p}{2} \qquad \text{(C.11)}$$

Since $\sigma_a = \sigma_f' \, (2\,N_f)^b$

and $$\varepsilon_e = \frac{\Delta\varepsilon_e}{2} = \frac{\sigma_a}{E} = \frac{\sigma_f'}{E} \, (2\,N_f)^b \qquad \text{(C.12)}$$

$$\varepsilon_p = \frac{\Delta\varepsilon_p}{2} = \varepsilon_f' \, (2\,N_f)^c$$

the total strain amplitude can be rewritten as

$$\varepsilon_a = \frac{\Delta\varepsilon}{2} = \frac{\sigma_a}{E} + \frac{\Delta\varepsilon_p}{2}$$

or $$\varepsilon_a = \frac{\Delta\varepsilon}{2} = \underbrace{\frac{\sigma_f'}{E} \, (2\,N_f)^b}_{\text{elastic}} + \underbrace{\varepsilon_f' \, (2\,N_f)^c}_{\text{plastic}} \qquad \text{(C.13)}$$

It should be noted that Eq. (C.13) is the foundation for the strain-based approach and is referred to as the strain-life relationship. It consists of two straight lines, one for the elastic and another for the plastic strain, as shown in Figure C.9.

Several conclusions may be made from the strain-life curve in Figure C.9.

1) The two straight lines intersect at the transition fatigue life, $(2\ N_t)$ where

$$\frac{\Delta\varepsilon_p}{2} = \frac{\Delta\varepsilon_e}{2} \qquad (C.14)$$

By equating the elastic and plastic components of the total strain, one obtains

$$(2\ N_t) = \left(\frac{\varepsilon_f'\ E}{\sigma_f'}\right)^{1/(b-c)} \qquad (C.15)$$

2) At short life range, less than $(2\ N_f)$, the plastic strain predominates and ductility controls material performance.

3) In the longer life range, greater than $(2\ N_t)$, the elastic strain is more dominant than the plastic, and the strength of materials governs the fatigue behavior.

4) The strain-life relationships, as presented above, apply only to wrought metals. When internal defects control life (as is the case with cast metals, higher-hardness wrought steels, notches, weldments and so forth), these relationships are not directly applicable, and appropriate modifications to account for such "internal micronotches" must be made (Ref. C.5).

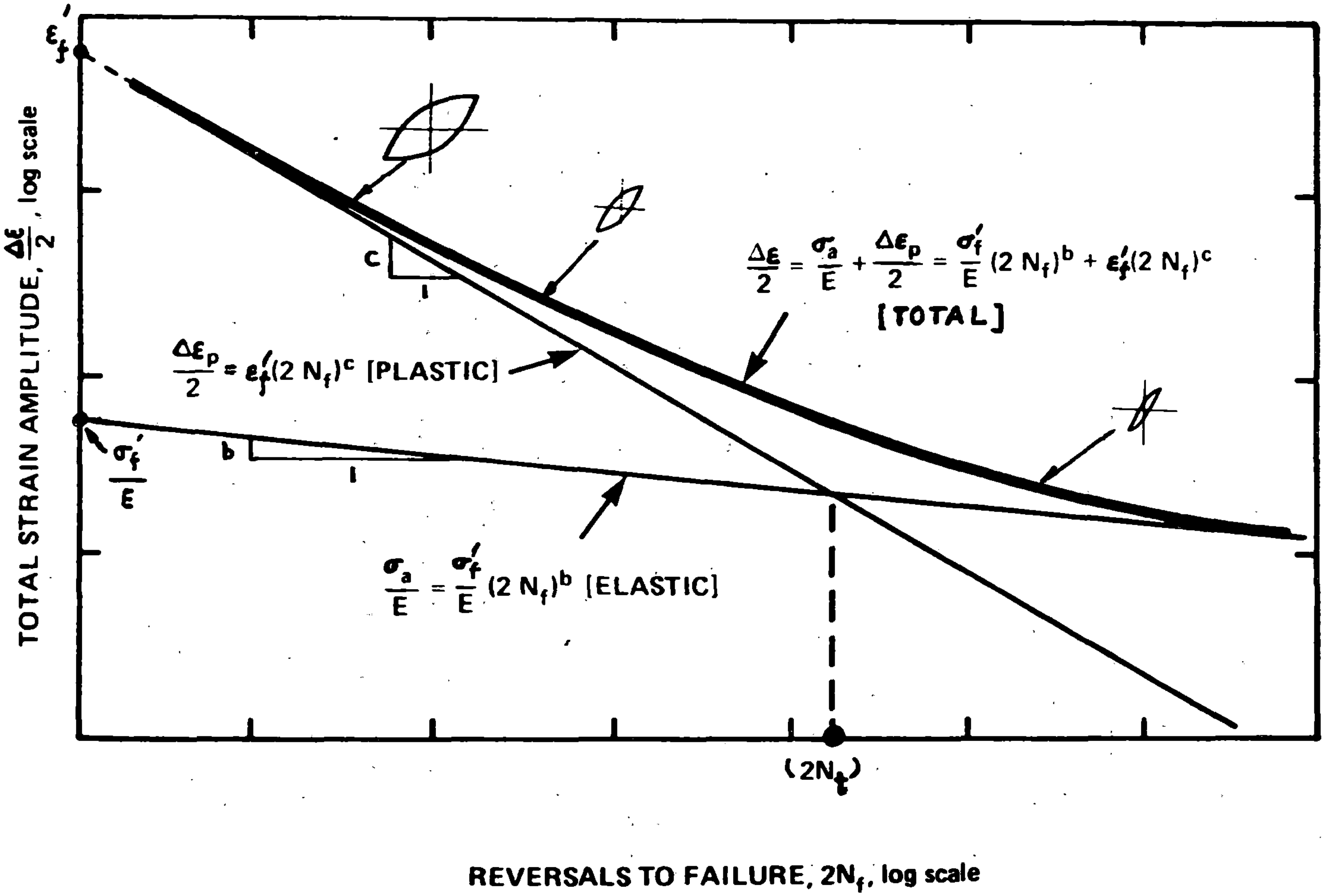


Figure C.9 STRAIN-(FATIGUE) LIFE RELATIONSHIP

C.3.2 Fatigue Property Data Relationships

Cyclic stress-strain material properties may be related to each other in the following manner:

$$K' = \frac{\sigma_f'}{(\epsilon_f')^{n'}} \tag{C.16}$$

Morrow (Ref. C.7) suggested that

$$b = \frac{-n'}{(1 + 5n')} \tag{C.17}$$

and

$$c = \frac{-1}{(1 + 5n')} \tag{C.18}$$

Thus,

$$n' = \frac{b}{c} \tag{C.19}$$

which allows a relationship between fatigue properties and cyclic stress-strain properties. If the average values of b and c (-0.09 and -0.6, respectively) are used in Eq. (C.19), then $n' = 0.15$. This agrees with the average value of n' observed for most metals. In the absence of adequate data from constant-strain-amplitude tests, it is often necessary to approximate the strain-life constants from the monotonic property constants. For example, the following approximate relationships are sometimes useful in estimating fatigue properties of ductile steels from tensile test data.

1) Fatigue Strength Coefficient, σ_f'

$$\sigma_f' \cong \sigma_f \text{ (corrected for necking)} \tag{C.20}$$

For many steels with hardness less than 500 HB, the fatigue limit at (2×10^6) reversals can be approximated by:

$$S_f \text{ (ksi)} \cong \frac{S_u}{2} = 0.25 \times (HB), \text{ for HB} < 500$$

or $S_u \cong \frac{HB}{2}$

where HB = Brinell hardness number

S_f = Fatigue-strength limit

S_u = Engineering ultimate tensile strength

For steels to about 500 HB:

$$\sigma_f \text{ (ksi)} \cong (S_u + 50) \quad \text{(C.21)}$$

For the intercept at $(2 N_f)_t$:

$$\frac{\Delta\varepsilon_e}{2} = \frac{\sigma_f'}{E} \cong \frac{\sigma_f}{E} \cong \frac{S_u + 50}{E} \quad \text{(C.22)}$$

2) Fatigue Ductility Coefficient, ε_f'

For practical purposes,

$$\varepsilon_f' \cong \varepsilon_f = \ln\left(\frac{1}{1 - RA}\right) \quad \text{(C.23)}$$

3) Fatigue Strength Exponent, b

As mentioned previously, the value of b varies from -0.05 to -0.12 and for most metals has an average of -0.085. In approximating the fatigue strength at (2×10^6) reversals with $1/2\ S_u$, it may be shown that:

$$b = -\frac{1}{6} \operatorname{Log}\left(\frac{2\sigma_f}{S_u}\right) \quad \text{(C.24)}$$

$= -0.10$ for soft steels

$= -0.05$ for hardened steels

4) Fatigue Ductility Exponent, c

The value of c has not been so well defined as the other constants. For instance:

$$c = -0.5 \text{ (by Coffin)} \tag{C.25}$$

$$c = -0.6 \text{ (by Manson)}$$

$$c = -0.5 \text{ to } -0.7 \text{ or average of } -0.6 \text{ (by Morrow)}$$

$$= -0.5 \text{ for soft steels}$$

$$= -0.7 \text{ for hardened steels}$$

It should be noted that the above crude approximations of the fatigue properties of steels should be used only for preliminary design estimates when proper laboratory data are not yet available.

