PROBABILISTIC FATIGUE SAFETY EVALUATION OF RAILROAD FREIGHT CAR COMPONENTS

J. Mohammadi*, V. K. Garg and N. Subei***

**Department of Civil Engineering, Illinois Institute of Technology, Chicago, IL, USA*
***Department of Mechanical Engineering, University of Maine, Orono, ME, USA*

ABSTRACT

The paper discusses a methodology for the fatigue safety evaluation of railroad freight cars. The data collected in the field for a car under a typical normal operating environment was used in the study. The uncertainty involved in the method of fatigue damage evaluation, method of stress analysis and field data were considered in this study. The fatigue reliability of a given component was calculated using a Weibull probability density function for the fatigue life of the component. The method was applied to a 70-ton box car. The results indicate that the failure probability for certain components is significant.

KEYWORDS

Probabilistic approach, railroad, freight car, Weibull probability density function, fatigue life, box car, safe operation.

INTRODUCTION

Railroad freight cars are quite vulnerable to damage causes by fluctuating stresses in their components. Generally, these fluctuating stresses arise due to track geometry, and in particular from track irregularities [3]. Under normal operating conditions, where the intensity of induced stresses is within allowable limits, the number of cycles of fluctuating stresses plays a major role in causing failure. Each stress cycle produces a certain amount of damage in a given component. This damage accumulates until a fatigue failure takes place. At this stage, the accumulated damage reaches a critical level. Understanding the behavior of a freight car and evaluating this critical level is an essential part of the design for "safe operation" in the freight car industry. Since fatigue failure is ultimately unavoidable, the term "safe operation," as used herein, is only meaningful when a fatigue life is specified for a given component. Furthermore, because of the uncertainty in fluctuating stresses and the mechanical properties of the component, the safety of the component can only be represented on the basis of a percentage probability, by describing the chance that the component will be able to reach its specified fatigue life.

This paper describes a probabilistic approach, which could be used for safety evaluations and fatigue life predictions of freight car components. Various sources of uncertainty in the stresses of a component are described. Actual field data for the forces developed in various components of a freight car, that were collected under normal operating conditions, are used. The formulations are applied for fatigue safety analysis of the major components of a 70-ton box car.

PROBLEM DESCRIPTION

1. Source of Stress Variation in Component: Stresses, induced in a given component of a freight car, show time variations under normal operating conditions. The main source of these variations is track irregularities [3, 5], which result from construction errors, or due to deterioration of the track structure. These irregularities are in the form of gage, alignment, and profile variations, or their combinations. During the operation of the freight car on such a track, the carbody behaves like a dynamic system with several degrees of freedom, each representing a particular type of motion, such as longitudinal, vertical and lateral displacements, or roll, yaw and pitch motions.

Stress evaluations in a given component can be conducted by performing dynamic analyses [3, 5]. These analyses are generally complex and require certain approximations for idealizing the track irregularities. A static analysis can be utilized, provided relationships between carbody forces and track geometry can be established. In this case, the stresses can be determined from the carbody forces by using a conventional static analysis. In the absence of the carbody forces vs. track geometry relationships, field data collected for carbody forces [8] may be used. These data, however, should be analyzed and their statistical variations calculated.

2. Field Data for Carbody Forces: Such data are currently being collected by the Association of American Railroads (AAR) [8]. The forces applied on the bolster, center plate and side bearings of a 70-ton box car, under typical operating conditions, have been collected [9]. The load range for each of the force data shows a variation in the carbody forces. The force data comprises necessary information for stress analyses. However, a statistical analysis of the data is needed before performing such analyses for the force variability and statistical distribution. Typical data for a 70-ton box car are shown in Figs. 1-3. Each figure indicates the percentage occurrence of the load ranges. A typical probability density function to fit the data in each case is also shown. With a 0.05 significance level [2], both beta and Rayleigh distributions fit the data. In Figs. 1-3, the beta distribution is shown for each set of data. The advantage of the beta distribution is its finite "tail."

3. Stress Analysis: The carbody can be analyzed using a conventional stress analysis approach, such as the finite element method. However, the applied forces are random variables, as described by one of the probability density functions shown in Figs. 1-3. If we consider the carbody to be an elastic system, with linear stress-strain material behavior, then the stresses can be calculated in terms of random applied forces. The linear material assumption for the carbody will lead to identical distributions of stresses and strains. In fact, for a beta force distribution, the calculated stresses will also have beta distributions with different parameters of distribution. The objective is to determine a critical random stress, Q, in a given component that causes fatigue failure.

