APTA STANDARDS DEVELOPMENT PROGRAM

RECOMMENDED PRACTICE

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Truck Design

Abstract: This recommended practice gives guidance for the design of railroad passenger trucks and for the application of existing truck designs that differ from the prior application(s) in terms of load, speed, track geometry, wheel profile, carbody and combinations of these factors.

Keywords: suspension system, truck design

Summary: The second revision of this recommended practice completely revises all previously existing sections and adds several new ones. Sections related to vehicle dynamic performance have been removed in lieu of updated guidance in 49 CFR 213 and a forthcoming APTA vehicle dynamics performance standard. This document gives guidance for the design of railroad passenger trucks and for the application of existing truck designs that differ from the prior application(s).

Scope and purpose: The recommended practice is applicable to all types of trucks used in regular passenger service, contracted one year or more after publication date. This includes trucks fitted to locomotive-hauled cab and trailer cars, MU cars and non-passenger-carrying cars and locomotives that are intended for use in passenger service. The truck designer should consider and satisfy all safety-related issues, including those from all applicable APTA standards and CFR requirements. The specific details of these safety-related issues should be identified in the specification by the purchasing railroad and/or the prime contractor. In addition to safety-related issues, there are other specification issues related to passenger comfort, wayside clearance, maintainability, reliability, quality control, interchangeability, etc., which are unique to each railroad and outside the scope of this document. The purchasing railroad should keep all truck design documentation as outlined in this recommended practice for the lifetime of the vehicle. This recommended practice is not applicable for requalification of trucks for life extension or transfer of equipment between purchasing railroads.

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Participants

The American Public Transportation Association greatly appreciates the contributions of the **Truck Design Recommended Practice Task Group** of the APTA Commuter, Intercity and High-speed Rail Mechanical Working Group, which provided the primary effort in the drafting of this document.

At the time this standard was completed, the task group included the following members:

Luke Morscheck, Hatch LTK, *Task Group Lead*
Mark Stewart, Hatch LTK, *Task Group Co-Lead*

Michael Barnes, *Jacobs (CH2M)*
Al Bieber, *STV Inc.*
Sylvain Boily, *Bombardier Transportation*
Glenn Brandimarte, *ORX Rail*
Steve Cavanaugh, *Metrolinx (GO Transit)*
Steve Chrismer, *Amtrak*
Dion Church, *SNC-Lavalin Inc.*
Richard Curtis, *Curtis Engineering Consulting*
K'Moy Daye, *MTA Metro-North Railroad*
Jacques Desjardins, *Bombardier Transportation*
Ed Gacsi, *New Jersey Transit*
Guillaume Ham-Livet, *ALSTOM Transport*
Jason Hesse, *STV Inc.*
Lew Hoens, *MTA Metro-North Railroad*
Joe Kenas, *Bombardier Transportation*
Bernhard Kittinger, *Siemens Mobility, Inc.*
Peter Klauser, *Vehicle Dynamics Group*
Heinz-Peter Kotz, *Siemens Mobility, Inc.*
Marie LaPosta, *Amtrak*
Daniel Luskin, *Amtrak*
Frank Maldari, *MTA Long Island Rail Road*
Brian Marquis, *Volpe Natl. Trans. Systs. Center*
Francis Mascarenhas, *Metra*
Eloy Munoz, *Hatch LTK*
Joe Patterson, *Amsted Rail*
Wolf Reimann, *retired (Bradken)*
Ralph Schorr, *Amsted Rail*
Martin Schroeder, *Jacobs (CH2M)*
Melissa Shurland, *FRA R&D*
Dave Skillman, *Amtrak*
Monique Stewart, *FRA R&D*
Lukasz Szymsiak, *VIA Rail Canada, Inc.*
Ali Tajaddini, *Federal Railroad Administration*
Jason Taylor, *Amsted Rail*
Jason Thomas, *Metra*
Eric Touzin, *Bombardier Transportation*
Mike Trosino, *Amtrak*
Travis Unzicker, *Bradken*
Rudy Vazquez, *Amtrak*
Brian Whitten, *SNC-Lavalin Inc.*
Aleksey Yelesin, *Amtrak*

At the time this recommended practice was updated, the Commuter, Intercity and High-speed Rail Mechanical Working Group included the following members:

David Warner, SEPTA, *Chair*
Rudy Vazquez, Amtrak, *Vice Chair*
Paul Jamieson, retired, *Secretary*

Mohamed Alimirah, *Metra*
Carl Atencio, *Denver Transit Operators*
Frank Banko, *WSP USA*
Michael Barnes, *Jacobs*
Taft Bearden, *Atkins Global NA*
David Bennett, *Capital Metro. Trans. Authority*
Jonathan Bernat, *New York Air Brake*
B.A. "Brad" Black, *Virginkar & Associates*
Stephen Bonina, *WSP USA*
Glenn Brandimarte, *ORX Rail*
Tony Brown, *MTA of Harris County*
Richard Bruss, *retired*
Michael Burshtin, *retired*
Greg Buzby, *SEPTA*

Dennis Cabigting, *STV Inc.*
Elvin Calderon, *Denver Transit Operators*
Paul Callaghan, *Transport Canada*
Gordon Campbell, *Crosslinx Transit Solutions*
Kevin Carmody, *STV Inc.*
David Carter, *New Jersey Transit*
Steve Cavanaugh, *Metrolinx (GO Transit)*
Steve Chrismer, *Amtrak*
Dion Church, *SNC-Lavalin Inc.*
John Condrasky, *retired*
Joshua Coran, *Talgo Inc.*
Michael Craft, *Paragon Robotics*
Brian Creely, *Siemens Mobility, Inc.*
Brendan Crowley, *New York Air Brake*
Ryan Crowley, *Atkins Global NA*
Richard Curtis, *Curtis Engineering Consulting*
Steven Dedmon, *Standard Steel LLC*
Joe Di Liello, *VIA Rail Canada Inc.*
David Diaz, *Hatch LTK*
Adam Eby, *Amtrak*
Phillippe Etchessahar, *ALSTOM Transport*
Gary Fairbanks, *Federal Railroad Administration*
Robert Festa, *MTA Long Island Rail Road*
Steve Finegan, *Atkins Global NA*
Gavin Fraser, *Jacobs*
Francesco Fumarola, *ALSTOM Transport*
Edward Gacsi, *New Jersey Transit*
Joe Gagliardino, *Arcosa*
Sebastien Geraud, *ALSTOM Transport*
Jeffrey Gordon, *Federal Railroad Administration*
Guillaume Ham-Livet, *ALSTOM Transport*
Eric Harden, *Knorr Brake Corp.*
Nick Harris, *Hatch LTK*
Jasen Haskins, *Atkins Global NA*
James Herzog, *Hatch LTK*
Kenneth Hesser, *Hatch LTK*
Lew Hoens, *MTA Metro-North Railroad*
Christopher Holliday, *STV Inc.*
Greg Holt, *Penn Machine Co.*
George Hud, *Hatch LTK*
John Janiszewski, *Hatch LTK*
MaryClara Jones, *Transportation Technology Center*
Robert Jones, *Stadler Rail Group*
Larry Kelterborn, *LDK Advisory, Inc.*
Joseph Kenas, *Bombardier Transportation*
Peter Klauser, *Vehicle Dynamics Group*
Heinz-Peter Kotz, *Siemens Mobility, Inc.*
Scott Kramer, *Arcosa*
Tammy Krause, *Atkins Global NA*
Pallavi Lal, *Hatch LTK*
Peter Lapre, *Federal Railroad Administration*
Nicolas Lessard, *Bombardier Transportation*
Cameron Lonsdale, *Standard Steel, LLC*
Daniel Luskin, *Amtrak*
Chris Madden, *Amtrak*
Francesco Maldari, *MTA Long Island Rail Road*
Brian Marquis, *Volpe Natl. Trans. Systs. Center*
Eloy Martinez, *Hatch LTK*
Francis Mascarenhas, *Metra*
Robert May, *Hatch LTK*
Ronald Mayville, *Simpson Gumpertz & Heger, Inc.*
Patrick McCunney, *Atkins Global NA*
Gerard McIntyre, *Knorr Brake Corp.*
Bryan McLaughlin, *New York Air Brake*
William Minnick, *Omni Strategy*
Luke Morscheck, *Hatch LTK*
Karl Mullinix, *Knorr Brake Corp.*
Joshua Munoz, *Hatch LTK*
Paul O'Brien, *Transit District of Utah*
Chase Patterson, *Voith Turbo, Inc.*
Joe Patterson, *Amsted Rail*
John Pearson, *Hatch LTK*
Martin Petzoldt, *Railroad Friction Products, LLC*
James Pilch, *Standard Steel, LLC*
Ian Pirie, *STV Inc.*
Brian Pitcavage, *Hatch LTK*
Brandon Reilly-Evans, *Hatch LTK*
Peter Reumueller, *Siemens Mobility, Inc.*
Danial Rice, *Wabtec Corp.*
Steven Roman, *Hatch LTK*
Carol Rose, *STV Inc.*
Thomas Rusin, *Rusin Consulting*
Thomas Rutkowski, *Virgin Trains*
Mehrdad Samani, *Hatch LTK*
Gerhard Schmidt, *Siemens Mobility, Inc.*
Martin Schroeder, *Jacobs*
Richard Seaton, *TDG Transit Design Group*
Frederic Setan, *ALSTOM Transport*
Patrick Sheeran, *Hatch LTK*
Melissa Shurland, *Federal Railroad Administration*
David Skillman, *Amtrak*
Benjamin Spears, *Hatch LTK*
Rick Spencer, *Knorr Brake Corp.*
Rex Springston, *AECOM*
Mark Stewart, *Hatch LTK*
Jonathan Sunde, *Strato Inc.*
Lukasz Szymsiak, *VIA Rail Canada, Inc.*
Mehdi Taheri, *Raul V. Bravo & Associates, Inc*
Ali Tajaddini, *Federal Railroad Administration*
Jeff Thompson, *SEPTA*
Matthew Todt, *Amsted Rail*
Ron Truitt, *HTSI*
Anthony Ursone, *UTC/Rail & Airsources, Inc.*
Frank Ursone, *UTC/Rail & Airsources, Inc.*

Michael Von Lange, *UTC/Rail & Airsources, Inc.*
Gary Wagner, *Amsted Rail*
Michael Wetherell, *McKissack & McKissack*
Brian Whitten, *SNC-Lavalin Inc.*
Kristian Williams, *Amtrak*
Todd Williams, *Penn Machine Co.*
Nicholas Wilson, *Transportation Technology Center*
Tim Wineke, *Knorr Brake Corp.*
Reggie Wingate, *Knorr Brake Corp.*
Aleksey Yelesin, *Amtrak*
Kevin Young, *Axis, LLC*
Gregory Yovich, *NICTD*
Steven Zuiderveen, *Federal Railroad Administration*

Staff advisers
Narayana Sundaram, *American Public Transportation Association*
Nathan Leventon, *American Public Transportation Association*

Introduction

This introduction is not part of APTA PR-M-RP-009-98, Rev. 2, "Truck Design," formerly titled "New Truck Design."