C.3.3 Effects of Mean Stresses

Mean stress effects may be introduced in various forms as shown in Figure C.10 in which mean stress data are generally presented in terms of constant life diagrams-plots of all combinations of alternating and mean stresses resulting in the same finite life to failure. The equation for the lines illustrated in the figure are:

$$\text{Soderberg:} \quad \frac{\sigma_a}{\sigma_{cr}} + \frac{\sigma_o}{\sigma_y} = 1.0 \qquad \text{(a)}$$

$$\text{Goodman:} \quad \frac{\sigma_a}{\sigma_{cr}} + \frac{\sigma_o}{\sigma_u} = 1.0 \qquad \text{(b)} \qquad \text{(C.26)}$$

$$\text{Gerber:} \quad \frac{\sigma_a}{\sigma_{cr}} + \left(\frac{\sigma_o}{\sigma_u}\right)^2 = 1.0 \qquad \text{(c)}$$

$$\text{Morrow:} \quad \frac{\sigma_a}{\sigma_{cr}} + \frac{\sigma_o}{\sigma_f} = 1.0 \qquad \text{(d)}$$

where σ_a = Alternating stress amplitude

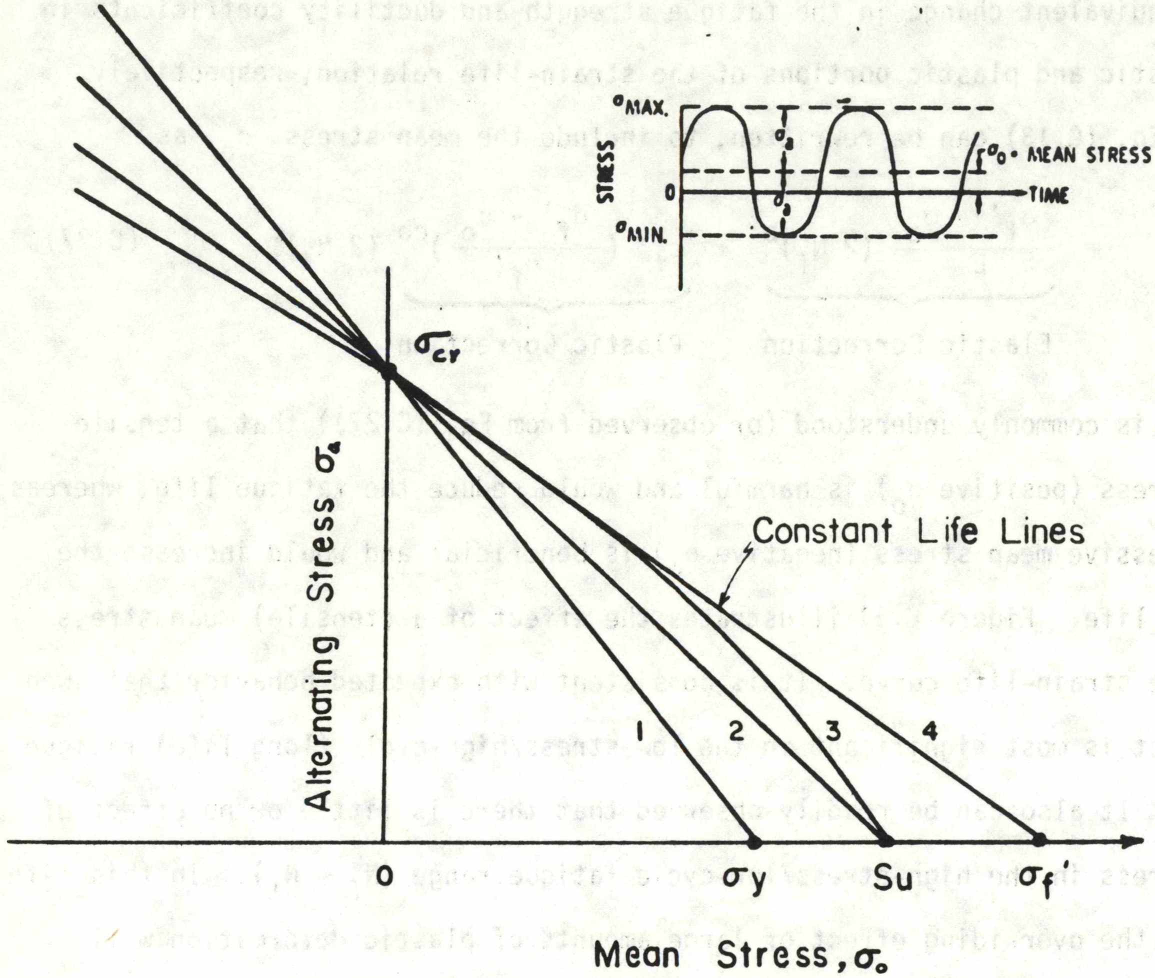
σ_{cr} = Completely reversed stress amplitude for a given life

S_u = Ultimate tensile strength

σ_o = Mean stress.

In general, it can be stated that:

1) Soderberg's relation is conservative for most cases.
2) Goodman's correction is good for brittle metals, but conservative for ductile metals.
3) Gerber's equation is applicable to ductile metals.
4) No comment has been offered regarding Morrow's approach.



1 Soderberg

2 Goodman

3 Gerber

4 Morrow

Figure C.10 MEAN STRESS CORRECTIONS

Morrow (Ref. C.8) suggested that mean stress effects may be incorporated as an equivalent change in the fatigue strength and ductility coefficients in the elastic and plastic portions of the strain-life relation, respectively. Hence, Eq. (C.13) can be rewritten, to include the mean stress, σ_o, as

$$\varepsilon = \underbrace{\frac{\sigma_f' - \sigma_o}{E} (2\ N_f)^b}_{\text{Elastic Correction}} + \underbrace{\varepsilon_f' \left(\frac{\sigma_f' - \sigma_o}{\sigma_f'}\right)^{cb}}_{\text{Plastic Correction}} (2\ N_f)^c \qquad \text{(C.27)}$$

It is commonly understood (or observed from Eq. (C.27)) that a tensile mean stress (positive σ_o) is harmful and would reduce the fatigue life, whereas a compressive mean stress (negative σ_o) is beneficial and would increase the fatigue life. Figure C.11 illustrates the effect of a (tensile) mean stress upon the strain-life curve. It is consistent with expected behavior that such an effect is most significant in the low-stress/high-cycle (long life) fatigue regime. It also can be readily observed that there is little or no effect of mean stress in the high-stress/low-cycle fatigue range ($N_f < N_t$). In this life region, the overriding effect of large amounts of plastic deformation will eradicate any detrimental (for tensile mean) or beneficial (for compressive mean) effect of mean stress.

