Evaluation of Occupied Volume Integrity (OVI) In Passenger Railroad Equipment

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ABSTRACT

This research program was sponsored by the Federal Railroad Administration (FRA) Office of Research and Development in support of the advancement of improved safety standards for passenger rail vehicles. FRA and the Volpe National Transportation Systems Center (Volpe Center) have conducted research into alternative methods of evaluating occupied volume integrity (OVI) in passenger railcars. This research has led to the development of a proposed alternative strategy for evaluating OVI that moves the applied loads from the line of draft to the collision load path. This alternative methodology establishes three evaluation load levels and a corresponding set of pass/fail criteria for each level. Additionally, the proposed methodology permits a combination of testing and analysis, whereas the conventional methodology only permits testing.

A program of compression testing and finite element (FE) analyses was used to examine the efficacy of applying this proposed methodology to evaluate a passenger railcar’s OVI. The conventional 800,000 pound buff strength test was performed on a passenger railcar both to verify its structural integrity and to assist in the validation of an FE model. The validated FE model was then used to simulate loading of the same railcar to its ultimate, or crippling, load. Finally, two passenger railcars were tested to crippling. The crippling test results, including the load-deflection behavior and crippling load magnitude, were compared to the analysis predictions and found to be in good agreement. This testing and analysis program has helped establish a technical basis for the proposed alternative OVI requirements and methodology.
INTRODUCTION

This research program was sponsored by the FRA Office of Research and Development in support of the advancement of improved safety standards for passenger rail vehicles. Since 2006, FRA, with the assistance of the Volpe Center, has been conducting research on alternative methods for evaluating OVI (1, 2, 3, and 4). In 2010, the Engineering Task Force (ETF) of the Railroad Safety Advisory Committee (RSAC) used the results of this ongoing research to help develop alternative criteria and procedures for assuring OVI (5). An OVI requirement is related to a passenger railcar structure’s ability to resist intrusion into the occupied space under load conditions that place a great demand on the occupant volume structure. The current method of evaluating OVI requires a passenger railcar to sustain no permanent deformation when subject to an 800,000 pound compressive load along its line of draft (6). This requirement establishes a baseline level of OVI for a railcar when loaded along its line of draft. However, as not all passenger railcars are designed to transmit collision loads along the line of draft, alternative methods of evaluating OVI are necessary. The research program has examined the load path through the occupant volume as well as alternative load paths that could be used to ensure a level of safety equivalent to that of the current requirement. The ETF’s criteria and procedures allow flexibility in evaluating a wider variety of passenger railcar designs while maintaining an equivalent level of OVI to equipment evaluated using the conventional methodology. The full set of criteria and procedures are intended to streamline the process of presenting technical information needed to determine equivalent safety between alternatively-designed and fully-compliant passenger equipment in support of a request for a waiver of certain current regulations.

Criteria and Procedures

The ETF adopted alternative criteria and procedures for demonstrating that a vehicle’s OVI is equivalent to that of a railcar that complies with the existing 800,000 pound (3.6 MN) regulation at 49 CFR 238.203. In practice, the ETF’s criteria and procedures may be applied to a railcar design for which a waiver of the existing regulation is sought. While the railcar design may not comply with the regulation itself, the ETF criteria and procedures can be used to demonstrate that the design possesses a level of safety equivalent to a compliant railcar. The criteria and procedures for assessing OVI are designed to work with the other criteria and procedures developed by the ETF to ensure overall occupant safety. The OVI criteria and procedures adopted by the ETF are for a quasi-static evaluation of OVI that may be performed using a combination of testing and a properly-validated analysis. The ETF’s criteria and procedures for evaluating OVI are conceptually similar to the existing 800,000 pound (3.6 MN) requirement, in that both methodologies use a quasi-static load to evaluate a railcar’s ability to maintain its structural integrity during a collision. The full details of the OVI criteria and procedures may be found in the ETF’s report (5).

