FREIGHT CAR FATIGUE ANALYSIS; GUIDELINES AND APPLICATION

BY

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INTRODUCTION

In recent years, the increase in occurrences of fatigue-induced failures in freight car structures has led to a need for the development of suitable fatigue analysis and prediction tools. In order to fill that need, the Track Train Dynamics Program initiated a project to develop a set of industry guidelines for fatigue analysis of freight car body components. This analysis was intended to complement the existing freight car design specifications [1], and fill an existing gap in the car design process.

The outgrowth of this project was the development of Interim Guidelines for Fatigue Analysis of Freight Cars [2], which have recently been adopted by the Mechanical Division of the Association of American Railroads (AAR) as an Industry Specification (M-1003). These guidelines introduce a fatigue analysis methodology which can be incorporated into the freight car design procedure. Thus, these guidelines represent an analytical tool, with which the freight car design engineer can determine the fatigue life of critical freight car components that are subjected to the fluctuating stresses experienced in railroad service.

It is the purpose of this paper to briefly introduce the methodology presented in the fatigue life analysis guidelines, and to discuss their validation, implementation and limitations.

Fatigue Design in the Railroad Industry

At present, according to AAR Specifications, a 100-ton capacity covered hopper would be designed to resist the following loads:*

Longitudinal Impact	1,250,000 lb.
Longitudinal Squeeze	1,000,000 lb.
Longitudinal Draft (with simultaneous dead and live loads)	±350,000 lb.
Vertical Load at Pulling Face of Coupler	50,000 lb.
Vertical Lift Load at Shank of Coupler	100,000 lb.

Center Plate Reaction $= \frac{\text{(Dead Load + Live Load)}}{2} \times 1.25$

for the following load conditions:

a) Uniform over center plate bearing surface

b) Applied on edge of center plate

c) Applied on side bearing

Jacking on Bolster at Jacking Pad $= \frac{\text{(Dead Load + Live Load)}}{2}$

* In converting these load conditions into allowable design stresses in the car structure, use of design load factors, as defined in [1], are required in the analysis.

Also for a new design, the car may be tested for impact, squeeze, vertical loads on couplers and jacking.

Assuming the car meets or exceeds all the static and impact design and test requirements, recent history has shown that the car may still develop fatigue failures while operating in service.

The fatigue guidelines [2] were developed to give a designer or analyst a technique for incorporating finite life fatigue analysis into his design process.

Some of the benefits that may be expected in the fatigue design of freight cars based on a finite fatigue life are:

1. It can give better assurance that a freight car can achieve its desired design life.

   In the past, failures of freight cars were more often attributed to loading conditions arising from impacts or other severe operating conditions than to service-induced fatigue. The latter had been considered to be one of the least critical conditions. However, with the increasing use of higher strength steels, combined with higher stress levels experienced by the cars, fatigue failures have recently emerged as the cause of the greatest number of failures occurring in service.

2. It may reduce weight and cost.

   At present, some freight car designs are based on stresses not exceeding a given endurance limit, which results in a design that has infinite fatigue life. Designing for a finite life will potentially decrease weight and cost.