In this study, the stress analysis of the carbody bolster was performed, using a three-dimensional finite element idealization, as shown in Figure 4. The bolster support changes as the car rocks. Therefore, different boundary conditions were considered in the finite element idealization. Figure 4 represents only a typical idealization. Furthermore, because of three-dimensional analysis, each point on the bolster is subjected to multiple stresses and requires a failure criterion for fatigue life evaluation. Generally, failure criteria based on maximum shear stress or maximum effective stress are used [3], however, their applications in this case is complex, since the applied stresses vary randomly. Use of these failure criteria would require additional computations for evaluating the probability of the maximum effective stress or maximum shear stress. For simplicity, the failure criterion, based on principal stress [7], was used in this study. It was assumed that the probability distributions derived for the applied forces can also be used for the critical principal stress, Q.

BASIC FORMULATION

For a given component, the critical stress Q is a random variable, defined with a probability density function f_Q. Using the Miner Cumulative Damage Rule [6], the damage produced in the component, due to one cycle of a stress $Q = q$, is $1/\bar{N}(q)$, where q is a specific value of the random stress Q and $\bar{N}(q)$ is the average number of cycles to failure, if the stress is q. For M_i cycles of q_i, the damage is $M_i/\bar{N}(q_i)$, and thus the total damage d will be:

$$d = \sum_{i=1}^{k} M_i/\bar{N}(q_i) \qquad i = 1,2,3, --, k \tag{1}$$

Since M_i is a random variable, d will also be a random variable and the expected damage is:

$$E(d) = E[\sum_{i=1}^{k} M_i/\bar{N}(q_i)] = \sum_{i=1}^{k} \bar{M}_i/\bar{N}(q_i) \tag{2}$$

where $\bar{M}_i$ is the expected value of M_i. The critical stress of the component is a continuous random variable. If $\bar{n}$ is the average number of cycles of the random stress Q to failure, then for a particualr stress, such as q, M is $\bar{n}\ f_Q$. Theoretically, Q may have any value from 0 to ∞. Equation (2) becomes [1]:

$$E(d) = \int_0^\infty [\bar{n}\ f_Q(q)/\bar{N}(q)]\ dq \tag{3}$$

Fatigue failure occurs when E(d) approaches unity. Thus at failure, $\bar{n}$ from Eq. (3) is:

$$\bar{n} = 1/[\int_0^\infty f_Q(q)/\bar{N}(q)]\ dq \tag{4}$$

Since $\bar{N}(q)$ is the average number of cycles to failure for $Q = q$, $\bar{N}(q)$ may be obtained from the stress vs. number of cycles relationship. If this relationship is written as:

$$\bar{N} = c\,q^{-b}$$

Substituting Eq. (5) into Eq. (4), we get:

$$\bar{n} = c/\int_0^\infty q^b f_Q(q)\,dq \qquad (6)$$

where c and b are constants. Eq. (6) gives the average number of cycles of random stress Q to failure (i.e., the expected fatigue life). A similar relationship was also obtained by Mohammadi and Garg [5] for a freight car dynamic system under stationary Gaussian random excitation.

The expected fatigue life $\bar{n}$ may be used to determine the failure probability of a component for a desired life of n. In fact, the fatigue life n of a given component is a random variable with an expectation $\bar{n}$. Using a Weibull probability density function for n, the probability that the component will survive n cycles of random stress Q is [1]:

$$L(n) = \exp\{-[(n/\bar{n})\ \Gamma(1+\delta)]^{1/\delta}\}$$

where L(n) is the probability of survival for n cycles, or the reliability for a fatigue life of n, Γ is the gamma function and $\delta = \Omega_N^{1.08}$, in which Ω_N is the uncertainty in predicting fatigue life of the component. According to Ang and Munse [1], this uncertainty depends upon: (i) the uncertainty in the stress vs. number of cycles relationship, (ii) error in the Miner Damage Rule and the variation in applied stress. In the case of a freight car Ω_N also depends upon the uncertainty in the finite element idealization and failure criterion used in the fatigue failure analysis.

NUMERICAL ILLUSTRATION

Figure 4 shows the finite element idealization of the body bolster for a 70 ton box car [8]. The data shown in Fig. 1-3 were used for a fatigue analysis of this structure. After performing the structural analysis, the critical stresses were calculated, using principal stress failure theory. The probability distribution of a critical stress was taken as beta, whose parameters were calculated from the parameters of the applied load distribution. The fatigue safety analysis was then performed for a number of standard details [1], with or without welds. For each detail an appropriate stress vs. number of cycles equation [1] was used. Furthermore, the uncertainty Ω_N, for each detail was calculated from the information given in Reference [1]. In Ω_N, the uncertainty in the method of stress analysis and failure theory was also considered. The expected fatigue life for a cover plate weld was 3.618 x 10^9 cycles. This gives a fatigue reliability of 0.9998 (i.e., probability of failure of 2.0 x 10^{-4}) for a performance life of 2 x 10^6 cycles. For a simple butt weld detail, the expected fatigue life was 1.08 x 10^9 cycles and indicated a fatigue reliability of 0.9993 (i.e., probability of fatigue failure of 7 x 10^{-4}) for 2 x 10^6 cycles. The results and the corresponding details are summarized in Table 1.