This recommended practice applies to all:

- railroads that operate intercity or commuter passenger train service on the general railroad system of transportation; and
- railroads that provide commuter or other short-haul rail passenger train service in a metropolitan or suburban area, including public authorities operating passenger train service.

This recommended practice does not apply to:

- rapid transit operations in an urban area that are not connected to the general railroad system of transportation;
- tourist, scenic, historic or excursion operations, off the general railroad system of transportation;
- operation of private cars, including business/office cars and circus trains unless otherwise required by other standards or regulations;
- railroads that operate only on track inside an installation that is not part of the general railroad system of transportation

Truck Design

1. General truck functions

The function of a railcar truck is to support the vehicle(s), to guide the vehicle along its intended route and to transmit the tractive/braking forces between the rail and carbody. The truck distributes the weight of the vehicle to the track through the truck structure, suspension and wheels and acts to provide the necessary steering forces to negotiate curves. Also, the truck acts to attenuate the effects of track geometric irregularities on wheel load and on acceleration and vibration within the car.

The suspension design should provide sufficient control of the roll movements of the car to keep the carbody within the dynamic clearance envelope and mitigate wheel unloading under all operating conditions.

2. Safety-related factors

As a minimum, the safety-related factors listed in this section should be satisfied in a new truck design to ensure safe operation. For existing truck designs that are to be applied to a car design or service condition that differs significantly from the truck's previous application(s), the portions of this section that could be affected by the car design or service condition differences should be reviewed and satisfied.

2.1 Loads and stresses

The trucks and their components should be capable of bearing the static and dynamic loads imposed in service, and the trucks should operate safely and without structural failures under all loading conditions from empty to maximum load condition. The trucks and their components should be capable of enduring the shock loads imposed by normal service.

2.2 Service conditions

The trucks and associated components should operate safely under the railroad specified conditions of operating speed, superelevation, cant deficiency, vehicle performance, environmental conditions, minimum curvature, special trackwork and track geometry conditions over the required service life of each of the respective components.

2.3 Clearances and mechanical stops

The truck design should have sufficient stiffness and clearances to ensure proper operation and to enable compliance with CFR requirements such as 49 CFR § 229.63, §229.71, §229.123 and §238.303(e)(9), and the car-level clearance diagram as defined by the purchasing railroad Technical Specification.

Mechanical stops should be provided to restrict movements beyond the limits that would damage the equipment. Truck rotational stops, if required or provided, should comply with the requirements of APTA PR-CS-S-034-99, "Design and Construction of Passenger Railroad Rolling Stock."

2.4 Locking means

Suitable locking means should be provided to secure the completely assembled truck, including truck frame, wheelsets, and bolster if used to the carbody. Please refer to the APTA PR-CS-S-034-99, “Design and Construction of Passenger Railroad Rolling Stock,” and the applicable requirements of 49 CFR §229.67(a) and (b), §229.141(a)(5), §238.219, §238.419 and §238.717.

2.5 Fail-safe design of critical components

The vehicle suspension system and the carbody tilting system, if used, should be designed with suitable backup suspension or safety stop means to reduce the likelihood of a catastrophic vehicle failure, derailment, collision or extension beyond the track clearance envelope in the event of a failure in the normal suspension and/or tilting element(s) or system(s).

Backup safety means should be provided for the traction motors, drive shafts, traction rods and gear units to prevent these components from causing a derailment as a result of a failure of the primary mounting or attachment devices.

Backup safety means also should be assessed for any other components that could cause derailment as a result of failure of the primary mounting or attachment devices.

The truck should also include safety hangers as required by 49 CFR §229.65(a), §229.99 and §238.303(e)(5)(i).

2.6 Shop safety

The truck and its components should be designed with consideration for the personnel safety during shop maintenance activities. Sharp edges and pinch points should be avoided where practical. Lifting and jacking locations of adequate structural strength should be provided for normal shop assembly and disassembly activities.

3. Design analysis and considerations

Design analysis should be conducted to demonstrate that new commuter and intercity railcar trucks will operate safely for the required service life under the environmental, operating and physical conditions as specified by the purchasing railroad. The design analysis should include elements such as structural analysis, assessment of P_2 dynamic track forces and evaluation of truck-mounted equipment. In the case of fabricated trucks, the standard used for stress evaluation of welds must be the same standard used for welder qualification, weld procedure qualification, fabrication and inspection (AWS D1.1 or other industry standard, based on demonstrated equivalence and approval by the purchasing railroad).

3.1 Structural analysis

All truck structural components should have the necessary strength with adequate safety factors to withstand the maximum stresses imposed in service, including static, exceptional and fatigue loads due to vehicle loading, operating conditions, forces from suspension components, vibration (inertial forces), track shocks, motor and braking loads, and tilting forces, and anticipated combinations of these forces.

Stress analysis should include the calculated stresses, allowable stresses and safety margins for all truck structural components under all anticipated loading conditions and load combinations. The stress analysis should consist of finite element analysis (FEA), supplemented as necessary by conventional calculations.

Static load stress analysis represents the maximum normal service loads and is generally compared with some fraction of material yield strength. Exceptional load stress analysis represents extreme loads and generally must be applied to the structure without permanent deformation. Fatigue load stress analysis represents expected service loads, which are established and evaluated using either the infinite life method or cumulative damage method as described below.

3.1.1 Infinite life method

- **Load cases:** A limited number of load cases are defined, which combine the upper and lower bounds of each individual service load. Load magnitudes should aim to be conservatively high and should be based on loads used for service-proven trucks in the same or similar operating environment. Maximum and minimum values of each individual service load are assumed to occur simultaneously, and load directions are phased to produce the worst-case loading.
- **Stress evaluation:** Stress results for all load cases are compared to determine the maximum stress range at each location.
- **Allowable stress:** The maximum stress range at each location must not exceed the material endurance limit. Stress range for welded steel (fabricated) truck base material and welds should be less than the fatigue stress threshold by respective stress category according to AWS D1.1, or equivalent standard as approved by the purchasing railroad. Endurance limits for cast materials should be established by the manufacturer, supported by fatigue test data and approved by the purchasing railroad.

3.1.2 Cumulative damage method

- **Load cases:** Load cases are intended to represent the actual operating environment (plus acceptable margin) and may be defined by estimated load conditions on the intended service routes, actual measured on-track load values or load values predicted by multibody simulation. Maximum and minimum values of each individual service load and load directions are phased to represent the actual operating environment. The intended service life of the truck must be defined by the purchasing railroad and the sequence/number of applications of each load case scaled to represent the expected operation over the service life of the vehicle (mileage).
- **Stress evaluation:** Stress results for each load case are calculated, and the stress range and number of cycles (stress spectrum) for the complete expected operation over the service life is tabulated at each location.
- **Allowable stress:** The stress range and number of cycles (stress spectrum) at each location is scaled to the intended service life of the truck. The cumulative damage at each location is summed using an industry-accepted cumulative damage model. The cumulative damage at each location must not exceed unity (1.0) less safety margin as approved by the purchasing railroad. Stress-Life (SN) curves for welded steel (fabricated) truck base material and welds are defined by respective stress category according to AWS D1.1, or equivalent standard as approved by the purchasing railroad. SN-Curves for cast materials should be established by the manufacturer, supported by fatigue test data and approved by the purchasing railroad.

3.1.3 Structural design loads

Structural design loads should be established on an individual project basis, based on the purchasing railroad's technical specification and the supplier's structural validation plan.

Static, exceptional and fatigue loads should include vertical static plus dynamic augment, twist, lateral, longitudinal, roll, braking and traction (if applicable) plus any other loads that are considered significant, subject to agreement between the builder and purchasing railroad.

Appendix A (informative) provides a detailed tabulation of historic structural design criteria and industry standards, which can aid in establishing structural design criteria for a given vehicle type.

3.1.4 Truck-mounted equipment design loads

Static, exceptional and fatigue design loads for truck-mounted equipment shall be selected to prevent sudden rupture and progressive fatigue failure for the duration of the truck's intended service life.

Design load levels shall be based on typical best practice in the North American railroad industry and may be based partly on the recommended loads in well-accepted international reference documents such as GM/RT 2100 and EN 13749. If required, these design loads should be adjusted to account for the specific conditions (for example, track quality) of the intended operating environment.

Design loads may also be partly or wholly based on the results of prior field testing of similar truck designs operating in the intended environment, with adequate safety factors and adjustments applied to account for any differences in equipment mounting.

Design loads must consider the mass of the mounted equipment and whether the equipment is rigidly or resiliently mounted to the truck frame. For example, rigidly mounted heavy equipment such as traction motors may influence the overall truck frame dynamic behavior, which will significantly affect the design loads.

Truck-mounted equipment design loads may be partly or wholly based on the results of multibody simulation (MBS) performed for the new truck design operating in the intended environment. Such analysis must consider higher frequency (above 30 Hz) responses than standard MBS models used for vehicle dynamics. Flexible bodies (truck frame, axle, bolster, etc.) must be properly represented in the MBS model, with special consideration given to track dynamic response. In such cases, design loads should incorporate safety factors to adequately cover the potential inaccuracies of MBS analysis.

Equipment mounting brackets and attachments must be designed to avoid resonance in the range of frequencies that may be excited at the truck frame and axle levels and result in rapid mechanical damage.

For non-service proven truck designs or applications of existing designs in new environments, it is recommended that equipment design loads be subsequently validated by field testing of the final truck design in its intended operating environment. Since such testing captures only a short-duration snapshot of the operating conditions, field testing results should demonstrate that the selected design loads incorporate enough safety margin to cover variations in operating conditions over time.