The ETF methodology for the quasi-static evaluation of OVI does make several changes to the current practice. Recognizing that different design approaches to structural crashworthiness may provide an equivalent level of occupant protection, the ETF methodology establishes three evaluation load levels and a set of pass/fail criteria corresponding to each level. A railcar need meet only one of the three options when seeking a waiver of the regulation, or the regulation itself. The options are shown in TABLE 1.
### TABLE 1 ETF Quasi-static Load Criteria Options

<table>
<thead>
<tr>
<th>ETF Option</th>
<th>Criteria for Each Option</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Option A</strong></td>
<td>Passenger equipment shall resist a minimum quasi-static end load of 800,000 pounds (3.6 MN) applied on the collision load path without permanent deformation of the occupied volume.</td>
</tr>
<tr>
<td><strong>Option B</strong></td>
<td>Passenger equipment shall resist a minimum quasi-static end load of 1,000,000 pounds (4.4 MN) applied on the collision load path with limited permanent deformation of the occupied volume.</td>
</tr>
<tr>
<td><strong>Option C</strong></td>
<td>Passenger equipment shall resist a minimum quasi-static end load of 1,200,000 pounds (5.3 MN) applied on the collision load path without crippling the body structure.</td>
</tr>
</tbody>
</table>

The evaluation procedures that may be used with any of the three options include several departures from the current regulation’s procedures. The full set of procedures and guidelines for applying each of the three options are provided in the ETF’s report (5). For each option, the load is applied along the collision load path, while the regulation requires the load to be applied on the line of draft. The collision load path is determined for each railcar design undergoing evaluation based upon the particular structure of each design. Loading along the collision load path more closely approximates the severe load environment of a collision than loading along the line of draft. Placing evaluation loads along the collision load path evaluates the OVI in the manner in which each individual railcar was designed to support loads.

The second departure from the conventional procedures is permitting a combination of full-scale testing and analysis to be used to evaluate the OVI of a railcar. The existing 800,000 pound (3.6 MN) load requirement was intended to represent a non-destructive test, as it was to result in no permanent deformation. Since this requirement was first applied, analytical techniques have developed that enable the response of the railcar to loads such as this to be modeled with a high level of confidence. While the ETF procedures do not restrict the use of testing alone to demonstrate that a design meets any of the options, Option B and Option C permit permanent deformation in the railcar’s structure in a successful evaluation. Because testing to either of these points would be a destructive test, the combination of elastic testing and plastic analysis will likely be the method of evaluation for the majority of the applications of the ETF methodology.

Before an analytical model can be used to evaluate any of the three options, the model must be validated with test data. An elastic test of the passenger railcar performed in accordance with a recognized design standard and with a load of at least 337,000 pounds (1.5 MN) along its service load path can be used to validate the model. The model would be used to simulate this load condition, and the results of test and analysis compared. The results from the model must compare closely with the results of the test (5). Once the model has been validated, it can be used to extrapolate the response of the railcar to greater loads under which the structure may permanently deform.

### QUASI-STATIC TESTING

The goal of this research program was to evaluate the efficacy of using a validated model to demonstrate a passenger railcar design’s OVI. The approach used in this research was to apply the ETF’s OVI criteria and procedures to an existing passenger railcar. Exceeding the ETF requirements, this passenger railcar would also be tested to its crippling point. The results of the...
Applying the ETF procedures, the response of the railcar in both the validation test and the validation analysis should remain elastic. The test will be a non-destructive test if the test article remains elastic. If the analysis results are in satisfactory agreement with the results of the test, the model is considered validated and can then be used to extrapolate the response of the railcar to the load required by any of the three options.

Because Option B and Option C both permit a model to undergo permanent deformation without failing the evaluation, the model must be capable of capturing both the elastic and plastic behavior of the railcar undergoing evaluation. Plastic analysis is more complex than elastic analysis. It is possible to build a model that predicts the elastic response of the carbody closely but does not predict the plastic behavior of the carbody as well. The current OVI research program has included a series of tests and analyses to determine what practices are necessary to assure a high level of confidence in the model predictions for the carbody response to high loads up to and including crippling.