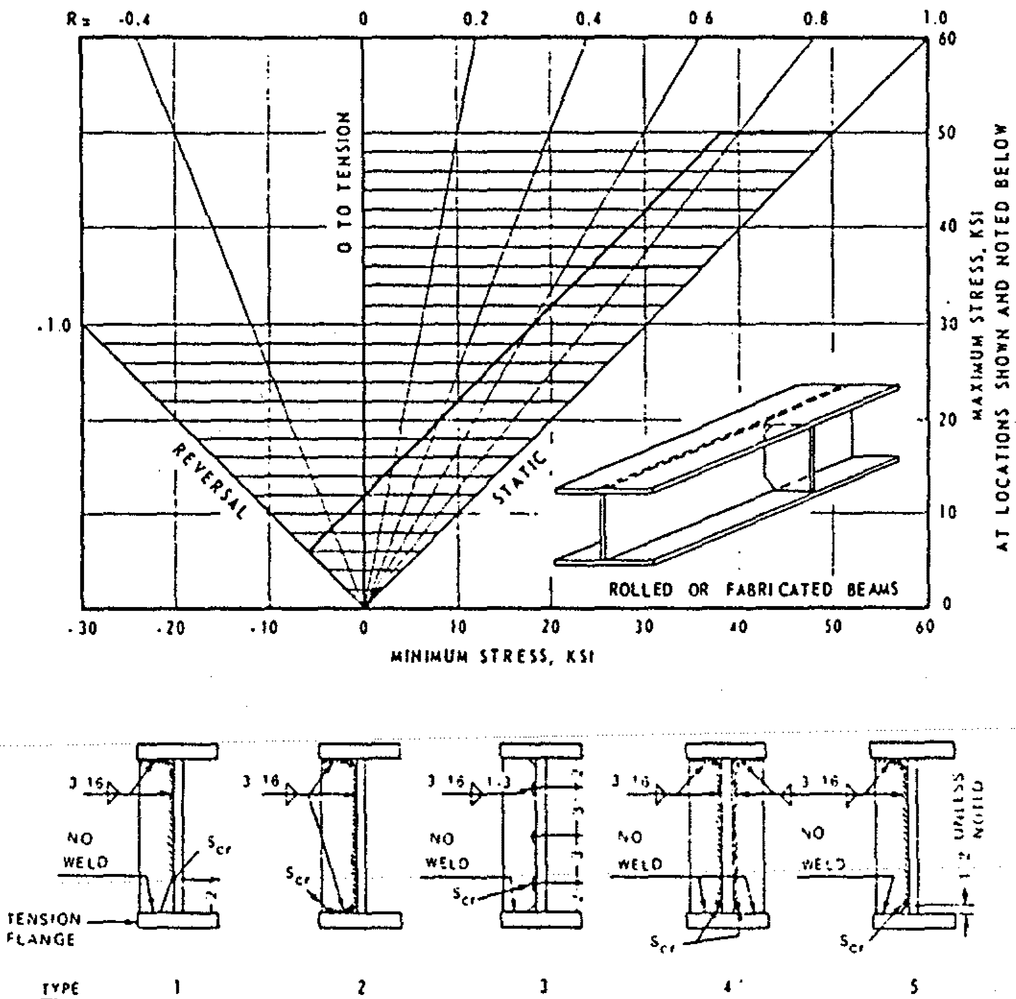
3. It can increase safety.

   Fatigue failures can go unnoticed until the cross section of the member has been reduced to such an extent that brittle or ductile failure occurs, resulting in collapse of the members. Fatigue analysis can be a valuable tool for identifying potential problem areas that should be inspected.

FATIGUE METHODOLOGY

The fatigue analysis methodology presented in the Interim AAR Guidelines [2] is based on Miner's linear cumulative damage rule. This rule is used to calculate fatigue damage in the car structure due to over-the-road service. By use of applicable material and detail characterization, the effect of geometric details is included (Figure 1).

The basic service data is generally in the form of an environmental load spectrum (Figure 2). This spectrum is in fact a three-dimensional folded spectrum which gives, for each maximum-minimum load combination, the percent occurrence of each of these combinations encountered in the environment. The



MAXIMUM (OR CRITICAL) STRESS LOCATIONS VARY WITH THE TYPE OF STIFFENER USED. THE CRITICAL STRESS LOCATIONS, S_{cr}, ARE SHOWN ABOVE FOR TYPICAL CONFIGURATIONS. GENERALLY, THE CRITICAL STRESS LOCATION OCCURS AT THE STIFFENER WELDING CLOSEST TO THE TENSION FLANGE OF THE BEAM.

Figure 1. Modified Goodman Diagram For Beams
With Stiffeners Clipped to Clear Welds