3.1.5 Structural validation plan

A structural validation plan should be established and followed for all new truck designs, existing truck designs with significant modifications and existing truck designs applied under a new car. The validation plan should describe in detail the list of the steps planned to demonstrate structural safety and compliance with the requirements defined in the technical specification. The validation plan should be prepared by the supplier and approved by the purchasing railroad. The validation plan should act as a guide for the design process, covering the pertinent details for each topic listed below:

- Stress analysis
    - Design load cases: exceptional, static and fatigue
    - FEA modeling approach: element types, loads, constraints
    - Fatigue analysis approach: infinite life or cumulative damage
    - Material properties and limiting stress values

    - Manual calculations as applicable
    - Guiding standards and regulations as applicable
- Testing – laboratory
    - Static and exceptional load cases
    - Fatigue load cases: load cycles, load phasing and counting method
    - Pass/fail criteria
    - Test laboratory and location
    - Test setup: fixturing, load application and validation concept
    - Instrumentation plan
- Validation – FEA Model
    - Comparison of predicted (FEA) and recorded (laboratory test) stresses
    - Definition of critical stress locations
    - Stress correlation limits (allowable stress deviation between FEA and test)
- Testing – on-track
    - Testing conditions: braking, tractive effort, speed, car loading
    - Test track location: number of test runs, mileage
    - Instrumentation plan
- Validation – fatigue assessment
    - Comparison of design stresses (FEA) and operating stresses (on-track test)
    - Comparison of laboratory test cumulative damage and on-track life assessment
- Validation – manufacturing of existing truck designs
    - Demonstrate dimensional similarity between proposed design and service-proven design
    - Comparison of truck materials, specifications and drawings
    - Comparison of manufacturing process: welding and casting process control
    - Comparison of non-destructive testing (NDT)

The extent of laboratory and on-track testing will depend on how design loads are established (degree of uncertainty or conservatism), the stress analysis approach, degree of modification for existing truck designs, maintenance and inspection intervals, and the level of safety margin in the stress analysis results.

For existing truck designs with significant modifications and existing truck designs applied under a new car, the structural validation plan should demonstrate why previously accepted analysis and tests are still applicable and then focus on validation of changes in design or application. Alternatively, evidence supporting the use of existing truck designs may be provided in the form of accumulated satisfactory service experience.

General guidelines for analysis and testing are given in **Table 1**, with suggested attributes marked "X" or described further by the supporting notes. Final scope must be uniquely established for each project and subject to the considerations previously listed for each truck design and application.

TABLE 1
General Guidelines for Analysis and Testing

Truck Type	New Truck Design Note 4		Existing Truck Design with Modification		Existing Truck Design New Application	
Stress Analysis Approach	**Infinite Life**	**Cumulative Damage**	**Infinite Life**	**Cumulative Damage**	**Infinite Life**	**Cumulative Damage**
Stress analysis	X	X	X	X	Note 2	Note 2
Testing – laboratory	X	X	Note 1	Note 1	Note 2	Note 2
Validation – FEA model	X	X			Note 3	Note 3
Testing – on-track		X	Note 1	Note 1		X
Validation – fatigue assessment		X		Note 1		X
Validation – manufacturing	Note 5	Note 5	X	X	X	X

Note 1: Requirements for laboratory testing, on-track testing and validation to be determined based on the extent of truck modification and uncertainty or conservatism in design loads. Limited localized laboratory testing may be sufficient for minor modifications.

Note 2: Previously accepted stress analysis and laboratory testing for an existing truck design may be submitted, as supported by comparison of loads and operating environment for previous and new cars.

Note 3: Service history of existing truck may be submitted, as supported by comparison of loads and operating environment for previous and new cars.

Note 4: Requirements for on-track testing and fatigue assessment validation should be established based on the degree of uncertainty and conservatism in the design loads, not solely on the stress analysis and testing approach. Table assumes high confidence and conservatism in infinite life loads, moderate confidence and conservatism in cumulative damage loads.

Note 5: Manufacturing process for new truck designs is reviewed in detail with the purchasing railroad during first article inspection.

3.2 Vehicle dynamic analysis

Vehicle dynamic analyses are recommended, and in some cases required by U.S. federal regulations, for new or unproven truck designs to evaluate track-worthiness, derailment potential, acceptable carbody and truck accelerations (hunting). Please refer to proposed 49 CFR §238.145, current 49 CFR §213.345 and §238.427 as applicable, and the forthcoming APTA vehicle dynamics performance standard.

3.3 P_2 dynamic track forces

Limits for P_2 dynamic track forces should be established on an individual project basis, based on the purchasing railroad's technical specification. Appendix B provides an overview of calculation methods for a number of purchasing railroads, along with a tabulation of historic P_2 dynamic track force limits.

3.4 Clearance analysis

The inputs of the clearance analysis should include combinations of normal operating conditions and certain abnormal conditions as defined by the purchasing railroad and technical specification. These requirements should govern all clearance analyses of the truck and carbody relative to the railroad clearance diagram (static and dynamic outline) and truck-to-carbody clearance analyses.

The following (informative) list includes the conditions which are typically considered in the three types of clearance analyses:

- Passenger loading as defined by the purchasing railroad.
- Truck swivel (yaw), roll and pitch motions for normal operation (track geometry conditions experienced by the vehicle).
- Primary and secondary suspension deflection for normal operation.
- Worst-case horizontal or vertical curve defined by the purchasing railroad Technical Specification.
- Worst-case combination of horizontal and vertical curves defined by the purchasing railroad Technical Specification.
- Wheel tread and flange wear.
- Suspension and friction element (side bearing) wear.
- Track gauge wear.

The following suspension failure modes and worst-case track conditions should be considered, in reasonable combination with the normal operating conditions listed above:

- Failed primary or secondary suspension elements.
- Worst case track geometry conditions experienced by the vehicle.

3.4.1 Car-level clearance

Perform a dynamic clearance envelope study of the completed car to ascertain that wayside clearance limits are not exceeded by any portion of the car or truck under any combination of load and operating condition.

3.4.2 Truck internal and truck to carbody-mounted equipment clearance

Perform a truck internal and truck to carbody-mounted equipment clearance study to ensure there is no unintended internal contact between truck components, traction links, suspension elements, lifting wires, cable and hoses and that there is ample clearance between the truck and carbody-mounted equipment. Additionally, wire, cable and hose lengths should be verified by this evaluation.

3.5 Helical coil spring design guidance

The section is intended to give general guidance on the design of coil springs. This guidance is intended to help avoid spring failures in service, avoid control suspension system variations resulting from manufacturing tolerances, and control the effects of real spring behavior (spring float, buckling, etc.) on suspension system performance. References listed are informative but should be followed when required by the purchasing railroad technical specification.

3.5.1 Design stress calculations

Design calculation references include EN 13906-1 and SAE HS-795.

Spring design calculations must consider both maximum stress under spring solid conditions and spring fatigue life under repeated load cycles. If the spring is subject to transverse deflections in service, both calculations (maximum stress and fatigue life) must consider additional stresses due to transverse deflection in combination with axial deflection. Fatigue calculations should consider load cycles based on a minimum vehicle weight of AW1 plus a minimum dynamic bounce factor of ±20 percent. Calculation inputs are subject to approval of the purchasing railroad. Supplemental informative fatigue calculations may also be performed considering vehicle weight of AW3 and solving for allowable dynamic bounce factor for infinite life.

3.5.2 Materials, workmanship and inspection

References related to spring materials, workmanship and inspection include ASTM A125-96 (2018), DIN 2096 (1981), EN 13298:2003, AAR M-114 and UIC 822 (2003).

3.5.3 Spring geometry

Typical spring geometry concerns include the following. Actual spring geometry requirements should be based on the specific application and related suspension performance issues.

- **Bar diameter:** The bar diameter is critical to determining spring rate and spring maximum stress. The associated tolerance must, at minimum, ensure spring rate variations meet suspension performance requirements.
- **Coil direction:** Adjacent nested coils should be wound in opposite coil directions.
- **Coil inner and outer diameters:** Tolerances must ensure that spring rate variations meet suspension performance requirements and avoid interference with adjacent coils. For long, slender springs (slenderness ratio, defined as free height divided by mean diameter, of approximately 4 or greater), potential interference caused by spring buckling should be considered.
- **Coil height:** Variations in spring free height and height at various load conditions must be limited to ensure that design spring loads versus actual spring loads meet suspension performance requirements.
- **Spring pitch uniformity:** Variations in coil-to-coil spacing at various load conditions must be limited to ensure spring linearity.
- **Spring end angular and transverse deviations:** Must be defined to ensure the spring meets the design assumption of a right cylinder with concentric top and bottom surfaces. The transverse deviation is the maximum lateral offset of the spring top surface to the bottom surface as measured at any radial position. The end angular deviation is the maximum vertical offset between the highest point versus the lowest point on the spring top surface with the spring bottom end placed on a horizontal surface.
- **Spring float direction:** Float is the tendency for coil springs, when loaded vertically with one end fixed and one end free, to deflect in the transverse direction at the free end. If this effect has a significant impact on the suspension system design and vehicle dynamic performance for a given truck design, the float direction should be measured/marked and the assembly orientation controlled.

3.5.4 End coil geometry

Spring fatigue failures can often be attributed to unfavorable geometry between the inactive coil at either spring end and the first active coil. Coil springs for railroad applications are typically hot wound with closed and ground spring ends. The bar stock from which the spring is wound is either tapered (tapered wound spring) or simply sheared (blunt wound spring).

Recommended practice is as follows:

- Each spring end shall be ground to provide a stable supporting surface for the spring. The bearing surface shall extend along a minimum arc of at least 240 deg. and ideally 270 deg. At least 180 deg. of the bearing surface shall have a width meeting or exceeding two-thirds the tapered width of the bar (with a tapered spring end) or diameter of the bar (with a blunt spring end). Roughness of the ground surface Ra must not exceed 125 µin (3.2 µm).
- The tip of the spring end should not extend radially beyond the spring minimum or maximum diameter (i.e., spring diameter plus allowable tolerance).

- Maximum thickness at the tip of the spring end should not be more than one-quarter of the bar diameter.
- Contact shall not occur between the tip of the spring end and the first coil. Under suspension tare load, the distance from the tip of the spring end to the first contact with the first coil must be at least one bar diameter (as measured along the circumference of the coil).
- Under suspension tare load, contact between the end coil and the first coil should occur over a distance of 0.75 in. (20 mm) or one-third of the mean coil diameter, whichever is greater. The object is to ensure contact over an extended distance to avoid excessive contact stress between coils. While desirable, the contact line need not be continuous.
- No sharp edges or burrs are allowed at the tip of the spring end. The minimum radius at all edges shall be 0.1 in. (2.5 mm). No nicks, gouges or any other surface damage are permitted at the spring end.