The passenger railcar design chosen for this research program was the Budd Pioneer. This design is a conventional, single-level passenger railcar with an 800,000 pound (3.6 MN) compliant underframe. As part of a previous FRA research program, the ends of two railcars of this design were modified to include crash energy management (CEM) components (7). These CEM components included multiple energy-absorbing elements, designed to dissipate collision energy in a controlled manner. The CEM railcars were then used in a series of dynamic impact tests (8, 9). FIGURE 1 shows Pioneer Car 244 with CEM components installed at each end of the car.

FIGURE 1 Pioneer car 244.

The CEM-equipped Pioneer car was chosen for the OVI research program for several reasons. Because the car was originally constructed to support the 800,000 pound (3.6 MN) line of draft load, this car design represents a structure that complies with the existing regulation. The addition of the CEM structures at both ends of the car move the collision loads away from the line of draft. The CEM structures permit evaluation loads to be introduced into the body structure at locations other than the line of draft for the purpose of assessing the ETF’s OVI methodology. Using this type of car for testing allows the ETF methodology of loading along the collision load path to be applied to a car designed to meet the conventional line of draft load requirement.

Two loading conditions were analyzed and tested. The first condition was the traditional 800,000 pound (3.6 MN) elastic load applied to the buff stops, along the line of draft. The second condition was a crippling load applied to the energy absorber supports, along the...
collision load path. The primary result from the analyses and tests was the calculated or measured reduction of the car’s length for a given applied load.

**Condition 1 – 800,000 Pounds along Line of Draft**

A conventional 800,000 pound (3.6 MN) line of draft compression load test was conducted on January 19, 2011 at the Transportation Technology Center in Pueblo, Colorado (4). As Car 244 was originally constructed to meet the 800,000 pound (3.6 MN) compressive load requirement, this load was selected to verify the car’s structural integrity. Pioneer Car 244 was instrumented with strain gages and displacement transducers, and a load cell measured the load applied to the line of draft. The car was loaded in 200,000 pound (0.9 MN) increments, up to a total load of 800,000 pounds (3.6 MN). This test was also simulated using FE analysis, and the test’s results were used to validate the model. The ETF’s procedures for validating a model call for an elastic test to be performed according to a recognized national or international standard with a minimum compressive load of 337,000 pounds (1.5 MN) applied to the railcar. The 800,000 (3.6 MN) line of draft load meets the criteria for a suitable validation test, as defined by the ETF.

The FE model of Pioneer 244 that was used to simulate this test is shown in **FIGURE 2**. This model was originally created for a previous research program’s first full-scale impact test (10), was modified to include the crush-zones in support of the CEM impact tests (11), and has been further refined for the quasi-static OVI tests (4). The full length of the car was modeled, but only one half of the width was modeled. A symmetry boundary condition was applied to represent the other half of the car. This simplification is acceptable for this particular design, as the railcar does not have any large asymmetries between its left and right sides.

**FIGURE 2 FE model of Pioneer 244 in 800,000 pound analysis.**

The key results that were compared between the model and the test were the longitudinal force versus displacement behavior as well as the vertical displacements of the car at various applied loads. The ETF report provides target tolerances for comparing measurements between tests and analyses; those target values were used in this research (5). For the longitudinal force versus displacement behavior, the tolerance for successful validation is an analytical response that is within +/- 10% of the test behavior. The test and analysis force-displacement behaviors are plotted in **FIGURE 3**. The horizontal axis in this figure indicates the reduction in the length of the car, which is the difference between the displacement at the live (loaded) end and the rear (restraint) end of the car. A +/- 10% tolerance on the test data is also shown in this figure.

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Vertical displacement measurements were taken at several locations along the length of the underframe of the car in the test. Results were recorded at corresponding locations within the FE model. Vertical deflection results are plotted in FIGURE 4 for the deflection of the underframe at a point at the center of the car in both the test and the analysis for different load values. The target tolerance of +/- 10% of the test deflection is also plotted in this figure.

![FIGURE 3 800,000 pound test and analysis force-displacement results.](image1)

![FIGURE 4 Vertical displacement at center of car 244 in 800,000 pound test and analysis.](image2)
FIGURE 3 and FIGURE 4 both exhibit oscillations in their respective FE analysis results. These oscillations are due to the analysis being executed with a slowly-applied dynamic load as an approximation of a quasi-static loading rate. After comparing the test data and the data collected from the analysis the model was found to be in sufficient agreement with the test to be considered validated. The comparison of test data to the analytical results is discussed in greater detail in Reference 4.