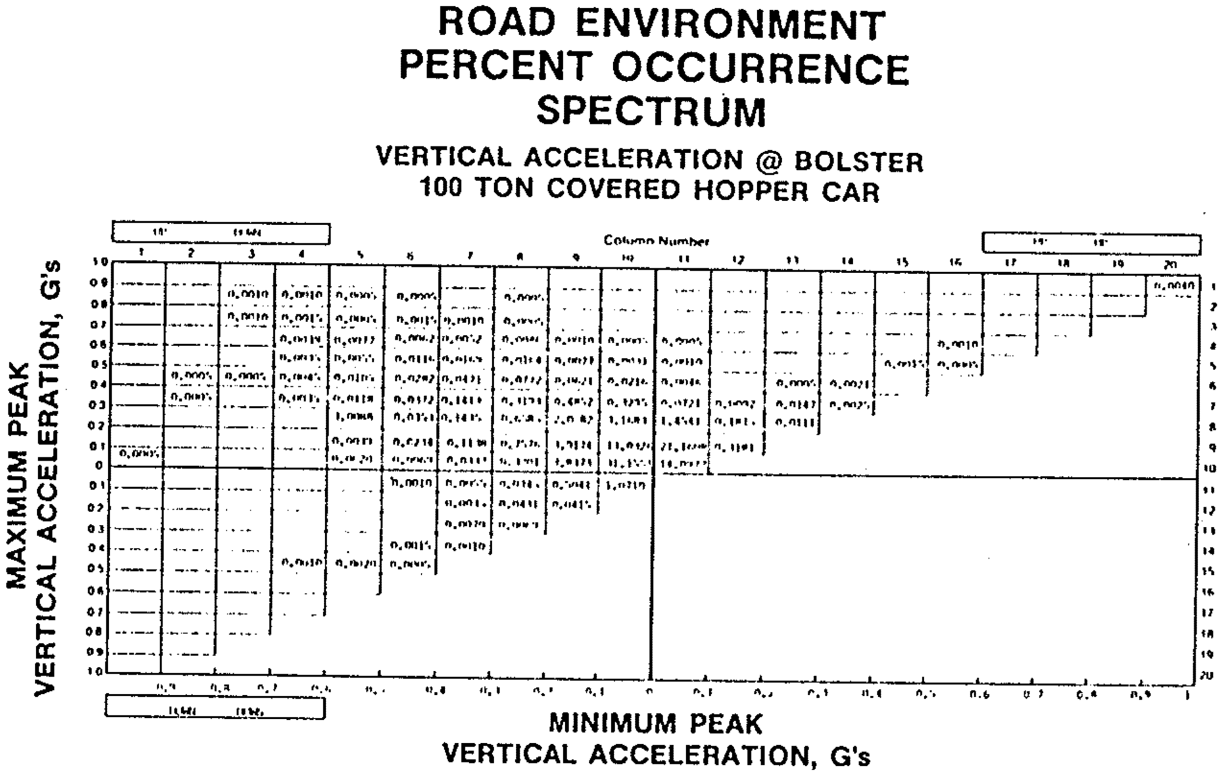


Figure 2. Typical Environmental Load Spectrum

load spectrum can then be converted into stress spectra for critical car locations, by means of structural analysis techniques. Alternately, raw data, such as strains, accelerations, etc., that are obtained from field measurements on instrumented test cars, can be converted from time history form into suitable load spectra by means of the rainflow counting technique [3]. This counting technique gives the user the capability of taking raw field test data and converting it into a form suitable for fatigue analysis input.

A summary of the fatigue life calculation sequence is shown in Figure 3. Using the input data defined in Table 1, the fatigue damage is calculated, and the total number of cycles to failure is determined. The loaded and unloaded fatigue lives are then determined by using the ratios of occurrences (or cycles) per mile and empty-to-loaded car-miles [2]. The results are a predicted fatigue life for the component being analyzed.

The decision, as to which location in the freight car should be subjected to the fatigue life analysis, must be made by the car designer, since it involves considerable engineering judgement.

VALIDATION

How valid is the Fatigue Guideline method? Comparisons have been made between actual service or test failures and the calculated fatigue lives, based on analytical and experimental stresses. These comparisons involved a stub center sill on a tank car, a center sill on an open-top hopper car and a lower bolster web on an open-top hopper car. In the stub sill tank car road and laboratory tests [3], good agreement was obtained between calculated and service failures. Fatigue cracks appeared in service

TABLE I

INPUT DATA

1. Material and Detailed Component Properties
   a. Material Properties
   b. Detailed Component Properties
      1. Modified Goodman Diagram [2]
      2. S-N Curve [2]
2. Load Environmental Spectra [2]
   a. Loaded
   b. Unloaded
3. Conversion of Load Spectra to Stress Spectra
4. Ratio of Empty to Loaded Car-Miles [2]