3.6 Rubber suspension components

EN 13913 is considered an informative reference for design and testing of rubber suspension components and can be used in conjunction with the requirements listed in the purchasing railroad technical specification and applicable ASTM standards.

3.7 Pneumatic, electrical and electronic equipment design analysis

All equipment attached to the truck should be designed to withstand vibration and shocks during regular service. The components should meet requirements of IEC 61373, category 3, for the unsprung portion of the truck (axle, wheelset and axle bearings) and category 2 for frame and bolster mounted equipment.

If the shock and vibration levels have not been previously characterized on a similar vehicle type operating in the intended environment, the truck supplier should conduct testing to determine the applicability of IEC 61373 criteria and adjust as required.

3.7.1 Ingress protection rating.

All electrical equipment and electrical enclosures should be protected from the effects of the environment and have NEMA 4 or at least IP64 ingress protection.

3.7.2 Electrical wiring and pneumatic piping

APTA PR-E-RP-002-98, latest revision, “Installation of Wire & Cable on Passenger Rolling Stock,” contains electrical wiring requirements that should be followed.

The arrangement for conduit, cable, wire routing and connections to equipment enclosures, and equipment contained in enclosures, shall be configured so structural, electrical and environmental integrity is maintained, and so the removal and replacement of the equipment enclosure are facilitated. Each arrangement employed shall be subject to vehicle buyer review and approval during design review.

Pneumatic piping installed on the truck shall meet the requirements of APTA PR-M-S-029-20, "Pneumatic Piping for Vehicles."

3.7.3 Grounding

Grounding of the truck should be designed as per APTA PR-E-S-005-98, latest revision, “Grounding and Bonding,” using at least one grounding brush assembly per truck. The carbody should be grounded through the truck as per APTA PR-E-S-005-98. Wheel-to-wheel resistance should also be as per APTA PR-E-S-005-98.

3.8 Axles, wheels and roller bearings

Axles should be specified in accordance with the guidelines contained in APTA PR-M-RP-008-98, latest revision, "Passenger Rolling Stock Axle Design."

Wheels should be selected based on the guidelines provided in APTA PR-M-RP-013-06, latest revision, "Selection of Wheels for Passenger Applications." Wheels should be manufactured in compliance with APTA PR-M-S-012-99, latest revision, "Manufacture of Wrought Steel Wheels for Passenger Cars and Locomotives."

Roller bearings should be designed for "no field lubrication" and should be fully enclosed, grease-lubricated roller bearings. All journal bearings used in a truck shall be fully interchangeable such that the same bearing is used at all positions in a truck.

Bearings should not require inspection more than the agreed-upon service interval determined by the supplier and purchasing railroad.

Bearings should have an ANSI/AFBMA L10 life that is consistent with the expected service life of the bearings, based on an application factor (AF), shock factor (C1) and passenger loading applied to the static load, as agreed upon between the supplier and purchasing railroad.

When required for wheel truing, bearing end caps should have plugs to allow the axle centers to be engaged by wheel truing machine centers, without removing the bearing end caps.

3.9 Wind loading analysis

For vehicles that will operate in an environment where side-wind loading is a specific concern (sustained wind greater than 50 mph), or as required by the purchasing railroad's Technical Specification, the supplier should perform a side-wind loading safety analysis. This analysis should address potential vehicle rollover from side-wind loading including operation under worst-case cant deficiency and cant excess conditions.

This analysis should either confirm safe operation of the vehicle under the current purchasing railroad operating rules or define the vehicle's maximum safe combination of side-wind loading speed and vehicle speed for incorporation into the operating rules.

Standards EN 14067-6 and GM/RT-2142 are considered an informative reference for wind loading calculations.

4. Structural qualification tests

Qualification tests of all new or unproven truck designs should be performed on a representative specimen to demonstrate conformance to specification requirements for structural components.

4.1 Structural integrity

For trucks similar in design or application to previous experience, the purchasing railroad may agree to waive any or all portions of these structural qualification tests. Adequacy must be proven by engineering analysis of previously completed equivalent testing and/or by successful service of the truck.

4.1.1 Truck static test

The purpose of this test is to verify that the static strength of the truck frame, bolster and other primary structural components is not exceeded. The truck may be tested either as individual load bearing components or as an assembly. Provision should be made to apply all input loads described below and to restrain these input loads in a manner that is functionally equivalent to the reactive forces that occur in service. Loads should enter the truck components at the normal application points and should be combined in each case to produce the most severe load combinations that are anticipated in normal service.

Input loads and allowable stress criteria should be as defined by the structural validation plan.

Strain measurement techniques should be used on the truck structure at maximum stress points as agreed to by the truck supplier and the purchasing railroad. The locations of maximum stress points should be determined by structural analysis.

4.1.2 Truck fatigue test

To demonstrate that the truck has adequate fatigue strength under dynamic loading, the truck frame, bolster and other primary structural components should be subjected to a base fatigue test of combined normal loading cycles as defined by the structural validation plan.

The truck may contain its internal elastomeric cushioning and springs or substitute blocking.

Before, during and at the conclusion of the fatigue test, the truck should be inspected by industry-recognized non-destructive test methods to detect evidence of crack initiation or progression. Precise definition of a “failure” should be agreed upon in advance of truck testing.

Specimens subject to test shall be of series production of the same type and manufacture according to an equivalent set of specifications, including drawings, procedures and quality plan, and shall not have differences in any critical factors that could influence the outcome of the tests.

The fatigue testing method should be either the infinite life or cumulative damage test method outlined below and should match the stress analysis method.

Strain measurement techniques should be used on the truck structure at maximum stress locations, maximum fatigue utilization locations and other points of interest as agreed to by the truck supplier and the purchasing railroad. The strain measurement locations should be determined by structural analysis.

An extended fatigue test, which subjects the truck to additional fatigue cycles under increased loads, shall be performed when required by the purchasing railroad technical specification. The extended fatigue test should be considered an “engineering information” test which quantifies the additional fatigue margin in the truck design. The number of cycles and loads applied, along with pass/fail criteria for crack initiation and growth, should be defined by the structural validation plan and suppliers test plan.

4.1.3 Fatigue test: infinite life method

The truck frame and bolster should be subjected to combined loading cycles to produce the highest stress range for the types of loads expected to occur in normal service. The number of cycles should be equal in number to the infinite fatigue life thresholds of the material and fabrication details used in truck construction.

The phasing of combined loads should be arranged to produce both maximum and minimum stress levels at critical locations for reasonably anticipated service load combinations.

For the types of load applications where the extreme excursions are expected to occur at a substantially lower rate, such as a full load reversal due to change in direction of travel or due to an emergency brake application, the truck fatigue test duty cycle should be adjusted to test at these extreme load excursion levels only for the number of these extreme load cycles that can realistically be expected to occur during the specified design life of the truck. The number of extreme load cycles to be tested is to be agreed upon by the truck supplier and purchasing railroad.

Typically, the infinite fatigue life threshold for cast steel structures is considered to be 2 million cycles.

Welded steel structures are not considered to have attained full endurance limit until the infinite fatigue life threshold for the most severe non-redundant fatigue strength weld detail is reached, as defined by AWS D1.1 or other industry accepted standard and approved by the purchasing railroad.

4.1.4 Fatigue test: cumulative damage method

The truck frame and bolster should be subjected to combined loading cycles to produce cumulative damage at all critical locations equal to that expected over the intended service life of the truck, plus acceptable margin.

The number of cycles and phasing of combined loads, including extreme load applications, should represent the actual operating environment over the intended service life of the truck. Amplified loads with reduced cycle count may be applied, based on demonstration of equivalent cumulative damage and as approved by the purchasing railroad.

4.2 Truck-mounted reservoirs

Reservoirs should be designed and tested in accordance to ASME Boiler and Pressure Vessel Code for Unfired Pressure Vessels. ASME certification and registration is not required unless specified or deemed safety-critical by the purchasing authority. Materials not listed in ASME Section II are acceptable with equivalent safety factors. The design and test pressure should meet or exceed the maximum possible pressure of the system, regardless of pressure regulators. Pressure testing should be performed on 100 percent of reservoirs per the ASME Boiler and Pressure Vessel Code.

Truck frames and bolsters that are used as air reservoirs shall have a drain plug at the low point in the air chamber, if practicable, to allow for draining of condensation and inspection of air chamber. The interiors of truck frame and bolsters used as air reservoirs shall be suitably cleaned to remove scale, weld flux, sand and other contaminates resulting from production, prior to use.

4.3 Structural model validation

It is anticipated that compliance with load cases required to achieve compliance with federal regulations, industry standards and the Technical Specification will not all be demonstrated through physical testing and that modeling results alone will be used to satisfy some of these requirements.

Therefore, it is critical that adequate documentation be provided to establish credibility in the modeling methodology and the ability of the model to produce realistic results.

A model validation report is to be provided that fulfills this purpose. The format, acceptance limits and content of the report shall be as agreed to by the truck supplier and purchasing railroad, but shall contain at a minimum:

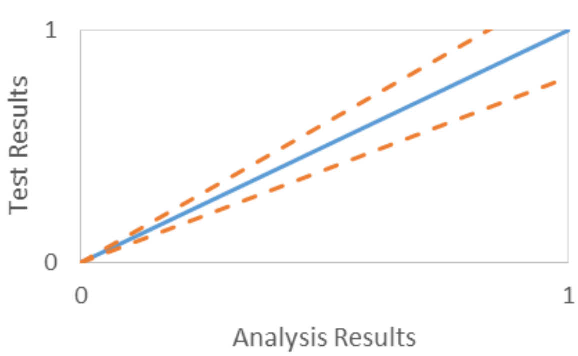
- Identification of which tests will be used for model validation. At a minimum, the static test results should be included in the model validation activity.
- Identification of which measurement device (strain gauges, displacement sensors, load cells, etc.) output will be used for model validation (if not all). It is recommended that strain gauge measurements that indicate stresses greater than or equal to 25 percent of the allowable be considered

in the validation activity. The manufacturer should provide an explanation of the rationale for the pre-selected list of strain gauges for each load case within the validation activity.