Condition 2 – Crippling

The second type of test performed in this research program was a quasi-static loading of a passenger railcar to its crippling load. Two Pioneer cars were loaded to crippling: Car 248 and Car 244. In accordance with the ETF procedures, the load was placed along the collision load path for both crippling tests. Because the Pioneer cars used in this program included CEM components, collision loads are transmitted through the energy-absorbing elements to the occupant volume through the energy-absorber support structures at the ends of the occupant volume. In order to apply loads to these structures, the end frames of the cars had to be removed. Pioneer 248 is shown in FIGURE 5 with its end frames removed and with its energy absorber support locations highlighted. During the test, the car was restrained at the opposite end through the corresponding structures.

![FIGURE 5 Pioneer 248 with end frames removed.](image)

Based on the successful validation of the FE model through the use of elastic test data, the model that was used to simulate the 800,000 pound (3.6 MN) compression test was also used to simulate the crippling tests. The model was modified to be used in the crippling analysis. Since Car 244 and Car 248 had their end frame structures removed to provide access to loading and restraining locations along the collision load path, the corresponding structures were removed from the model as well. Additionally, because the car was loaded independently on the left and right sides in the test, the full-width of the car was modeled. The FE model as employed in the crippling simulations is shown in FIGURE 6. Loading of the FE model was accomplished.
by displacing each energy-absorber support plate on the live end and calculating the force necessary to achieve that displacement. The displacement on the live end was increased until a maximum force was achieved. This maximum load is defined as the crippling load.

FIGURE 6 FE model used to simulate crippling load.

Car 248 was tested on June 15, 2011 at the Transportation Technology Center. This car was tested as part of the “shakedown” of a newly-installed load frame and hydraulic control system. As such, limited instrumentation was installed for this test. The force at each loading location, the displacement at each loading location, and the force at each restraint location were measured. The control system used in this test maintained an equal displacement for each of the four hydraulic actuators used to load the car. This control system allowed each actuator to apply a different level of force necessary to maintain the target displacement. The displacement in each actuator was slowly increased until the load reached a global maximum, followed by a sudden drop.

The fully-instrumented crippling test of Car 244 was performed on June 21, 2011. Strain gages and displacement transducers were placed on the car in the same locations as had been used in the 800,000 pound (3.6 MN) elastic test of Car 244. Additional displacement transducers were added to record the crippling behavior of the underframe. The forces at each loading and reaction point on the car were also recorded in this test. Car 244 was loaded quasi-statically until its crippling load was reached using similar procedures to the test of Car 248.

The overall force-displacement characteristics from both crippling tests and the FE analysis are plotted in FIGURE 7. The force-displacement characteristics plotted in this figure are obtained by adding the four live-end load measurements to obtain the total load applied to the car. The displacement measurements used for the horizontal axis of this plot represent the extension of the hydraulic actuators used to apply load at the live end of the car.
FIGURE 7 Predicted and measured force-displacement characteristics for crippling.

The force-displacement results of both tests and the analysis are in agreement. In all three cases the force-displacement results agree closely for loading up to more than 800,000 pounds (3.6 MN), the load required for evaluation of Option A. Each test or analysis response is approximately linear in this region, and all three have approximately the same slope.

Beyond 800,000 pounds all three results exhibit the same types of behavior, but at different values of displacement. In both tests and the analysis the roof buckled at a load of approximately 1 million pounds. The total applied load decreased as the roof’s ability to support loads was overcome. Because the underframe had not exhausted its load-carrying capacity, the load began to increase again as the actuator displacement continued to increase.

Following roof buckling each of the three results again exhibit similar slopes. The load continued to increase until the underframe reached crippling. Car 248 had a measured crippling load of 1.15 million pounds (5.1 MN) and car 244 had a crippling load of 1.19 million pounds (5.3 MN). The pre-test simulation calculated a crippling load of 1.19 million pounds (5.3 MN) for cars of this design.