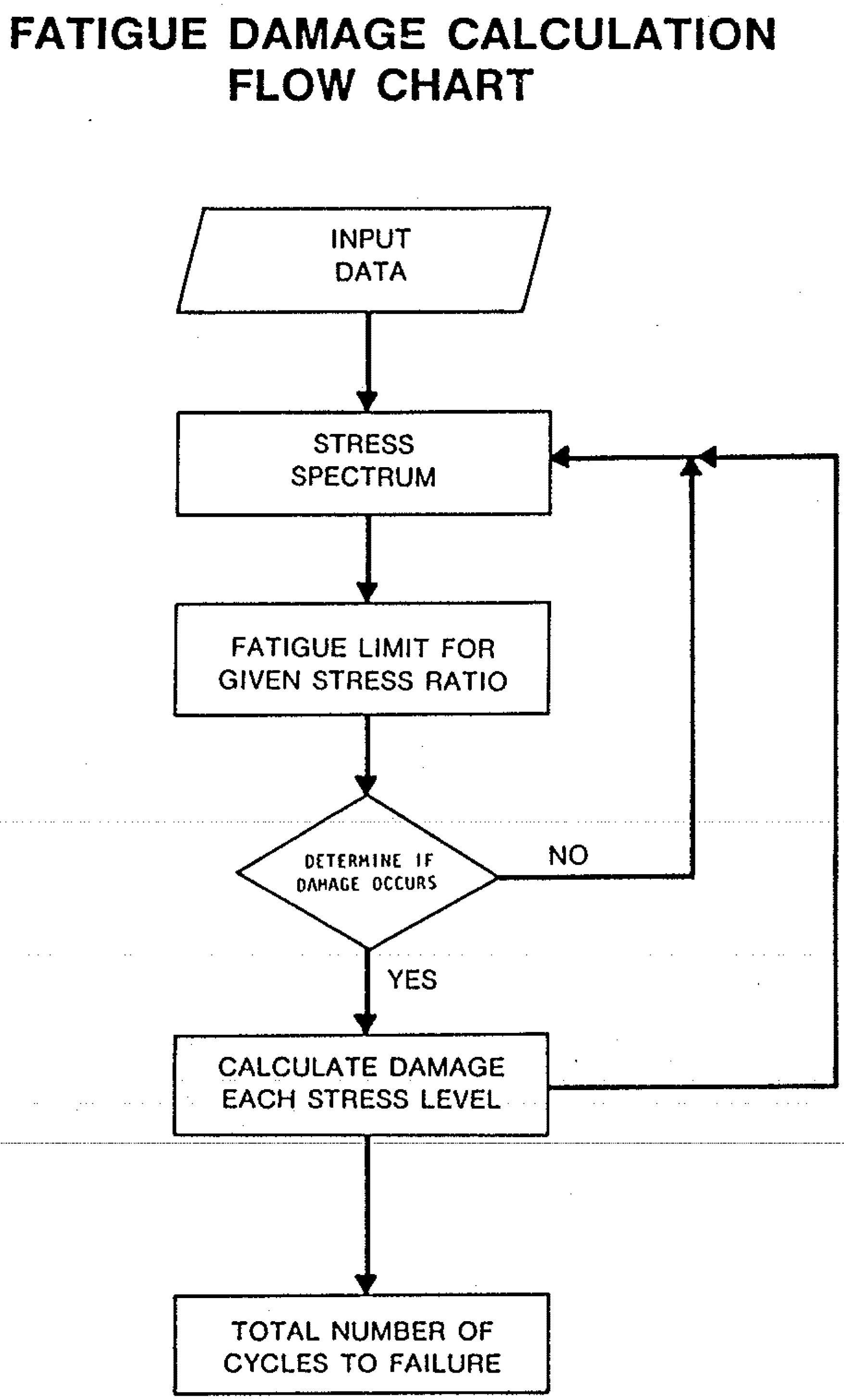


Figure 3. Flow Chart Showing Fatigue Life Damage Calculation Sequence.