- Identification of the version of the model used to perform the validation activities.
- Appropriate tabulations or other graphical depictions of the comparisons of the model and test results for each of the relevant load cases and for each of the relevant measurement devices and documentation of the relative differences between the two results.
- Explanation of reason(s) for instances in which model results do not correlate with the measured value using the prescribed criteria.
- For the purposes of the validation report, the following maximum correlation criteria should be applied:
    - Model-predicted values of strains/stresses should be within ±20 percent of the measured values at the relevant locations.
    - Model-predicted values of deflections/displacements, if measured, should be within ±10 percent of the measured values at the relevant locations.
    - Model-predicted load reactions, if measured, should be within 5 percent of the measured values for pure vertical loads and within 10 percent for combinations of vertical, longitudinal and lateral loads at the relevant locations.
    - For cast truck and bolster designs, which are subject to larger dimensional variations inherent to the manufacturing process:
        - For gauge locations with recorded stresses lower than predicted by FEA by more than 20%, explanation of the difference is not required provided the gauge location and load applications/reactions have been confirmed.
        - For gauge locations with recorded stresses higher than predicted by FEA by more than 20%, an explanation of the difference is required to the satisfaction of the purchasing railroad. Explanation of the differences may include casting configuration details, stress levels below allowable, gauge placement or further analysis of the specific area(s) of the casting.

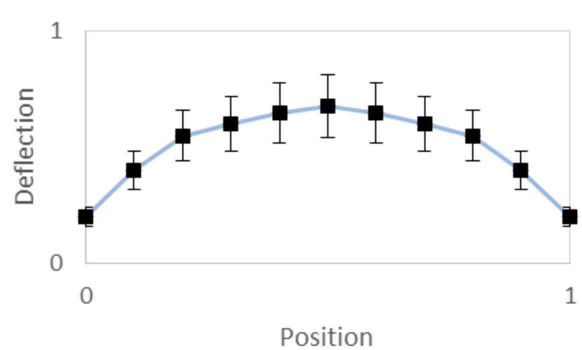
Correlation between measured and predicted values shall be presented in a form similar to that shown in **Figure 1** and **Figure 2**, in which the dashed curves or the error bars represent the correlation tolerance and the solid curves represent the test result. Data for the model-predicted values are added to these plots. Quantities represented by the horizontal and vertical axes are selected based on the relevant load case. Depending on the quantities compared, alternate representations are allowed based on agreement between the truck supplier and purchasing railroad.

FIGURE 1
Stress and Strain Depictions



Typical depiction of comparison between test and analysis results for stresses or strains

FIGURE 2
Deflection and Displacement Depictions



Typical depiction of comparison between test and analysis results for deflections/displacements

In the event that validation cannot be achieved within the correlation tolerances and model refinement or revision is required for any reason to improve correlation, results for all load cases must be reproduced using the revised model unless the truck supplier can provide a documented, compelling case describing why this is not necessary.

5. Lean and curving tests

5.1 Dynamic clearance

To verify the vehicle dynamic clearance envelope, a completed car with simulated maximum load should be elevated at one rail by the greater of 6 in. superelevation or the maximum superelevation permitted by the purchasing railroad.

Lateral displacements, vertical displacements and roll angle of the carbody should be measured and compared for compliance with the purchasing railroad's specification for the vehicle dynamic clearance outline.

5.1.1 Maximum cant deficiency

Lean testing for maximum cant deficiency is outside the scope of this document. Please refer to 49 CFR §213.57 and §213.329.

5.1.2 Maximum cant excess

To verify wheel unloading at maximum cant excess (to address potential vehicle rollover and passenger safety issues from side-wind loading when the vehicle is stopped or traveling at very low speeds on highly superelevated curves), a completed car in a ready-for-service load condition should be elevated at one rail by the greater of 6 in. superelevation or the maximum superelevation permitted by the purchasing railroad.

Under these test conditions, no wheel of the vehicle should unload to a value less than 50 percent of its static value on perfectly level track.

5.2 Curve clearance test

Truck clearances should be tested by moving the vehicle over a curve or crossover, or by using a transfer table to duplicate the worst-case combinations of horizontal and vertical curves and load conditions. Worst-case conditions should match those outlined in Section 3.4, as reasonably achievable, or be accounted for in the test evaluation. Clearances should be checked internal to the truck and between the truck and other items such as truck-mounted equipment, carbody and carbody-mounted equipment. Additionally, cables and hoses should be checked for overextension, interference, slack and minimum bend radius.

Related APTA standards

APTA PR-CS-S-034-99, "Design and Construction of Passenger Railroad Rolling Stock"
APTA PR-E-RP-002-98, "Installation of Wire and Cable on Passenger Rolling Stock"
APTA PR-E-S-005-98, "Grounding and Bonding,"
APTA-PR-M-RP-008-98, "Passenger Rolling Stock Axle Design"
APTA-PR-M-RP-013-06, "Selection of Wheels for Passenger Applications"
APTA-PR-M-S-012-99, "Manufacture of Wrought Steel Wheels for Passenger Cars and Locomotives"
APTA-PR-M-S-014-06, "Wheel Load Equalization of Passenger Rolling Stock"
APTA-PR-M-S-015-06, "Wheel Flange Angle for Passenger Equipment"
APTA PR-M-S-029-20, "Pneumatic Piping for Vehicles"
APTA PR-M-S-031-20, "Low-Speed Curving Performance of Railroad Passenger Equipment"

Forthcoming: Proposed APTA Vehicle Dynamic Performance Standard

References

This recommended practice shall be used in conjunction with the following publications. When the following publications are superseded by an approved revision, the revision shall apply.

Association of American Railroads, Specification M-114, Specification for Helical Springs, Heat Treated Steel

American Boiler Manufacturers Association, ANSI/ABMA 11-2014, Load Ratings and Fatigue Life for Roller Bearings

American Society of Mechanical Engineers, ASME Boiler and Pressure Vessel Code – Unfired Pressure Vessels

American Welding Society, Inc., AWS D1.1/D1.1M:2015 Structural Welding Code - Steel

ASTM International, ASTM A125-96 (2018), Standard Specification For Steel Springs, Helical, Heat-Treated

Code of Federal Regulations:
- Title 49 CFR, Part 213, Track Safety Standards
  - Subpart C, Track Geometry
    - Section 57, Curves; elevation and speed limitations.
  - Subpart G, Train Operations at Track Classes of 6 and Higher
    - Section 329, Curves; elevation and speed limitations.
    - Section 345, Vehicle/track system qualification.
- Title 49 CFR, Part 229, Railroad Locomotive Safety Standards
  - Subpart C, Safety Spring rigging. Paragraph (a)
    - Section 67, Trucks. Paragraphs (a) and (b).
    - Section 99, Safety hangers.
    - Section 141, Body structure, MU locomotives, Paragraph (a)(5)
- Title 49 CFR, Part 238, Passenger Equipment Safety Standards
  - Subpart B, Safety Planning and General Requirements
    - Section 145 (Proposed)
  - Subpart C, Specific Requirements for Tier I Passenger Equipment
    - Section 219, Truck-to-car-body attachment.
  - Subpart D, Inspection, Testing, and Maintenance Requirements for Tier I Passenger Equipment

Section 303, Exterior calendar day mechanical inspection of passenger equipment, Paragraph (e)(5)(i)
Subpart E, Specific Requirements for Tier II Passenger Equipment
Section 419, Truck-to-car-body and truck component attachment.
Section 427, Suspension system.
Subpart H, Specific Requirements for Tier III Passenger Equipment
Section 717, Truck-to-car-body attachment.

Deutsches Institut fur Normung E.V., DIN 2096 (1981), Helical compression springs made of round wire and rod; Quality requirements for hot formed compression springs

European Standards
EN 13298:2003 – Railway applications. Suspension components Helical suspension springs, steel.

EN 13749:2011 – Railway applications. Wheelsets and bogies. Method of specifying the structural requirements of bogie frames.

EN 13906-1:2013 – Cylindrical helical springs made from round wire and bar — Calculation and design Part 1 : Compression springs.

EN 13913:2003 – Railway applications. Rubber suspension components – Elastomer-based mechanical parts.

EN 14067-6:2018 - Railway applications. Aerodynamics – Part 6: Requirements and test procedures for cross wind assessment

CENELEC EN 60529:1991 – Degrees of Protection Provided by Enclosures (IP Code)

International Electrotechnical Commission, IEC 61373: Railway applications – Rolling stock equipment – Shock and vibration tests

International Union of Railways, UIC 822 (2003) – Technical Specification For The Supply Of Helical Compression Springs, Hot Or Cold Coiled For Tractive And Trailing Stock

National Electrical Manufacturers Association, NEMA Enclosure Type 4: Watertight

RSSB, Railway Group Standard GM/RT2142, Issue 4.1 (2019) - Resistance of Railway Vehicles to Roll-Over in Gales

SAE International, SAE HS-795:1997 – Manual on Design and Application of Helical and Spiral Springs

Definitions

allowable stress: The maximum stress permitted under working loads by codes and specifications.

cant deficiency: The amount of superelevation that would need to be added to the curved track to achieve vehicle balance speed.

cant excess: The opposite of cant deficiency, i.e. the amount of superelevation that would need to be removed from the track to achieve vehicle balance speed on a curve.

dynamic clearance envelope: Space (area perpendicular to the trajectory of moving rail vehicle along the track) that can be occupied by the rail vehicle due to any combination of loading, lateral motion, vertical motion or suspension failure.

dynamic load: A non-static, time-depended load caused by acceleration or deceleration of a mass.

endurance limit: Represents a stress level below which a load may be repeatedly applied an indefinitely large number of times without causing failure. Unless qualified, the endurance limit is usually understood to be that for completely reversed bending.

> **NOTE:** The above definition originated from "The Testing and Inspection of Engineering Materials," H.E. Davis, G. E. Troxell and C.T. Clement, McGraw-Hill Book Co., p. 239.

exceptional load: An extreme load representing the maximum load at which full serviceability is to be maintained and used for assessment against static material properties. Exceptional loads should not produce permanent deformation or excessive deflections. (Source: EN 13749)

fatigue load: A cyclic load on a structure due to expected operating conditions.

fatigue strength: The stress range a material can withstand for a given number of cycles without fracture.

finite element analysis (FEA): Simulation of a physical phenomenon using a numerical mathematic technique referred to as the finite element method, or FEM. Used to predict how a product reacts to real-world physical effects.

frequency response: A quantitative measure of the output spectrum of a system or device in response to a stimulus, used to characterize the dynamics of the system. Measure of magnitude and phase of the output as a function of frequency, in comparison with the input.

inertial force: A force opposite in direction to an accelerating force acting on a body and equal to the product of the accelerating force and the mass of the body.

infinite fatigue life threshold: Represents the cycle count threshold at which the endurance limit is achieved.