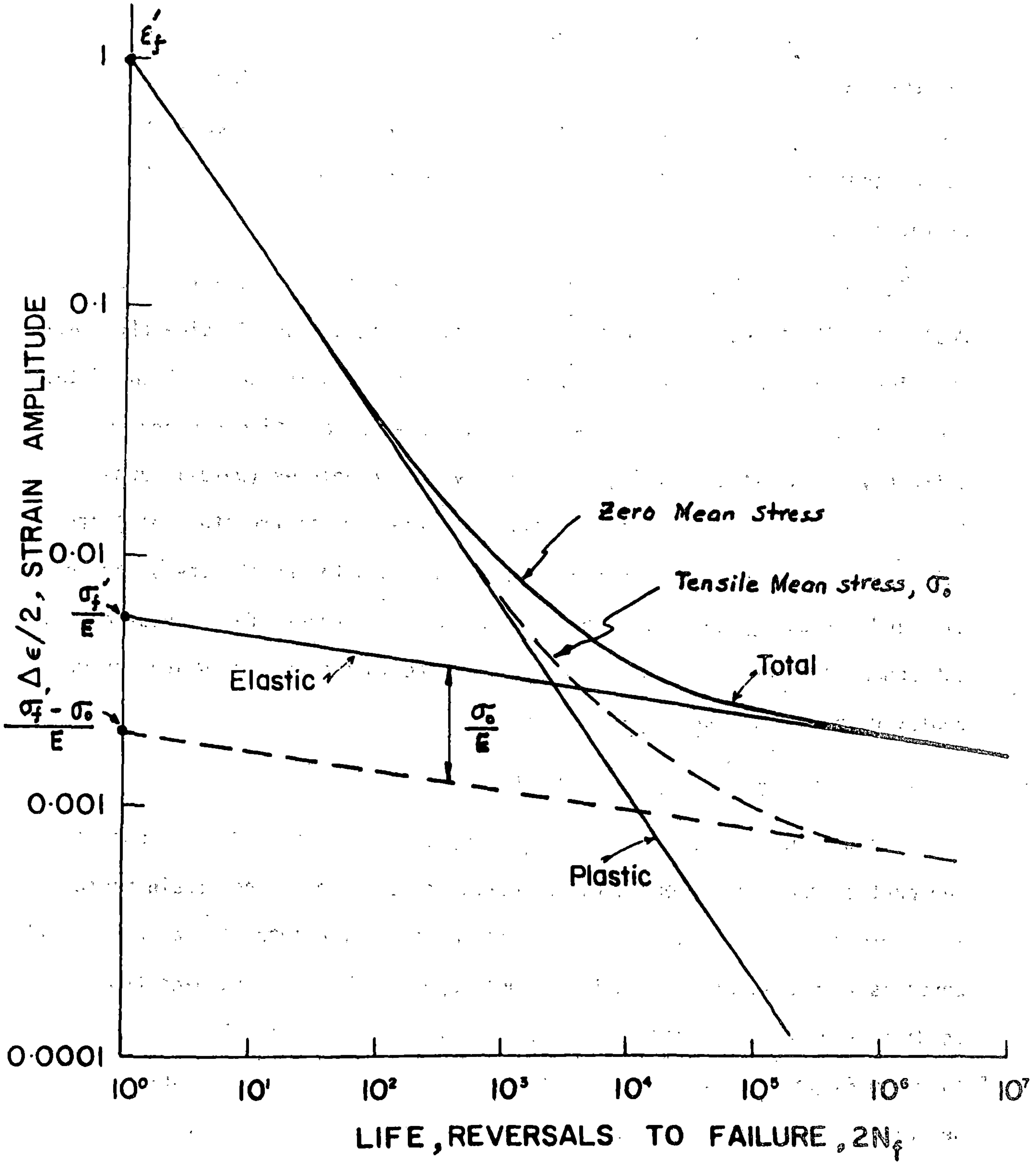


Figure C.11 STRAIN-LIFE CURVE WITH MEAN STRESS MODIFICATION

C.4 Local Strain-Life Analysis Procedures (Fatigue Analysis of Notched Components)

Since failures are likely to occur where notches, defects, or stress raisers are located, it becomes desirable to predict the fatigue life of a member with a notch directly, under a complex loading. This necessitates the determination of the cyclic history of the resulting strain and the stress at the notch root.

It should be clearly understood that the foregoing strain-life procedures deal with situations in which the local strain-time history is directly available for either smooth specimens or notched components at the location of the greatest strain. However, in practice, the local strain-time history is extremely difficult to obtain, particularly at the notches (welds), under field conditions. Under such circumstances, the nominal (remote field) strain data should be used in conjunction with corrections for notch stress concentration and fatigue notch factor before entering the strain-life computation. The subsequent sections contain discussions related to the methods which can be employed to achieve such objectives.

C.4.1 Method 1: Experimental Approach

One method is to attach strain gage(s) to the notch, and then subject the part to the prescribed loading. During this process, the strain history is recorded at the notch. The fatigue behavior at the notch (localized plastic zone) is then simulated by testing smooth specimens under strain-controlled conditions using the recorded strain history data as input. However, as shown in Figure C.1, the minimum cross section of the smooth specimen should be some fraction of the plastic-zone size. Furthermore, suitable correction factors must be used to account for differences in stress state, size, and strain gradient between the smooth specimen and the actual plastic zone for the structural detail of interest (Ref. C.23).

C.4.2 Method 2: Applications of Neuber's Rule

In dealing with complex geometries, it is necessary to relate loads and nominal stresses to the local stress-strain response at the critical locations (notches). This can be conveniently accomplished by using the finite-element method with elastic elements. The resulting strains and stresses at the notch are used to determine the theoretical stress concentration factor, K_t. One of the most popular means of determining K_t is the Neuber approach (Ref. C.24) in which K_t is determined by the geometric mean value of the elastic stress and strain concentration factors K_σ and K_ε, respectively.

$$K_t^2 = K_\sigma \times K_\varepsilon \tag{C.28}$$

For monotonic loading,

$$K_\sigma = \frac{\sigma \text{ (notch stress)}}{S \text{ (nominal, unnotched stress)}} \tag{C.29}$$

$$K_\varepsilon = \frac{\varepsilon \text{ (notched strain)}}{e \text{ (unnotched strain)}} \tag{C.30}$$

Hence,

$$K_t = \left(\frac{\sigma\varepsilon}{Se}\right)^{1/2} \tag{C.31}$$

or

$$K_t^2 \, Se = \sigma\varepsilon \tag{C.32}$$

For fatigue analysis, the stress concentration factor K_t, is often replaced by the fatigue notch factor, K_f, which can be similarly derived from the Neuber relationship in terms of the stress and strain ranges (Ref. C.25).

$$K_f^2 \, \Delta S \, \Delta e = \Delta\sigma\Delta\varepsilon \tag{C.33}$$

where $\Delta\sigma$, $\Delta\varepsilon$ = Local stress and strain ranges at notch root

ΔS, Δe = Nominal stress and strain ranges remote from notch

K_f = Fatigue notch factor for a particular material and geometry at a finite life, for example, 10^7 cycles.

Again, in most cases, the far field is elastic and Eq. (C-33) can be rewritten as:

$$\frac{(K_f \; \Delta S)^2}{E} = \Delta\sigma\Delta\varepsilon \tag{C.34}$$

Values of K_f are either experimentally determined for various materials, by testing small notched specimens in the laboratory for all stress ranges of ΔS, or calculated from Peterson's formula. This subject will be discussed in the following section (C.5).

It is noted that Eq. (C.34) is of the form of a rectangular hyperbola, often referred to as "Neuber's hyperbola", and is only valid when the nominal stresses are elastic.

Three methods of solution in association with Neuber's rule are envisioned here.

C.4.2.1 <u>Neuber's Rule with Strain-Life Relation</u>

Recalling the stress-life and strain-life relationships from Eq. (C.9) and Eq. (C.13), respectively:

$$\Delta\sigma = 2\,\sigma_f'\,(2\,N_f)^b \qquad \text{(a)}$$

and

$$\Delta\varepsilon E = 2\,\sigma_f'\,(2\,N_f)^b + 2\,\varepsilon_f' E\,(2\,N_f)^c \qquad \text{(b)} \tag{C.35}$$

Then, the strain-life relation of Eq. (3.53) can be rewritten as

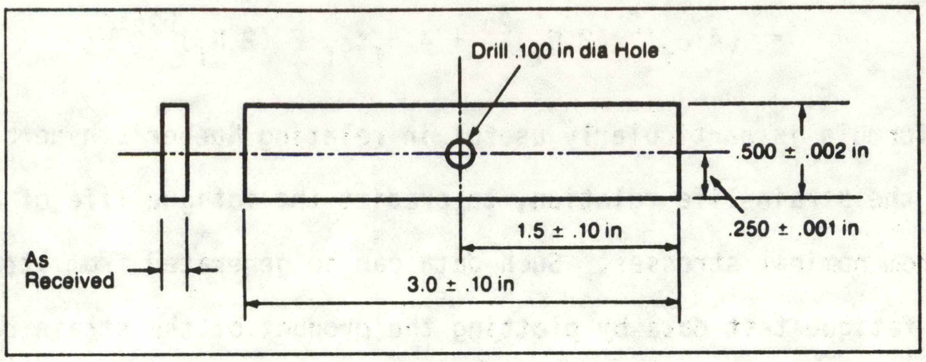
$$(K_f \, \Delta S)^2 = (\Delta\sigma\Delta\varepsilon E)$$

$$= \{4 \, \sigma_f'^2 \, (2 \, N_f)^{2b} + 4 \, \sigma_f'\varepsilon_f'E \, (2 \, N_f)^{b+c}\} \quad \text{(C.36)}$$

This formula is particularly useful in relating Nueber's hyperbolic parameter with the strain-life relation, to predict the fatigue life of a notched specimen from nominal stresses. Such data can be generated from strain-controlled fatigue test data by plotting the product of the strain range and corresponding steady-state stress range as a function of fatigue life ($2 \, N_f$); Figure C.12. By entering the appropriate value of nominal stress excursion, ΔS together with K_f, the life of a notched member can be predicted from either Eq. (C.36) or directly from Figure C.12. It is noted that mean stresses are ignored in this type of treatment.