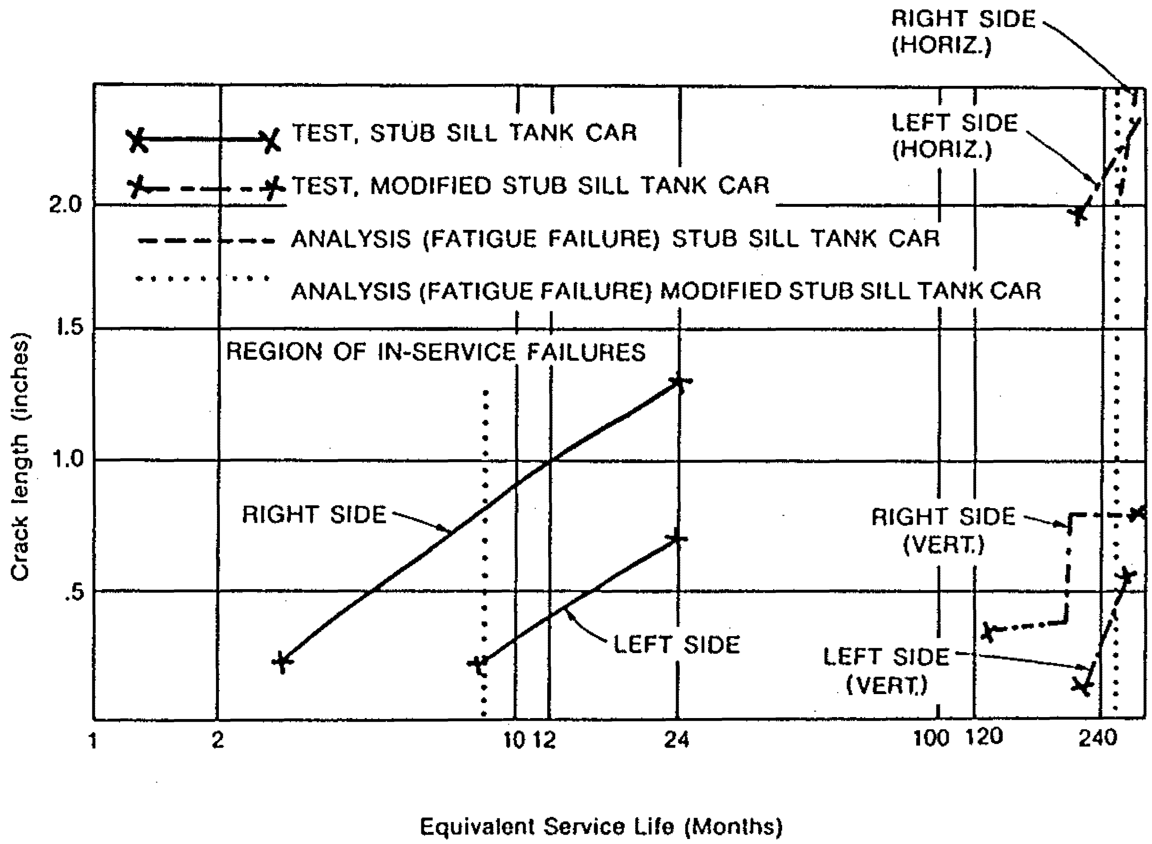


FIG. 4
FATIGUE LIFE OF STUB SILL TANK CAR
Laboratory Test and Analysis Results

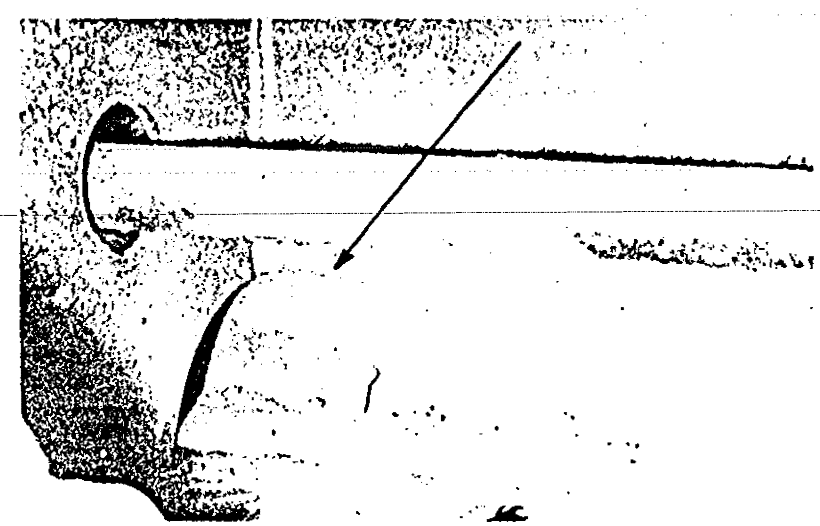


Figure 5. Fatigue Crack in Center Sill of Open-Top Hopper Car involved in FAST Tests.