Ingress protection (IP) rating: Defined levels of sealing effectiveness of electrical enclosures against intrusion from foreign bodies (tools, dirt, etc.) and moisture. IP ratings are defined under international standard EN 60529.

load cell: Type of force transducer that converts a force such as tension, compression, pressure, or torque into an electrical signal that can be measured and standardized.

multibody simulation (MBS): A method of numerical simulation in which multibody systems are composed of various rigid or elastic bodies. Connections between the bodies can be modeled with kinematic constraints (such as joints) or force elements (such as spring dampers). Friction elements can also be used to model frictional contacts between bodies.

nondestructive testing (NDT): A testing and analysis technique to evaluate properties of a material, component, structure or system for characteristic differences or welding defects and discontinuities without causing damage to the original part.

purchasing railroad: The term "railroad" and "purchasing railroad" are used throughout the document to identify the authority having jurisdiction over the purchased equipment. This may be a "railroad" as defined by the FRA or a state agency responsible for the purchase of the equipment.

P_2 force: Vertical impulse load on rails caused by track defect or rail joint. See Appendix B for fully detailed definition.

shock load: A special case of dynamic load, defined as a sudden and relatively large increase of load in a system.

special trackwork: Basic special trackwork components include switches, frogs, turnouts, guardrails, crossovers, etc.

static load: Any load, as on a structure, that does not change in magnitude or position with time.

strain gauge: A sensor that is permanently bonded to a structure and whose resistance varies with strain.

superelevation (cant): The difference in elevation (height) between the outside and inside rail on curved track, expressed in inches or millimeters.

tare load: Weight of an empty vehicle

track curvature: Horizontal radius of a railway track. Can be expressed in radius (meters or feet), degrees (subtended by 100 ft cord) or rad/km (radians of rotation per distance along track).

unsprung mass: Mass below primary suspension, typically wheelset and equipment directly attached to the wheelset (journal bearings, brake discs, ground brushes, etc.).

vehicle balance speed: Vehicle speed which results in zero net lateral acceleration at the axle for a given combination of curvature and superelevation.

yielding/yield point: Point on a stress-strain curve that indicates the limit of elastic behavior and the beginning of plastic behavior.

Abbreviations and acronyms

μin	micro-inches
μm	micrometers
AAR	Association of American Railroads
AF	application factor
ABMA	American Boiler Manufacturers Association
ANSI	American National Standards Institute
ASME	American Society of Mechanical Engineers
AW1	weight of vehicle with fully seated passenger load
AW3	weight of vehicle with fully seated passenger load and maximum number of standees
AWS	American Welding Society
C1	shock factor related to roller bearing life evaluation
CFR	Code of Federal Regulations
DIN	Deutsches Institut fur Normung E.V.
EN	Euronorm
FEA	Finite Element Analysis

g	acceleration due to gravity
Hz	hertz
IEC	International Electrotechncial Commission
IP	ingress protection
IP64	ingress protection, Level 6 solids and Level 4 liquid
L10	bearing life with 10 percent failure rate
MBS	multibody simulation
mm	millimeters
mph	miles per hour
MU	multiple unit
NDT	nondestructive testing
NEMA	National Electrical Manufacturers Association
P_2	vertical wheel-rail impulse force
PV	pressure vessel
R_a	roughness average of the ground surface
UIC	International Union of Railways

Summary of document changes

The second revision of this recommended practice completely revises all previously existing sections and adds several new sections. Sections related to vehicle dynamic performance have been removed in lieu of updated guidance in 49 CFR 213 and a forthcoming APTA vehicle dynamics performance standard.

- Title Changed from “New Truck Design” to “Truck Design”
- Document formatted to the new APTA standard format.
- Sections have been moved and renumbered.
- “Summary” and “Scope and purpose” moved to the front page.
- Definitions, abbreviations and acronyms moved to the rear of the document.
- Two new sections added: “Summary of document changes” and “Document history.”
- Some global changes to section headings and numberings resulted when sections dealing with references and acronyms were moved to the end of the document, along with other changes, such as capitalization, punctuation, spelling, grammar and general flow of text.
- Participants updated.
- Added infinite life and cumulative damage methods to Section 3.1, “Structural analysis.”
- Added Section 3.1.4, “Truck mounted equipment design loads.”
- Added Section 3.1.5, “Structural validation plan.”
- Added Section 3.3, “P_2 dynamic track forces.”
- Consolidated clearance study requirements into Section 3.4.
- Added Section 3.5, “Helical coil spring design guidance.”
- Added Section 3.6, “Rubber suspension components”
- Added Section 3.7, “Pneumatic, electrical and electronic equipment design analysis.”
- Added Section 3.8, “Axles, wheels and roller bearings.”
- Added Section 3.9, “Wind loading analysis.”
- Added infinite life and cumulative damage test methods to Section 4.1, “Structural integrity.”
- Added Section 4.3, “Structural model validation.”
- Added Appendix A, “Truck structural design criteria.”
- Added Appendix B, “P_2 dynamic track forces.”

Document history

Document Version	Working Group Vote	Public Comment/ Technical Oversight	Rail CEO Approval	Policy & Planning Approval	Publish Date
First published	—	—	—	March 4, 1999	March 17, 1999
First revision	—	—	—	—	Feb. 13, 2004
Second revision	Nov. 3, 2021	Mar, 31, 2021	Apr. 23, 2021	May 26, 2021	May 28, 2021

The passenger rail industry originally phased this recommended practice into practice over the six-month period from July 1 to Dec. 31, 1999. The recommended practice took effect Jan. 1, 2000.

Appendix A (informative): Truck structural design criteria

This appendix provides a tabulation of historic analysis and test loads used for bolster and truck frame design. Loads have been grouped by vehicle type and include static, exceptional and fatigue loads. This appendix is informative and does not contain requirements that must be evaluated for demonstrating compliance to this recommended practice.

Static cases represent the maximum normal service loads and are generally compared to some fraction of material yield strength. Exceptional cases represent extreme loads and must be applied to the structure without permanent deformation. Fatigue cases represent expected service loads and are limited by the endurance strength of base material and fatigue threshold of welds (infinite life method). In some cases, the cumulative damage method to fatigue is also allowed.

The following important notes must be considered regarding Appendix A:

1. The appendix has been included as reference information to support future design efforts but is not intended to establish a single best practice for analysis and test loads.
2. The information included in the appendix has been gathered from technical specifications, analysis reports and test reports that may have been modified or revised prior to final design.
3. The loads are tabulated using a common set of variables to the greatest extend possible. Some loading equations were reconfigured to use the common variables and therefore will appear different than the source documents.
4. All documents included in the “Standards” section were current at the time of appendix publication but may have since been superseded.

Truck STATIC Design/Test Loads

Agency/Standard		STANDARDS							ELECTRIC MULTIPLE UNITS		
		UIC Code	**UIC Code**	**EN 13749:2011**	**PRIIA 305-001**	**PRIIA 305-003**	**PRIIA 305-007**	**PRIIA 305-009**	**METRA**	**SEPTA**	**LIRR**
Spec No. Date		515-4 1st Ed. 1993	615-4 2nd Ed. 2003	Issue 2 March 2011	Amtrak 962 Rev C.4 Aug 2016	Amtrak 964 Rev B.1 June 2018	Amtrak 979 Rev- Aug 2011	Amtrak 995 Rev- Sept 2012	Spec No. M-01042 Jan 2004	Conformed Version April 2006	Contract 929 September 2013
Vehicle Type Model No. Vehicle Name		Trailer Bogies	Motive Power Units Bogies	Category B-I, B-II	Bi-Level	Single Level	Trainset	DMU	EMU Gallery Type Highliner	EMU Silverliner V	EMU M-9
Builder		NA	NA	NA	NA	NA	NA	NA	Nippon Sharyo	Hyundai Rotem - Car CSC (Amsted) - Truck	Kawasaki
Vertical Load per truck	Fz	$(m_{vom}+1.2*C_2-2m^+)*g/2$	$(m_v+1.2*C_2-2m^+)*g/2$	$(Mv + 1.2\ P_2 - 2m^+)g/2$	AW2 highest loaded truck	AW3 highest loaded truck	AW3 highest loaded truck	AW3 highest loaded truck	0.5x AW3 car weight	AW3 highest loaded truck	0.55x AW3 car weight
Bounce Factor	β	0.20	0.20	0.20	0.10	0.10	0.10	0.10	0.3	0.10	-
Roll Factor	α	0.10	0.10	0.10	Derive from CoG and lateral load	Derive from CoG and lateral load	Derive from CoG and lateral load	Derive from CoG and lateral load	-	Derive from CoG and lateral load	Derive from CoG and lateral load
Pitch Factor	Φ	-	-	-	Derive from CoG and longitudinal load	Derive from CoG and longitudinal load	Derive from CoG and longitudinal load	Derive from CoG and longitudinal load	-	-	Derive from CoG and longitudinal load
Vertical Load per Side Frame	Fz1 Fz2	$(Fz/2)*(1 + \beta + \alpha)$ $(Fz/2)*(1 + \beta - \alpha)$	$(Fz/2)*(1 + \beta + \alpha)$ $(Fz/2)*(1 + \beta - \alpha)$	$(Fz/2)*(1 + \beta + \alpha)$ $(Fz/2)*(1 + \beta - \alpha)$	$(Fz/2)*(1 + \beta + \alpha + \Phi)$ $(Fz/2)*(1 + \beta - \alpha + \Phi)$	$(Fz/2)*(1 + \beta + \alpha + \Phi)$ $(Fz/2)*(1 + \beta - \alpha + \Phi)$	$(Fz/2)*(1 + \beta + \alpha + \Phi)$ $(Fz/2)*(1 + \beta - \alpha + \Phi)$	$(Fz/2)*(1 + \beta + \alpha + \Phi)$ $(Fz/2)*(1 + \beta - \alpha + \Phi)$	$(Fz/2)*(1 + \beta)$	$(Fz/2)*(1 + \beta + \alpha)$ $(Fz/2)*(1 + \beta - \alpha)$	$(Fz/2)*(1 + \alpha + \Phi)$ $(Fz/2)*(1 - \alpha + \Phi)$
Lateral per truck	Fy	$0.5*((Fz/2)+0.5*m^+g)$ [Approx. 0.25 Fz per truck]	$0.5*((Fz/2)+0.5*m^+g)$ [Approx. 0.25 Fz per truck]	$(Fz + m^+g)/8$ Applied to each axle [Approx. 0.25 Fz per truck]	0.25 Fz	0.25 Fz	0.25 Fz	0.25 Fz	0.25 Fz	0.25 * 1.1 Fz	0.25 Fz
Longitudinal per truck	Fx	-	-	-	0.15 Fz	0.15 Fz	0.15 Fz	0.15 Fz	0.15 Fz	0.15 * 1.1 Fz	0.15 Fz
Lozenging each wheel	Fx1	0.1*((Fz/2)+0.5*m+g)	0.1*((Fz/2)+0.5*m+g)	$0.05 * (Fz + m^+g)$	-	-	-	-	-	-	-
Twist		0.50%	0.50%	0.5% over wheelbase	-	-	-	-	0.674" diagonally opposite base plates	-	1.25% over wheelbase
Accessory Loads		-	-	-	Determined by Contractor	Determined by Contractor	Determined by Contractor	Determined by Contractor	-	±100% of maximum steady state or harmonic dynamic conditions	±100% of maximum steady state or harmonic dynamic conditions
Traction		-	Normal reaction torque plus 1.2x weight on central transom and 3x weight on headstock	1.1x Normal acceleration or deceleration	Determined by Contractor	Determined by Contractor	Determined by Contractor	Determined by Contractor	1/3 GU*13 plus Traction Motor Torque at AW3	-	Maximum steady state torque
Braking		Deceleration rate of 1 m/s^2	Maximum in-service braking	1.1x Service braking	Determined by Contractor	Determined by Contractor	Determined by Contractor	Determined by Contractor	Full service braking at AW3 w/ μ=0.245 at brake shoe	Maximum specified deceleration w/ μ=0.3	Emergency braking
Dampers		1.2x Damper fatigue load (definition speed)	1.5x Damper fatigue load (definition speed)	Reference force (definition speed)	Determined by Contractor	Determined by Contractor	Determined by Contractor	Determined by Contractor	Maximum damping force	-	Force at maximum velocity
Static Test		A complete bogie with suspension is recommended for static testing	Test set-up to allow application and distribution of forces exactly as they act in service.	All loads applied simultaneously	> 75 strain gauges All loads applied simultaneously	> 75 strain gauges All loads applied simultaneously	> 75 strain gauges All loads applied simultaneously	> 75 strain gauges All loads applied simultaneously	179 strain gauges	> 75 strain gauges All loads applied simultaneously	> 75 strain gauges truck and 25 bolster
Allowable Stress		Fatigue strength per ERRI Report B12/RP17, Appendix 6	Fatigue strength per ERRI Report B12/RP17, Appendix 6	Material limits based on European or national standards, either endurance limit or cumulative damage approach	Determined by Contractor using industry standard practice, submit to Customer for approval	Determined by Contractor using industry standard practice, submit to Customer for approval	Determined by Contractor using industry standard practice, submit to Customer for approval	Determined by Contractor using industry standard practice, submit to Customer for approval	80% Yield strength	55% Yield strength	40% Yield strength base material, AWS D1.1 allowable static stress for welds