C.4.2.2 Neuber's Hyperbola with Cyclic Stress-Strain Curve

An alternative method is to use Neuber's relation in conjunction with the cyclic stress-strain curve of the material mentioned earlier. For a given nominal stress excursion, ΔS, the local (notched) region undergoes deformation until the product of local stress and strain is equal to Neuber's constant, the left side of Eq. (C.34). This process is illustrated in Figure C.13 (a), in which the local (notched) region is assumed to deform along the cyclic stress-strain curve until it intersects with Neuber's hyperbola defined by Eq. (C.34). This intersection point determines the value of the local stress and strain ranges resulting from the excursion stress, ΔS. Utilization of this technique in a step-by-step complex history analysis is described in Ref. C.26 through Ref. C.28 and is illustrated in Figure C.13 (b).



Notch-Sensitivity Specimen (K_t = 2.5)

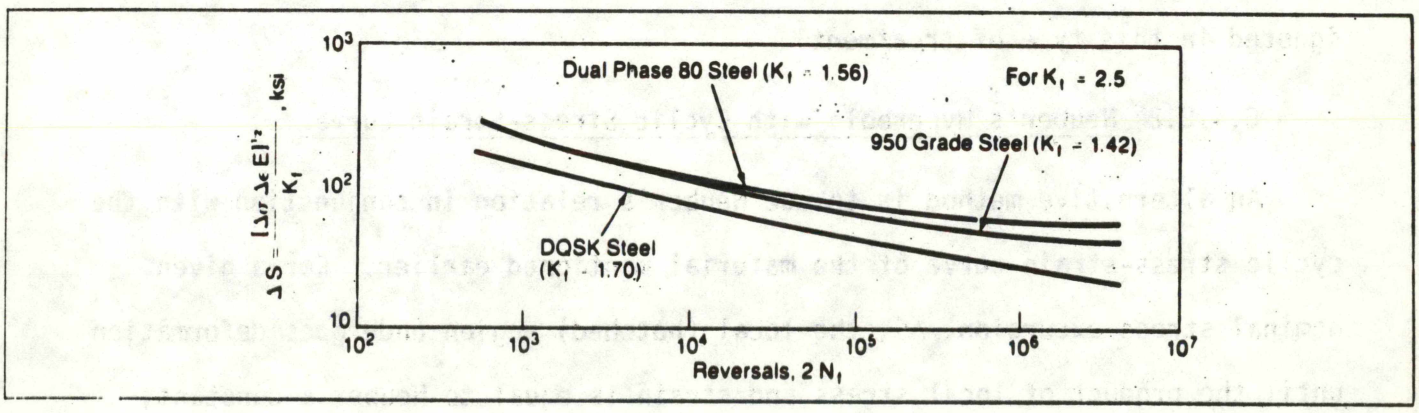
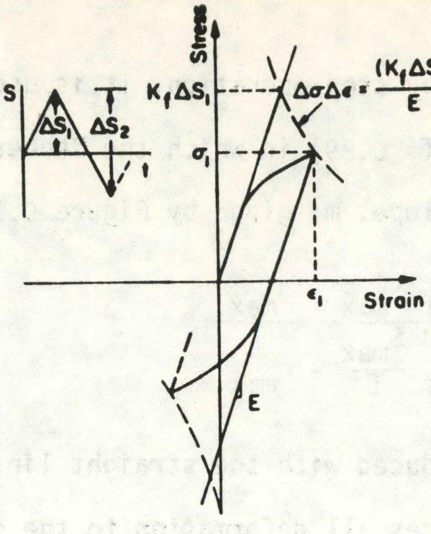
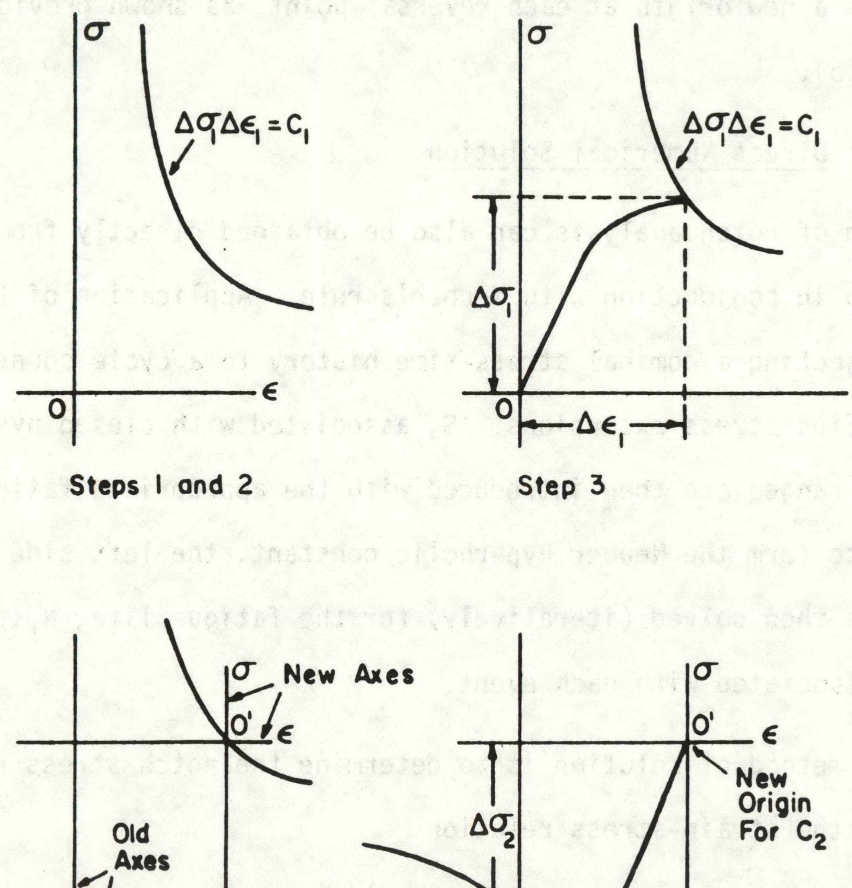


Figure C.12 NEUBER PLOTS OF NOTCHED FATIGUE-TEST DATA



(a)



(b)

Figure C.13 PROCEDURE FOR DETERMINING LOCAL STRESS AND STRAIN AT NOTCH ROOT DUE TO NOMINAL STRESS EXCURSION

In lieu of the step-by-step operation, it is useful to employ a modification developed by Stadnick (Ref. C.29) in which the Neuber hyperbola is approximated by a straight line with slope, m, given by Figure C.14:

$$m = \frac{K_f\, S_{max} - \sigma_{max}}{K_f\, \frac{S_{max}}{E} - \varepsilon_{max}} \qquad (C.37)$$

While some error is introduced with the straight line approximation, it has the advantage that it references all deformation to the original stress-strain origin. Hence, it will accumulate less total error than the Neuber method, which defines a new origin at each reversal point, as shown previously in Figure C.13 (b).

C.4.2.3 Direct Numerical Solution

Solution of notch analysis can also be obtained directly from the strain-life relation in conjunction with Neuber's rule. Application of this approach involves subjecting a nominal stress-time history to a cycle counting algorithm which identifies stress excursions, ΔS, associated with closed hysteresis loops. These stress ranges are then introduced with the appropriate fatigue notch factor, K_f, to form the Neuber hyperbolic constant, the left side of Eq. (C.34). Eq. (C.36) is then solved (iteratively) for the fatigue life, N_f, to determine the damage associated with each event.