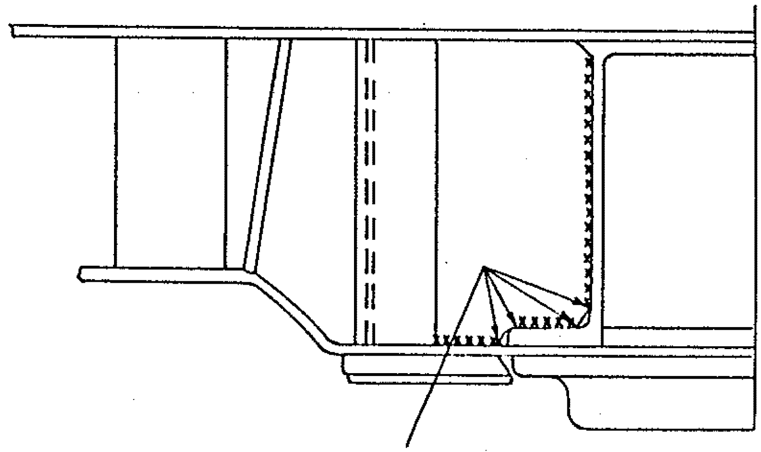


FIGURE 6(a) WELD ATTACHMENT DETAILS FOR BODY BOLSTER-CENTER SILL WEB AREA IN AN OPEN-TOP HOPPER CAR.

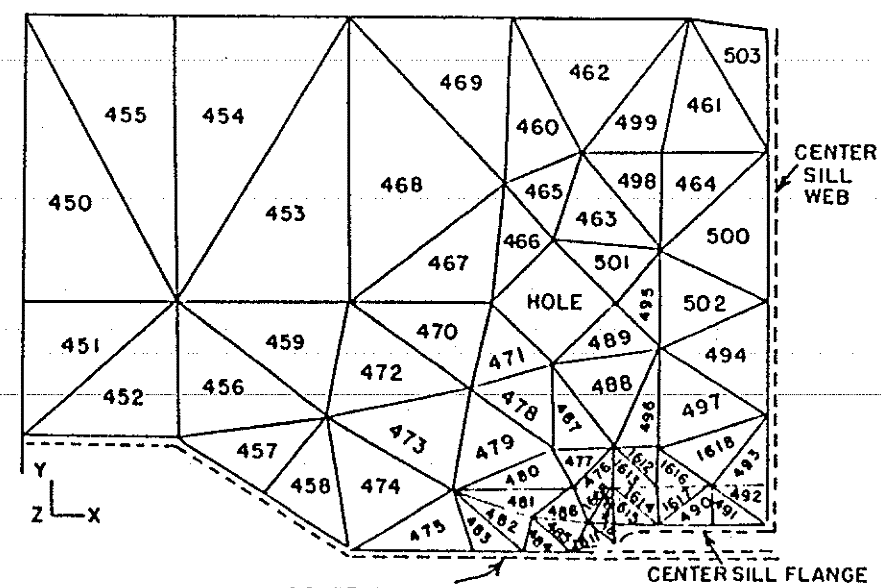


FIGURE 6(b) FINITE ELEMENT ANALYSIS MODEL OF BODY BOLSTER-CENTER SILL WEB AREA IN AN OPEN-TOP HOPPER CAR.

after approximately 2 years (between 50,000 and 100,000 miles), and the calculated life was indicated to be 8½ months (35,000 miles based on 50,000 miles per year) (Figure 4). In 1977, longitudinal cracks in the center sill webs at the bolster webs on open-top hoppers were observed in the FAST tests (Figure 5). Some cracks were found after approximately 23,000 miles of testing. These cars were then instrumented and retested. Based on the test results, the calculated fatigue life was approximately 4,000 miles [4].