Agency/Standard		LOCOMOTIVES and POWER CARS				COACHES					
		AMTRAK	GO Transit	AMTRAK	MBTA	METRA	NJT	SCRRA	CALTRANS	SEPTA	AMT
Spec No. Date		Amtrak 588 May 1996	RFP-2004-RE-006 2008	Amtrak 865 Rev 5 2008	Contract 671 July 2010	Spec No. M9960 Dec 2000	March 2003	IFB No. EP142-06 2008	PRIIA 305-001 Rev C.1 Sept 2012	Revision 0 March 2017	Revision 6.2 January 2018
Vehicle Type Model No. Vehicle Name		Power Car & Coach Acela	Diesel-Elec Loco MP40 MPXpress	Electric Locomotive ACS-64 Cities Sprinter	Diesel-Elec Loco HSP46	Gallery Car	Multilevel Coach	Bi-Level Guardian	Bi-Level Intercity	Multilevel Coach	Bi-Level Coach
Builder		Alstom & Bombardier	MotivePower - Loco Bradken - Truck	Siemens	MotivePower - Loco Bradken - Truck	Nippon Sharyo	Bombardier	Hyundai Rotem - Car Bradken - Truck	Nippon Sharyo - Car CSC (Amsted) - Truck	CRRC - Car Bradken - Truck	CRRC - Car Bradken - Truck
Vertical Load per truck	Fz	AW1	RtR average loaded truck	RtR highest loaded truck	RtR average loaded truck	0.5x AW3 car weight	0.55x AW3 car weight	AW3 highest loaded truck	AW2 highest loaded truck	0.55x AW3 car weight	AW3 highest loaded truck
Bounce Factor	β	0.10	0.3	0.10	0.3	0.3	-	-	0.10	-	-
Roll Factor	α	Derive from CoG and lateral load	-	Derive from CoG and lateral load	0.2	-	Derive from CoG and lateral load	Derive from CoG and lateral load	Derive from CoG and lateral load	Derive from CoG and lateral load	Derive from CoG and lateral load
Pitch Factor	Φ	Derive from CoG and longitudinal load	-	Derive from CoG and longitudinal load	-	-	Derive from CoG and longitudinal load	Derive from CoG and longitudinal load	-	Derive from CoG and longitudinal load	Derive from CoG and longitudinal load
Vertical Load per Side Frame	Fz1 Fz2	(Fz/2)*(1 + β + α) (Fz/2)*(1 + β - α)	(Fz/2)*(1 + β)	(Fz/2)*(1 + β + α + Φ) (Fz/2)*(1 + β - α + Φ)	(Fz/2)*(1 + β + α) (Fz/2)*(1 + β - α)	(Fz/2)*(1 + β)	(Fz/2)*(1 + α + Φ) (Fz/2)*(1 - α + Φ)	(Fz/2)*(1 + α + Φ) (Fz/2)*(1 - α + Φ)	(Fz/2)*(1 + β + α) (Fz/2)*(1 + β - α)	(Fz/2)*(1 + α + Φ) (Fz/2)*(1 - α + Φ)	(Fz/2)*(1 + α + Φ) (Fz/2)*(1 - α + Φ)
Lateral per truck	Fy	Load at overturning	0.3 Fz		0.25 Fz + 0.25 m^+	0.3 Fz	0.15 Fz	0.25 Fz	0.25 * 1.1 Fz	0.15 Fz	0.25 Fz
Longitudinal per truck	Fx	0.15 Fz	0.45 Fz	0.15 Fz	Full service braking or maximum tractive effort	(4 * Fz) / 21.95	0.15 Fz	0.15 Fz	0.15 * 1.1 Fz	0.15 Fz	Maximum braking with w/ μ=0.5
Lozenging each wheel	Fx1	-	-	-	-	-	-	-	-	-	-
Twist		-	-	-	1% over wheelbase	-	1 Wheel ± 2.5" at AW0 (yard condition)	-	-	-	-
Accessory Loads		±100% of maximum steady state or harmonic dynamic conditions	Maximum braking w/ μ=0.5	Maximum steady state conditions	-	-	-	-	Determined by Contractor	-	-
Traction		-	Maximum tractive effort	Maximum steady state torque plus five (5) times the weight supported by the truck frame	Maximum tractive effort plus five (5) times the weight supported by the truck frame	-	-	-	Determined by Contractor	Maximum steady state torque	-
Braking		-	Maximum dynamic and air brakes	Max TBU/DBU reaction plus six (6) times the TBU/DBU weight	Full service braking	4 mph/sec Deceleration	Braking rate of 0.15g plus six (6) times TBU/DBU weight in the vertical direction	Emergency braking w/ μ=0.5 plus six (6) times TBU/DBU weight in the vertical direction	Emergency braking at AW2 load	Maximum steady state plus six (6) times TBU/DBU weight in the vertical direction	Emergency braking plus six (6) times TBU/DBU weight in the vertical direction
Dampers		-	-	Peak force from damper at maximum operating velocity	-	-	10x Max expected Service Load	Maximum possible damper force, from manufacturer	Determined by Contractor	-	Maximum possible damper force, from manufacturer
Static Test		> 75 strain gauges All loads applied simultaneously	>50 strain gauges	> 100 strain gauges All loads applied simultaneously	Numerous cases involving combinations of: left and right curving, min and max vertical and lateral, and superposition warp loads.	51 strain gauges All loads applied simultaneously	75-100 strain gauges Vertical, lateral, longitudinal loads applied simultaneously, twist and damper loads applied separately	> 75 strain gauges All loads applied simultaneously	> 75 strain gauges All loads applied simultaneously	75-100 strain gauges All loads applied simultaneously	-
Allowable Stress		40% Yield strength	80% Yield strength	55% Yield strength base material, AWS D1.1 allowable static stress for welds	Modified Goodman diagram endurance limit for base (cast) material	80% Yield strength	55% Yield strength base material, AWS D1.1 allowable static stress for welds 90% Yield strength for damper loads	55% Yield strength base material, AWS D1.1 allowable static stress for welds	< Allowable Stress established by Contractor and approved by Customer	55% Yield strength base material, AWS D1.1 allowable static stress for welds	55% Yield strength base material, AWS D1.1 allowable static stress for welds