Another method of solution is to determine the notch stress range by substituting the strain-stress relation

$$\frac{\Delta\varepsilon}{2} = \frac{\Delta\sigma}{2E} + \left(\frac{\Delta\sigma}{2K'}\right)^{1/n'}$$

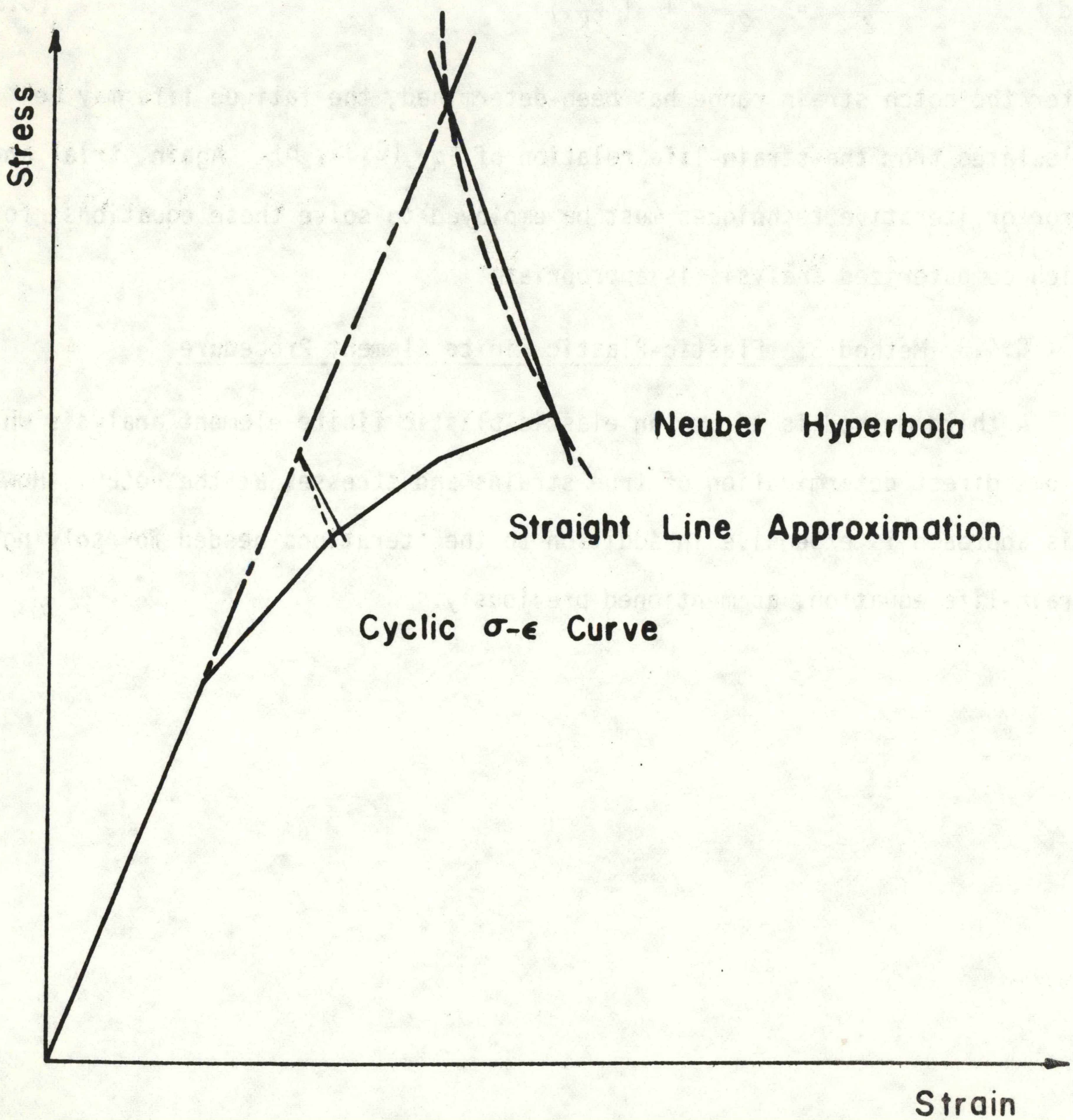


Figure C.14 CONSTRUCTION FOR DETERMINING NOTCH ROOT STRESS AND STRAIN into Neuber's hyperbola (Eq. (C.34)), to yield

$$\frac{(K_f\ \Delta S)^2}{4E} = \frac{(\Delta\sigma)^2}{4E} + \frac{\Delta\sigma}{2}\left(\frac{\Delta\sigma}{2K'}\right)^{1/n'} \tag{C.38}$$

and

$$\frac{\Delta\varepsilon}{2} = \frac{\Delta\sigma}{2E} + \left(\frac{\Delta\sigma}{2K'}\right)^{1/n'} \tag{C.39}$$

After the notch strain range has been determined, the fatigue life may be calculated from the strain-life relation of Eq. (C.35, b). Again, trial and error or iterative techniques must be employed to solve these equations, for which computerized analysis is appropriate.

C.4.3 Method 3: Elastic-Plastic Finite Element Procedure

A third method is to use an elastic-plastic finite element analysis which allows direct determination of true strains and stresses at the notch. However, this approach is expensive in addition to the iterations needed for solving the strain-life equation, as mentioned previously.

C.5 Determination of Fatigue Notch Factor, K_f

C.5.1 Experimental Method

The fatigue notch factor can be experimentally determined from a series of strain controlled cyclic tests. A Neuber parameter-fatigue life curve can be constructed from either the cyclic stress-strain curve as shown in Figure C.15, or the Neuber parameter equation of Eq. (C.36).

Since the nominal stresses are assumed to be elastic, the Neuber parameter reduces to

$$(\Delta S\ \Delta e) = \frac{\Delta S^2}{E} \tag{C.40}$$

which can be plotted on the same figure as the smooth specimen data. The fatigue notch factor is then the square root of the difference between these curves, as depicted in Figure C.15.

C.5.2 Analytical Solution

Many investigators have attempted to determine the value of K_f analytically. One, which is considered more successful, is attributed to Peterson (Ref. C.30). It uses a fatigue notch factor determined from