SUMMARY AND CONCLUSIONS

A method for the safety analysis of railroad freight cars for fatigue failure was presented. The method used data collected in the field for the car to determine its performance, in terms of the fatigue reliabilities of the components. For a 70-ton box car, the analysis showed that, for 2 x 10^6 cycles, the fatigue failure probability may be as high as 7 x 10^{-4}.

Although this number may appear to be insignificant, it is considered to be critical in a safety analysis. This number represents a chance of failure for one component only. The contribution of such small probabilities from several components of a car could ultimately add up to a significant level. Furthermore, the failure probabilities calculated herein may be used in design processes, based on the designer's intuitive judgements. If the number, (i.e., 7 failures in 10,000 identical components), in the judgement of the design engineer, is considered to be high, a revision in design would then be necessary.

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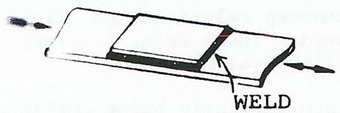
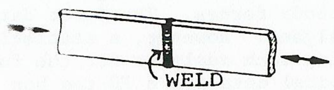
TABLE 1. NUMERICAL ILLUSTRATION

Detail	Expected Life $\bar{n}$ (Cycles)	Probability Failure For $2\mathrm{x}10^6$ Cycles
	$3.618\mathrm{x}10^9$	$2.0\mathrm{x}10^{-4}$
	$1.08\mathrm{x}10^9$	$7.0\mathrm{x}10^{-4}$

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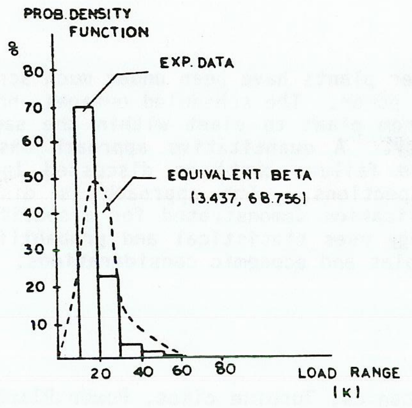

Fig. 1 Bolster Load Data

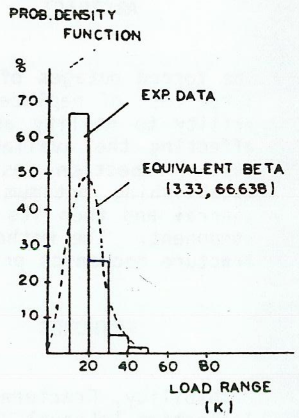

Fig. 2 Center Plate

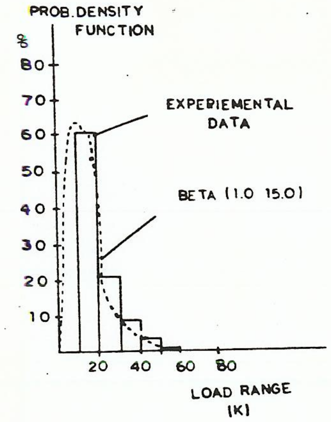

Fig. 3 Side Bearing

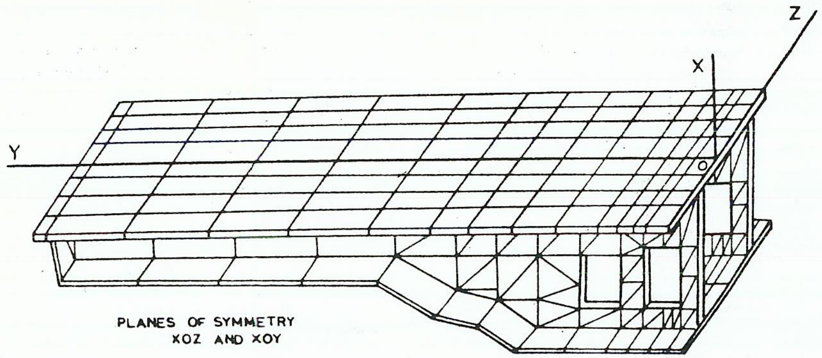

Fig. 4 A Typical Finite Element Idealization