In other FAST tests, cracks in the lower bolster web of another 100-ton capacity open-top hopper car were discovered after approximately 60,000 miles. A finite element stress analysis was made to determine the stresses in the critical areas of the bolster (Figure 6). The car was assumed to be rocked over on the edge of the body center plate bowl 50% of the total mileage, and was then subjected to the vertical accelerations given in Table 3.3.2 of the Fatigue Guidelines [2]. The dynamic stresses were assumed to be equal to the static stresses times the vertical accelerations (in g's). The calculated fatigue life was 92,700 miles [5].

IMPLEMENTATION

Fatigue analysis is usually the last step in the design process. Of the many factors which affect fatigue resistance, that of local stress concentration is the most important, and this makes fatigue design more difficult than design for static (or equivalent static) loads.

To implement the guidelines, environmental stresses must be determined. The more accurate the stress data, the more reliable the fatigue analysis. In the initial design stage, it is usually impossible to determine the stresses experimentally. However, to conduct a fatigue analysis of an existing structure, experimentally-determined environmental stress levels may be the most reliable.

To analytically calculate the stresses in a component, the designer must know the environmental loads. The guidelines contain both environmental loads and accelerations for various types of cars. For years, one carbuilder has converted acceleration spectra into equivalent dynamic load spectra. This is approximate, but does give results that are considered fairly reliable by that particular carbuilder.

LIMITATIONS OF METHODOLOGY

In order to properly utilize the fatigue analysis methodology presented in [2], the designer must be aware of its basic limitation.

The cumulative damage technique, utilizing the stress life behaviour of materials, is essentially a high cycle fatigue technique [6]. Thus, for components or materials subjected to more than 100,000 cycles of load application, this technique is considered to be valid. For low cycle fatigue, i.e., when failure occurs in less than 100,000 cycles of load application, the designer must use a specific low cycle fatigue technique, such as the strain life cumulative damage technique described in References [6] and [7]. In this latter technique, it is assumed that stresses exceed the yield strength of the material, and consequently local plastic strain must be considered. In the stress life technique specified in [2], stresses are assumed to be less than the yield strength of the material under analysis.

As noted previously, in order to utilize this fatigue analysis methodology in the design process, i.e., before a prototype is built and can be instrumented, the designer must be able to determine the local stresses in the critical area of investigation, utilizing over-the-road load spectra, such as those given in Reference [2]. This places the burden of developing an adequate stress analysis on the designer. Investigations into the effect of using more sophisticated, and consequently time consuming analyses, such as shown in References [4] and [8], indicate that the designer must be aware of the trade off between accuracy of analysis and the resulting accuracy of the predicted fatigue life.

Furthermore, although the use of the detailed Modified Goodman Diagram (MGD) (Figure 1) enables the designer to avoid development of local stress concentration factors [2], considerable judgement must be utilized in order to properly match the structural detail under analysis with the MGD.

A further limitation in the use of this technique is the difficulty involved in developing a fatigue life prediction for a critical area subjected to different simultaneous loadings, such as combined vertical and longitudinal loadings. Although use of strain gage data avoids this problem of combined loading effects, only limited information and techniques are available for analytically combining the effects of two or more load environment spectra [9].

Finally, it should be noted that, even under ideal analytical conditions, fatigue analysis in general is not <u>exact</u>. When the local stress or strain behaviour is fully defined and understood, and the material properties fully quantified, the analysis will be quite "reasonable." However, under conditions of preliminary design analysis, considerable judgement and experience is required to properly utilize the fatigue life values predicted by this methodology.

FUTURE DIRECTIONS

As noted previously, the fatigue guidelines represent an introduction of finite life analysis techniques to the freight car design process. As such, the adoption of the Interim Guidelines [2] represents a first step in the ultimate incorporation of fatigue analysis techniques. However, it is just the first step. Incorporation of more sophisticated analytical tools, such as strain life analysis [6] and fracture mechanics [7], must be pursued in the future. Adoption of new techniques of testing, such as represented by a new generation of cycle counting instrument packages, is needed. Finally, the industry, both railroads and car manufacturers, must aggressively pursue the latest developments in fatigue analysis and incorporate them into their design procedures, so that this valuable predictive tool can be more fully utilized.

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