$$K_f = 1 + \frac{K_t - 1}{1 + \frac{a}{r}} < K_t \tag{C.41}$$

where K_t = Theoretical stress concentration factor = $\frac{\sigma}{S}$

a = Material constant depending on strength and ductility

r = Notch tip radius

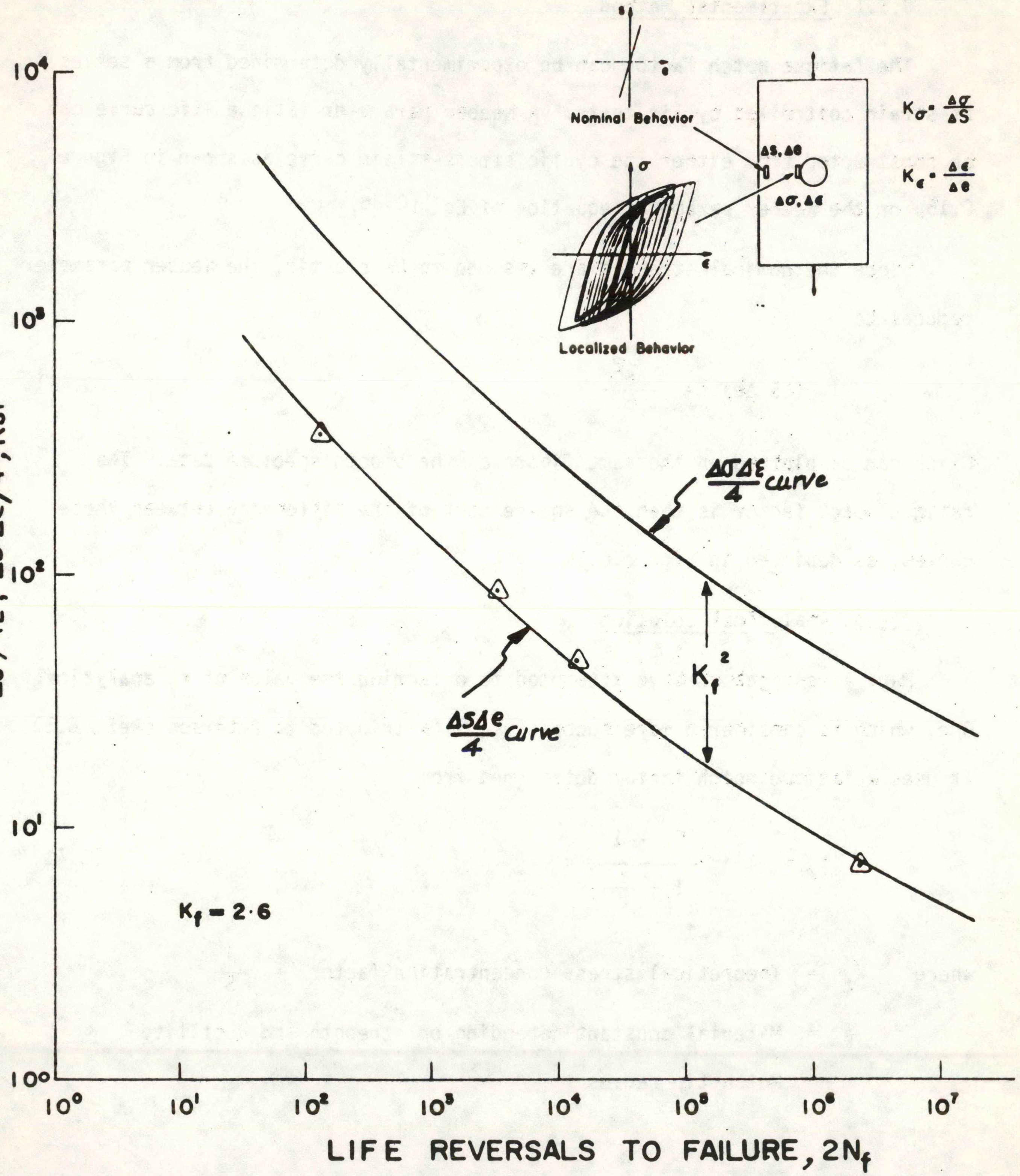


Figure C.15 NEUBER'S PARAMETER - LIFE CURVE

For ferrous-based wrought metals,

$$a = \left(\frac{300}{S_u}\right)^{1.8} \times 10^{-3} \text{ in.}$$

or $\quad = 0.01$ in. for normalized or annealed steels

$= 0.001$ in. for highly hardened steels

$= 0.0025$ in. for quenched-and-tempered steels

Figure C.16 illustrates the effect of r on K_f for hard and soft metal. The figure indicates that:

- o $K_f = 1$: for small notch radius ($r < 0.1a$)
- o $K_f = K_t$: for large notch radius ($r > 10a$)

In addition it should be noted that in the elastic range

$$K_\sigma = K_\varepsilon$$

or

$$K_f = K_t$$

However, after yielding takes place, the strain concentration factor, K_ε, begins to increase whereas the stress concentration factor, K_σ, decreases. This is illustrated in Figure C.17.

In addition, a notch-sensitivity index is defined as

$$q = \frac{K_f - 1}{K_t - 1} \qquad \text{(C.42)}$$

which varies from 0 (no notch effect) to 1 (full effect). It should be apparent that small notches (such as inclusions) are less effective than larger notches created by geometric discontinuities that reduce the fatigue resistance.

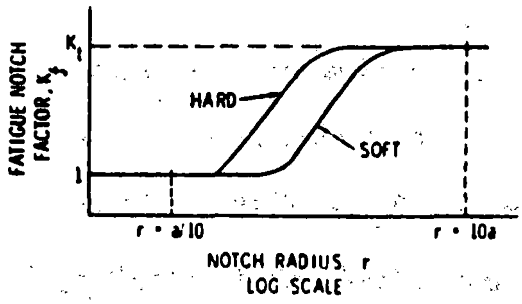


Figure C.16 FATIGUE-NOTCH FACTOR VERSUS NOTCH RADIUS AS A FUNCTION OF RELATIVE HARDNESS

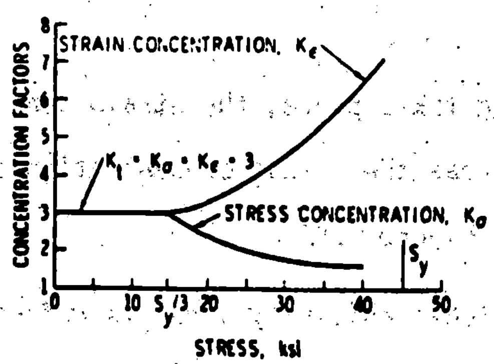


Figure C.17 SCHEMATIC OF CHANGE IN STRESS-CONCENTRATION AND STRAIN-CONCENTRATION FACTORS AS YIELDING OCCURS AT NOTCH ROOT

C.5.3 Determination of K_f for Weld Joints

It is observed through microscopic examination of weld toes that the notch-root radius of discontinuities at weld toes is variable; practically any value of radius can be observed. Thus, notches, such as weld toes, must have all possible values of K_f. To logically characterize the notch factor for a given weld configuration, the application of a maximum value of K_f, $(K_f)_{max}$, computed from Eq. (C.41) for various values of r, was suggested (Ref. C.31) in which the value of K_t can be determined by finite element computations for each value of notch radius examined. A typical application of this technique is demonstrated in Figure C.18, for a butt weld joint.