While the elastic behavior and crippling loads closely match between the model and the two tests, there are variations in the displacements at which buckling and crippling occur in both tests and in the analysis. Car 248 and Car 244 each experienced a different crippling mode. The crippling mode predicted by the analysis and the crippling mode observed in the test of Car 248 match more closely.

The crippled carbody from the analysis result is shown in FIGURE 8. In this analysis, the roof and sidewall buckle at a lower load than the underframe structure. The large-scale buckle in the sidewall and roof of the car occurs in the vicinity of the seventh and eighth windows from the left end of the car in this figure. This area is just beyond halfway down the length of the car.
FIGURE 8  Predicted carbody deformation.

FIGURE 9 shows Car 248 following its crippling test. A large buckle in the sidewall and roof can be observed toward the right end of the seventh window from the left. This buckle is slightly closer to the live end than was observed in the FE model. The buckle in Car 248 also has a shorter wavelength than the buckle in the pre-test FE model. Prior to the test of Car 248, damage was noted in the floor and wall structures in the area that eventually buckled during the test.

FIGURE 9  Carbody deformation in car 248.

Overall, Car 244 behaved similarly to Car 248 and the FE model. The sidewall and roof structures buckled at a lower load than the underframe. The load continued to increase in all three cases until the underframe buckled. FIGURE 10 shows the deformation of Car 244 before its removal from the test frame. The sidewall deformation occurred between windows 3 and 4, closer to the live end than the deformation predicted by the model and the deformation observed in the test of Car 248.
Because both Car 244 and Car 248 had been previously involved in a research program of dynamic impact testing, the differences in crippling locations may be attributed to undetected localized areas of existing damage in the cars. As seen in FIGURE 7, these variations in the locations of deformation do not have a large influence on the magnitude of the crippling load. For both cars tested and for the model, the damage occurred in the same sequence: buckling of the sidewall and roof followed by buckling of the underframe at a larger load. While the permanent deformation occurred at different locations, the crippling loads in both tests and in the analysis are in close agreement.

CONCLUSIONS

FRA and the Volpe Center have conducted research on alternative methods of evaluating OVI in passenger railcars. This research has helped to develop a new methodology for evaluating OVI: evaluation of the occupant volume response to loads along the collision load path through a combination of elastic testing and properly-validated analysis. This methodology has been adopted by the RSAC’s ETF to assess the OVI of passenger equipment that may not comply with the current regulations.

This methodology attempts to update the conventional approach to OVI currently in effect, the 800,000 pound elastic buff strength requirement, while maintaining an equivalent level of safety. The proposed methodology attempts to better connect the OVI requirement with the loads experienced by the railcar in a collision by moving the evaluation load from the line of draft to the collision load path, which may not coincide with the line of draft. The proposed methodology also permits generalization of the evaluation to a wider variety of equipment both by moving the load to the collision load path and by presenting three load magnitudes and corresponding pass/fail criteria to better reflect different railcar design methodologies. Finally, the proposed evaluation methodology updates the requirements for demonstrating a car design’s OVI by permitting elastic testing and plastic analysis to be used where elastic testing alone was used in the past.
As part of the current research program on OVI, a series of full-scale tests was performed in parallel with detailed computer simulations to examine the efficacy of the proposed criteria and procedures. The ETF procedures for performing an elastic analysis, validating this analysis with test data, and using the validated model to predict the plastic and crippling behavior were followed. As an examination of the efficacy of using analysis to evaluate OVI, two crippling tests were carried out and the results were compared to the results of the crippling analysis.

The results of the crippling tests and analysis show that alternative OVI requirements are an effective alternative to the traditional requirement in assuring the occupant volume integrity of rail passenger equipment. The crippling tests of Budd Pioneer 244 and 248 resulted in damage to the railcars in different locations but at nearly the same crippling load. The analysis resulted in a crippling load that was nearly the same as that of the tested cars, and a similar buckling mode as one of the tested cars. This testing and analysis program has successfully established a technical basis for the proposed alternative OVI requirements and methodology.

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REFERENCES