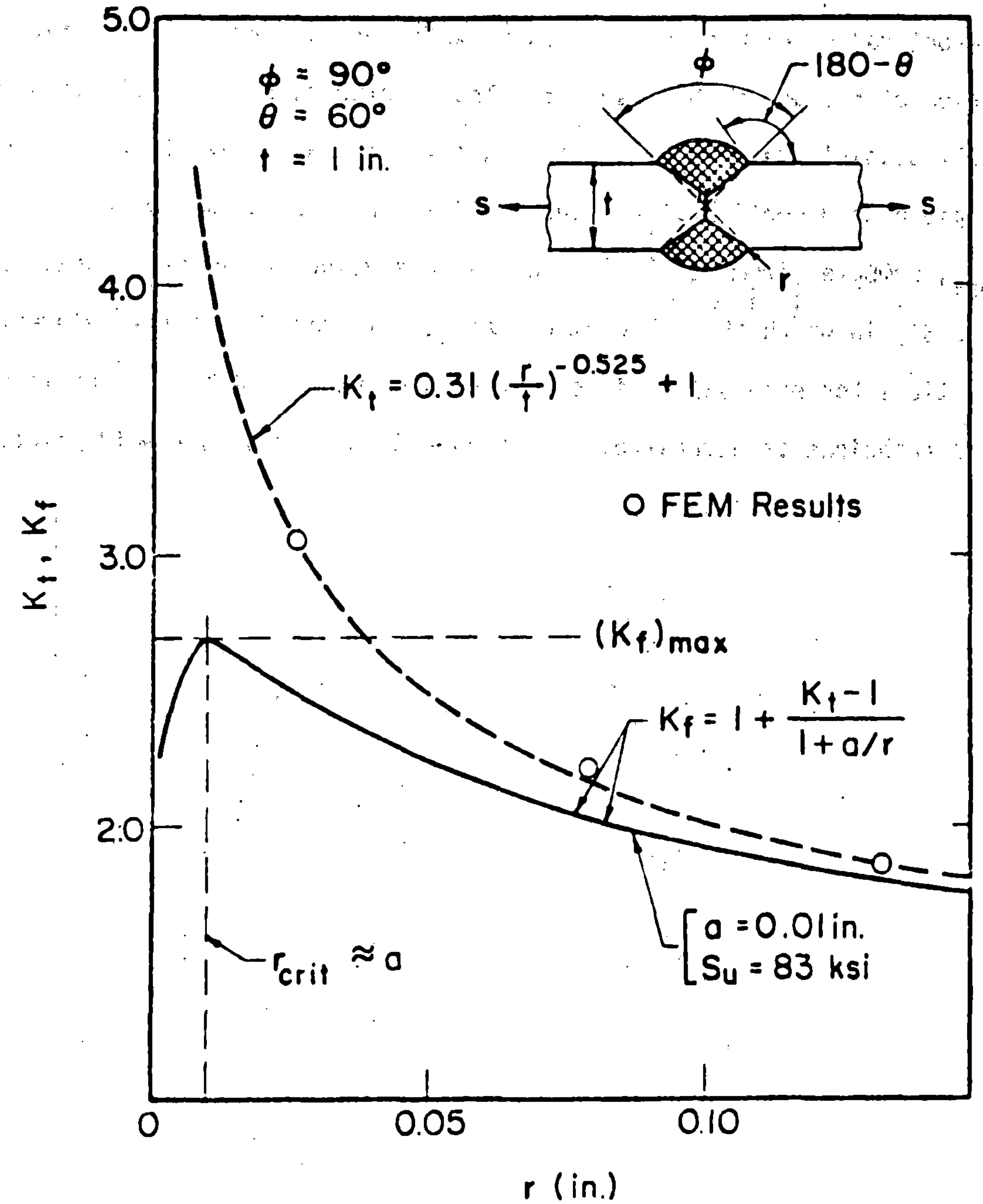


Figure C.18 ELASTIC STRESS CONCENTRATION FACTOR (K_t) AND FATIGUE NOTCH FACTOR (K_f) AS A FUNCTION OF TOE ROOT RADIUS

APPENDIX D

FORTRAN Program Listing of Strain-Life Equation Solution

STRAIN-LIFE EQUATION FORTRAN PROGRAM

```
      DIMENSION PMN(100) , PMX(100) , FRQ(100)

  100 FORMAT( '   TABLE OF CONSTANTS'/
     $      '         STATIC BASE STRESS..............' , F8.0 , ' PSI' /
     $      '         RESIDUAL STRESS.................' , F8.0 , ' PSI' /
     $      '         S/P RATIO-NEAR SIDE.............' , F8.2 ,' PSI/KIP'
     $/     '                    -FAR SIDE............' , F8.2 ,' PSI/KIP'
     $/     '         FATIGUE STRENGTH COEFFICIENT....' , F8.0 , ' PSI' /
     $      '         FATIGUE DUCTILITY COEFFICIENT...' , F8.5 /
     $      '         FATIGUE STRENGTH EXPONENT.......' , F8.3 /
     $      '         FATIGUE DUCTILITY EXPONENT......' , F8.3 /
     $      '         YOUNGS MODULUS..................' , F10.0 , ' PSI' /
     $      '         FATIGUE NOTCH FACTOR............' , F8.3 / /
     $    '     CYCLE      MAX LOAD  MIN LOAD  MAX STRESS  MIN STRESS',
     $      '   MEAN STRESS  CYCLES TO          ' /
     $    '   FRACTION     (KIP)     (KIP)       (PSI)        (PSI)   ',
     $      '     (PSI)       FAILURE      DAMAGE  ' /
     $      '-----------------------------------------------------------',
     $      '------------------------------------------' / / )
  101 FORMAT( 2X , F10.8 , 3X , F6.0 , 4X , F6.0 , 3X , F9.0 , 3X ,
     $        F9.0 , 4X , F9.0 , 2X , E11.4 , 2X , E11.4 )
  102 FORMAT( / / '   DAMAGE SUM....' , E11.4 /
     $          '   LIFE..........' , E11.4 , ' CYCLES' )
  200 FORMAT( '   NO CONVERGENCE FOR ITEM NUMBER ' , I2 /
     $        '       ERROR FRACTION = ' , E12.5 )
  201 FORMAT( ' PRESS ENTER TO PROCEED' )

      ERROR = 0.000001
      LIMIT = 100
      SUMD = 0.
      READ(2,*)
      READ(2,*) N
      READ(2,*)
      DO 900 I = 1,N
        READ(2,*)  J , PMX(I) , PMN(I) , FRQ(I)
        FRQ(I) = FRQ(I) / 100.
  900 CONTINUE
      READ(2,*)
      READ(2,*) SS , SRES , PSP , PSN , SIG , EPS
      READ(2,*)
      READ(2,*) B , C , E , AKF

      CMB = C - B
      CBM1 = CMB / B

      WRITE(5,100)  SS , SRES , PSP , PSN , SIG , EPS , B , C , E , AKF

      I = 1
      DO 4000 IDLE = 1,100
        IF( I .GT. N ) GO TO 4100

        PMIN = PMN(I)
        PMAX = PMX(I)
        PSMIN = PSP
        IF( PMIN .LT. 0. ) PSMIN = PSN
```

```
      PSMAX = PSP
      IF( PMAX .LT. 0. )  PSMAX = PSN
C
      SMAX = SS + ( PMAX*PSMAX ) + SRES
      SMIN = SS + ( PMIN*PSMIN ) + SRES
C
      DELS = SMAX - SMIN
      SIGO = ( SMAX + SMIN ) / 2.
C
      CA = AKF*AKF*DELS*DELS / ( 4.*( SIG - SIGO )*( SIG - SIGO ) )
      CB = ( E*EPS / SIG )*( ( ( SIG - SIGO ) / SIG )**CBM1 )
C
      TNF1 = CA**( 0.5 / B )
      DO 1000 IDLE1 = 1,500
        IF( IDLE1 .GT. LIMIT )  GO TO 1010
C
        TEMP = CA / ( 1. + CB*( TNF1**CMB ) )
        TNF2 = TEMP**( 0.5 / B )
        TEMP = ( TNF2 - TNF1 ) / TNF2
        IF( ABS(TEMP) .LT. ERROR )  GO TO 1100
        TNF1 = TNF2
 1000 CONTINUE
 1010 CONTINUE
      WRITE(3,200)  I , TEMP
C
 1100 CONTINUE
C
      TNF = 0.5*( TNF1 + TNF2 )
      D = 2.*FRO(I) / TNF
      SUMD = SUMD + D
      ANF = TNF / 2.
C
      WRITE(5,101) FRO(I) , PMAX , PMIN , SMAX , SMIN , SIGO , ANF , D
C
      I = I + 1
 4000 CONTINUE
 4100 CONTINUE
C
      ALIFE = 1. / SUMD
      WRITE(5,102)  SUMD , ALIFE
C
      WRITE(3,201)
      READ(4,*) DUMMY
C
      STOP
      END
```

NUMBER OF CASES
42

CASE NO.	PMAX	PMIN	% FREQUENCY
1	100.0	0.0	1.415000
2	200.0	0.0	0.013000
3	300.0	0.0	0.001000
4	0.0	-100.0	49.842000
5	100.0	-100.0	38.179000
6	200.0	-100.0	0.775000
7	300.0	-100.0	0.018000
8	400.0	-100.0	0.001000
9	-100.0	-200.0	0.001000
10	0.0	-200.0	1.363000
11	100.0	-200.0	5.463000
12	200.0	-200.0	1.629000
13	300.0	-200.0	0.116000
14	400.0	-200.0	0.006000
15	0.0	-300.0	0.007000
16	100.0	-300.0	0.252000
17	200.0	-300.0	0.453000
18	300.0	-300.0	0.174000
19	400.0	-300.0	0.014000
20	500.0	-300.0	0.002000
21	-200.0	-400.0	0.001000
22	-100.0	-400.0	0.001000
23	0.0	-400.0	0.001000
24	100.0	-400.0	0.009000
25	200.0	-400.0	0.047000
26	300.0	-400.0	0.067000
27	400.0	-400.0	0.026000
28	500.0	-400.0	0.004000
29	100.0	-500.0	0.001000
30	200.0	-500.0	0.003000
31	300.0	-500.0	0.017000
32	400.0	-500.0	0.019000
33	500.0	-500.0	0.002000
34	200.0	-600.0	0.001000
35	300.0	-600.0	0.002000
36	400.0	-600.0	0.003000
37	500.0	-600.0	0.004000
38	600.0	-600.0	0.002000
39	-200.0	-700.0	0.001000
40	200.0	-700.0	0.001000
41	400.0	-700.0	0.001000
42	500.0	-700.0	0.002000

SS	SRES	S/P+	S/P-	SIGF	EPSF
10000.	50000.	10.00	10.00	120000.	0.50

B	C	E	AKF
-0.039	-0.6	29000000.	3.00

TABLE OF CONSTANTS

STATIC BASE STRESS.............. 10000. PSI
RESIDUAL STRESS................. 50000. PSI
S/P RATIO-NEAR SIDE............. 10.00 PSI/KIP
-FAR SIDE............. 10.00 PSI/KIP
FATIGUE STRENGTH COEFFICIENT... 120000. PSI
FATIGUE DUCTILITY COEFFICIENT.. 0.50000
FATIGUE STRENGTH EXPONENT...... -0.089
FATIGUE DUCTILITY EXPONENT..... -0.600
YOUNGS MODULUS.................. 29000000. PSI
FATIGUE NOTCH FACTOR........... 3.000

CYCLE FRACTION	MAX LOAD (KIP)	MIN LOAD (KIP)	MAX STRESS (PSI)	MIN STRESS (PSI)	MEAN STRESS (PSI)	CYCLES TO FAILURE	DAMAGE
0.01415000	100.	0.	61000.	60000.	60500.	0.4558E+18	0.3104E-19
0.00013000	200.	0.	62000.	60000.	61000.	0.1719E+15	0.7563E-19
0.00001000	300.	0.	63000.	60000.	61500.	0.1641E+13	0.6093E-17
0.49841994	0.	-100.	60000.	59000.	59500.	0.5497E+18	0.9067E-19
0.38178998	100.	-100.	61000.	59000.	60000.	0.2076E+15	0.1839E-14
0.00775000	200.	-100.	62000.	59000.	60500.	0.1986E+13	0.3903E-14
0.00018000	300.	-100.	63000.	59000.	61000.	0.7127E+11	0.2526E-14
0.00001000	400.	-100.	64000.	59000.	61500.	0.5279E+10	0.1894E-14
0.00001000	-100.	-200.	59000.	58000.	58500.	0.6609E+19	0.1513E-22
0.01363000	0.	-200.	60000.	58000.	59000.	0.2500E+15	0.5452E-15
0.05463000	100.	-200.	61000.	58000.	59500.	0.2395E+13	0.2281E-13
0.01628999	200.	-200.	62000.	58000.	60000.	0.8609E+11	0.1892E-12
0.00116000	300.	-200.	63000.	58000.	60500.	0.6336E+10	0.1816E-12
0.00006000	400.	-200.	64000.	58000.	61000.	0.7490E+09	0.8011E-13
0.00007000	0.	-300.	60000.	57000.	58500.	0.2879E+13	0.2431E-16
0.00252000	100.	-300.	61000.	57000.	59000.	0.1037E+12	0.2431E-13
0.00453000	200.	-300.	62000.	57000.	59500.	0.7702E+10	0.5882E-12
0.00174000	300.	-300.	63000.	57000.	60000.	0.9046E+09	0.1923E-11
0.00014000	400.	-300.	64000.	57000.	60500.	0.1457E+09	0.9606E-12
0.00002000	500.	-300.	65000.	57000.	61000.	0.2959E+08	0.6760E-12
0.00001000	-200.	-400.	58000.	56000.	57000.	0.3592E+15	0.2784E-19
0.00001000	-100.	-400.	59000.	56000.	57500.	0.3451E+13	0.2898E-17
0.00001000	0.	-400.	60000.	56000.	58000.	0.1244E+12	0.8036E-16
0.00009000	100.	-400.	61000.	56000.	58500.	0.9259E+10	0.9720E-14
0.00047000	200.	-400.	62000.	56000.	59000.	0.1089E+10	0.4315E-12
0.00067000	300.	-400.	63000.	56000.	59500.	0.1758E+09	0.3812E-11
0.00026000	400.	-400.	64000.	56000.	60000.	0.3574E+08	0.7276E-11
0.00004000	500.	-400.	65000.	56000.	60500.	0.8670E+07	0.4614E-11
0.00001000	100.	-500.	61000.	55000.	58000.	0.1308E+10	0.7649E-14
0.00003000	200.	-500.	62000.	55000.	58500.	0.2113E+09	0.1420E-12
0.00017000	300.	-500.	63000.	55000.	59000.	0.4303E+08	0.3951E-11
0.00019000	400.	-500.	64000.	55000.	59500.	0.1046E+08	0.1817E-10
0.00002000	500.	-500.	65000.	55000.	60000.	0.2922E+07	0.6846E-11
0.00001000	200.	-600.	62000.	54000.	58000.	0.5165E+08	0.1936E-12
0.00002000	300.	-600.	63000.	54000.	58500.	0.1257E+08	0.1591E-11
0.00003000	400.	-600.	64000.	54000.	59000.	0.3518E+07	0.8528E-11
0.00004000	500.	-600.	65000.	54000.	59500.	0.1103E+07	0.3628E-10
0.[illegible]020	[illegible]	-60[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

STRAIN-LIFE ANALYSIS OUTPUT

0.00001000	-200.	-700.	58000.	53000.	55500.	0.1531E+11	0.6324E-15
0.00001000	200.	-700.	62000.	53000.	57500.	0.1507E+09	0.6636E-12
0.00001000	400.	-700.	64000.	53000.	58500.	0.1326E+07	0.7544E-11
0.00002000	500.	-700.	65000.	53000.	59000.	0.4572E+06	0.4374E-10

DAMAGE SUM.... 0.2011E-09
LIFE.......... 0.4972E+10 CYCLES

STRAIN-LIFE ANALYSIS OUTPUT

STRESS-LIFE ANALYSIS OUTPUT

TABLE OF CONSTANTS
STATIC BASE STRESS............. 10000. PSI
S/P RATIO-NEAR SIDE............ 10.00 PSI/KIP
-FAR SIDE............ 10.00 PSI/KIP
NOMINAL YIELD STRESS........... 50000. PSI
MGD Y INTERCEPT................ 7400.00 PSI
MGD SLOPE...................... 1.00
YOUNGS MODULUS................. 29000000. PSI
S-N CURVE SLOPE................ 0.35

CYCLE FRACTION	MAX LOAD (KIP)	MIN LOAD (KIP)	MAX STRESS (PSI)	MIN STRESS (PSI)	REVERSAL RATIO	ENDURANCE LIMIT	CYCLES TO FAILURE
0.0[illegible]415000	100.	0.	11000.	10000.	0.909	50000.	0.2000E+07
0.00013000	200.	0.	12000.	10000.	0.833	44400.	0.2000E+07
0.00001000	300.	0.	13000.	10000.	0.769	32067.	0.2000E+07
0.49841994	0.	-100.	10000.	9000.	0.900	50000.	0.2000E+07
0.38178998	100.	-100.	11000.	9000.	0.818	40700.	0.2000E+07
0.00775000	200.	-100.	12000.	9000.	0.750	29600.	0.2000E+07
0.00018000	300.	-100.	13000.	9000.	0.692	24050.	0.2000E+07
0.00001000	400.	-100.	14000.	9000.	0.643	20720.	0.2000E+07
0.00001000	-100.	-200.	9000.	8000.	0.889	50000.	0.2000E+07
0.01363000	0.	-200.	10000.	8000.	0.800	37000.	0.2000E+07
0.05463000	100.	-200.	11000.	8000.	0.727	27133.	0.2000E+07
0.01628999	200.	-200.	12000.	8000.	0.667	22200.	0.2000E+07
0.00116000	300.	-200.	13000.	8000.	0.615	19240.	0.2000E+07
0.00006000	400.	-200.	14000.	8000.	0.571	17267.	0.2000E+07
0.00007000	0.	-300.	10000.	7000.	0.700	24667.	0.2000E+07
0.00252000	100.	-300.	11000.	7000.	0.636	20350.	0.2000E+07
0.00453000	200.	-300.	12000.	7000.	0.583	17760.	0.2000E+07
0.00174000	300.	-300.	13000.	7000.	0.538	16033.	0.2000E+07
0.00014000	400.	-300.	14000.	7000.	0.500	14800.	0.2000E+07
0.00002000	500.	-300.	15000.	7000.	0.467	13875.	0.1601E+07
0.00001000	-200.	-400.	8000.	6000.	0.750	29600.	0.2000E+07
0.00001000	-100.	-400.	9000.	6000.	0.667	22200.	0.2000E+07
0.00001000	0.	-400.	10000.	6000.	0.600	18500.	0.2000E+07
0.00009000	100.	-400.	11000.	6000.	0.545	16280.	0.2000E+07
0.00047000	200.	-400.	12000.	6000.	0.500	14800.	0.2000E+07
0.00067000	300.	-400.	13000.	6000.	0.462	13743.	0.2000E+07
0.00026000	400.	-400.	14000.	6000.	0.429	12950.	0.1601E+07
0.00004000	500.	-400.	15000.	6000.	0.400	12333.	0.1143E+07
0.00001000	100.	-500.	11000.	5000.	0.455	13567.	0.2000E+07
0.00003000	200.	-500.	12000.	5000.	0.417	12686.	0.2000E+07
0.00017000	300.	-500.	13000.	5000.	0.385	12025.	0.1601E+07
0.00019000	400.	-500.	14000.	5000.	0.357	11511.	0.1143E+07
0.00002000	500.	-500.	15000.	5000.	0.333	11100.	0.8461E+06
0.00001000	200.	-600.	12000.	4000.	0.333	11100.	0.1601E+07
0.00002000	300.	-600.	13000.	4000.	0.308	10689.	0.1143E+07
0.00003000	400.	-600.	14000.	4000.	0.286	10360.	0.8461E+06
0.00004000	500.	-600.	15000.	4000.	0.267	10091.	0.6444E+06
0.00002000	600.	-600.	16000.	4000.	0.250	9867.	0.5025E+06
0.00001000	-200.	-700.	8000.	3000.	0.375	11840.	0.2000E+07
0.00001000	200.	-700.	12000.	3000.	0.250	9867.	0.1143E+07

STRESS-LIFE ANALYSIS OUTPUT

0.00001000	400.	-700.	14000.	3000.	0.214	9418.	0.6444E+06
0.00002000	500.	-700.	15000.	3000.	0.200	9250.	0.5025E+06

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DAMAGE SUM.... 0.7311E-09
LIFE.......... 0.1368E+10 CYCLES
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Strain-Life Fatigue Analysis (TTD), 1984
Association of American Railroads, JC Shang

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