Truck EXCEPTIONAL Design/Test Loads

Agency/Standard		STANDARDS								ELECTRIC MULTIPLE UNITS		
		BRB CP/DDE/115	UIC Code	UIC Code	EN 13749:2011	PRIIA 305-001	PRIIA 305-003	PRIIA 305-007	PRIIA 305-009	SEPTA	Denver	LIRR
Spec No. Date		Issue 2 March 1988	515-4 1st Ed. 1993	615-4 2nd Ed. 2003	Issue 2 March 2011	Amtrak 962 Rev C.4 Aug 2016	Amtrak 964 Rev B.1 June 2018	Amtrak 979 Rev- Aug 2011	Amtrak 995 Rev- Sept 2012	Conformed Version April 2006	Revision 01 April 2009	Contract 929 September 2013
Vehicle Type Model No. Vehicle Name		Suburban MU	Trailer Bogies	Motive Power Units Bogies	Category B-I, B-II	Bi-Level	Single Level	Trainset	DMU	EMU Silverliner V	EMU Eagle	EMU M-9
Builder		NA	NA	NA	NA	NA	NA	NA	NA	Hyundai Rotem - Car CSC (Amsted) - Truck	Hyundai Rotem - Car CSC (Amsted) - Truck	Kawasaki
Vertical Load per truck	Fz	Max. Static Pivot Load	$(m_{vom}+C1-2m^{+})*g/2$	$(m_v+C_1-2m^{+})*g/2$	$(Mv + P_1 - 2m^{+})g/2$	AW2 highest loaded truck	AW3 highest loaded truck	AW3 highest loaded truck	AW3 highest loaded truck	AW3 highest loaded truck	AW4 highest loaded truck	1.00x AW3 car weight (2g load)
Bounce Factor	β	1.00	0.40	0.40	0.40	0.50	0.50	0.50	0.50	1.00	0.07	-
Roll Factor	α	Derive from CoG and lateral load	-	-	-	Derive from CoG and lateral load	Derive from CoG and lateral load	Derive from CoG and lateral load	Derive from CoG and lateral load	Derive from CoG and lateral load	Derive from CoG and lateral load	Derive from CoG and lateral load
Pitch Factor	Φ	-	-	-	-	Derive from CoG and longitudinal load	Derive from CoG and longitudinal load	Derive from CoG and longitudinal load	Derive from CoG and longitudinal load	-	-	Derive from CoG and longitudinal load
Vertical Load per Side Frame	Fz1 Fz2	$(Fz/2)*(1 + \beta + \alpha)$ $(Fz/2)*(1 + \beta - \alpha)$	$(Fz/2)*(1 + \beta)$	$(Fz/2)*(1 + \beta)$	$(Fz/2)*(1 + \beta)$	$(Fz/2)*(1 + \beta + \alpha + \Phi)$ $(Fz/2)*(1 + \beta - \alpha + \Phi)$	$(Fz/2)*(1 + \beta + \alpha + \Phi)$ $(Fz/2)*(1 + \beta - \alpha + \Phi)$	$(Fz/2)*(1 + \beta + \alpha + \Phi)$ $(Fz/2)*(1 + \beta - \alpha + \Phi)$	$(Fz/2)*(1 + \beta + \alpha + \Phi)$ $(Fz/2)*(1 + \beta - \alpha + \Phi)$	$(Fz/2)*(1 + \beta + \alpha)$ $(Fz/2)*(1 + \beta - \alpha)$	$(Fz/2)*(1 + \beta + \alpha)$ $(Fz/2)*(1 + \beta - \alpha)$	$(Fz/2)*(1 + \alpha + \Phi)$ $(Fz/2)*(1 - \alpha + \Phi)$
Lateral per truck	Fy	Lesser of 2(Fymax) or half overturning force, applied at CoG	$2*(10^4+ (m_{vom}+C1)*g/12)$ In Newtons	$2*(10^4+ (m_v+C1)*g/12)$ In Newtons	$10^4+(Mv+P_1)g/12$ In Newtons Applied to each axle	0.30 Fz	0.30 Fz	0.30 Fz	0.30 Fz	0.25 * 2.0 Fz	0.15 Fz	0.25 Fz
Longitudinal per truck	Fx	$+ 5.0\ m^{+}$	-	-	-	$1.0\ m^{+}$	$1.0\ m^{+}$	$1.0\ m^{+}$	$1.0\ m^{+}$	0.15 * 2.0 Fz	0.15 Fz	0.15 Fz
Lozenging each wheel	Fx1	-	-	-	$0.10 * (Fz + m^{+}g)$	-	-	-	-	-	$0.10 * (Fz + m^{+})$	-
Twist		60% WUL of diagonal wheels	1% Track twist	100% WUL	Case 1 - Track twist 1.0% Case 2 - Complete wheel unloading, Mv	-	-	-	-	-	2.9% = 3/102	1.25% over wheelbase
Accessory Loads		-	-	-	-	Determined by Contractor	Determined by Contractor	Determined by Contractor	Determined by Contractor	-	±100% of maximum steady state or harmonic dynamic conditions	2x Maximum steady state or harmonic dynamic conditions
Traction		-	-	-	1.3x Maximum acceleration or deceleration	Determined by Contractor	Determined by Contractor	Determined by Contractor	Determined by Contractor	-	Based on μ=0.15	Short circuit torque
Braking		-	-	-	Emergency braking	Max. specified deceleration	Max. specified deceleration	Max. specified deceleration	Max. specified deceleration	Maximum specified deceleration w/ μ=0.3	Based on 3 mph/sec deceleration or μ=0.14	2x Emergency braking
Dampers		-	-	-	2x Reference force (definition speed)	Determined by Contractor	Determined by Contractor	Determined by Contractor	Determined by Contractor	-	-	1.25x Maximum allowable damper force
Exceptional Test		Lateral, longitudinal and twist loads applied with max. static pivot load.	Bogie frame shall be fitted with strain gauges and rosette strain gauges at all highly-stressed points, particularly where stresses are concentrated	Locations of strain gauges shall be defined using the results of finite-element analysis.	All loads applied simultaneously	> 75 strain gauges All loads applied simultaneously	> 75 strain gauges All loads applied simultaneously	> 75 strain gauges All loads applied simultaneously	> 75 strain gauges All loads applied simultaneously	> 75 strain gauges All loads applied simultaneously	-	> 75 strain gauges truck and 25 bolster
Allowable Stress		No permanent deformation	No permanent deformation	No permanent deformation	No permanent deformation	No permanent deformation	No permanent deformation	No permanent deformation	No permanent deformation	No permanent deformation	Yield strength of the material	Yield strength of the material

Agency/Standard	LOCOMOTIVES and POWER CARS	COACHES

		AMTRAK	NJT	AMTRAK	NJT	SFRTA	MBTA	AMTRAK	NJT	SCRRA	CALTRANS	SEPTA	AMT
Spec No. Date		Amtrak 588 May 1996	Contract 07-035 2001	Amtrak 865 Rev 5 2008	Contract 07-062 Aug 2008	ITB No. 10-005 June 2010	Contract 671 July 2010	Amtrak Spec No. 576, Rev. 6, Aug 1994	March 2003	IFB No. EP142-06 2008	PRIIA 305-001 Rev C.1 Sept 2012	Revision 0 March 2017	Revision 6.2 January 2018
Vehicle Type Model No. Vehicle Name		Power Car & Coach Acela	Electric Loco ALP46	Electric Locomotive ACS-64 Cities Sprinter	Dual Mode Loco ALP45	Diesel-Elec Loco BL36PH	Diesel-Elec Loco HSP46	Single Level Viewliner I	Multilevel Coach	Bi-Level Guardian	Bi-Level Intercity	Multilevel Coach	Bi-Level Coach
Builder		Alstom & Bombardier	Bombardier	Siemens	Bombardier	Brookville - Loco Bradken - Truck	MotivePower - Loco Bradken - Truck	Morrison Knudsen / CSC	Bombardier	Hyundai Rotem - Car Bradken - Truck	Nippon Sharyo - Car CSC (Amsted) - Truck	CRRC - Car Bradken - Truck	CRRC - Car Bradken - Truck
Vertical Load per truck	Fz	AW1	RtR average loaded truck	RtR highest loaded truck	RtR average loaded truck	2x (W - A) (2g load)	RtR carbody weight (2g load)	AW3 highest loaded truck	AW0 car weight + 50,000 lb	AW3 highest loaded truck	AW2 highest loaded truck	AW0 car weight + 50,000 lb	AW3 highest loaded truck
Bounce Factor	β	0.10	0.40	-	0.70	-	-	0.50	-	-	0.50	-	-
Roll Factor	α	Derive from CoG and lateral load	-	Derive from CoG and lateral load	-	-	Derive from CoG and lateral load	Derive from CoG and lateral load	Derive from CoG and lateral load	Derive from CoG and lateral load	Derive from CoG and lateral load	-	Derive from CoG and lateral load
Pitch Factor	Φ	Derive from CoG and longitudinal load	-	Derive from CoG and longitudinal load	-	-	Derive from CoG and longitudinal load	-	Derive from CoG and longitudinal load	Derive from CoG and longitudinal load	-	-	Derive from CoG and longitudinal load
Vertical Load per Side Frame	Fz1 Fz2	(Fz/2)*(1 + β + α) (Fz/2)*(1 + β - α)	(Fz/2)*(1 + β)	(Fz/2)*(1 + α + Φ) (Fz/2)*(1 - α + Φ)	(Fz/2)*(1 + β)	(Fz/2)	(Fz/2)*(1 + α + Φ) (Fz/2)*(1 - α + Φ)	(Fz/2)*(1 + β + α) (Fz/2)*(1 + β - α)	(Fz/2)*(1 + α + Φ) (Fz/2)*(1 - α + Φ)	(Fz/2)*(1 + α + Φ) (Fz/2)*(1 - α + Φ)	(Fz/2)*(1 + β + α) (Fz/2)*(1 + β - α)	(Fz/2)	(Fz/2)*(1 + α + Φ) (Fz/2)*(1 - α + Φ)
Lateral per truck	Fy	Load at overturning	2*(10^4+RtR/12) In Newtons	0.25 Fz	2*(10^4+RtR/12) In Newtons	0.5x (W - m^+)	0.25 Fz	0.30 Fz	0.15 Fz	Load at overturning	0.30 Fz	0.15 Fz	Load at overturning
Longitudinal per truck	Fx	0.15 Fz	-	0.25 Fz	-	0.25 Fz	0.25 Fz	+ 1.0 m^+	0.15 Fz	BCP @ maximum MRP w/ μ=1.0 at wheel-rail	1.0 m^+	0.15 Fz	Maximum braking with w/ μ=0.5
Lozenging each wheel	Fx1	-	-	-	-	-	-	-	-	-	-	-	-
Twist		-	100% WUL	-	1%	-	Complete wheel unloading	-	-	-	-	-	-
Accessory Loads		Determined by Contractor	-	-	-	-	-	-	-	-	Determined by Contractor	-	-
Traction		-	2x Max steady state torque	Maximum applied loads due to short circuit torque	-	Maximum Torque and μ=0.5	-	-	-	-	Determined by Contractor	-	-
Braking		-	-	2x Maximum normal reaction or, BCP @ maximum MRP w/ μ=1.0	-	Braking at 2.5 mph/sec deceleration	-	40% of TBU with 2.75 mph/sec deceleration 60% of DBU with 2.75 mph/sec deceleration	2x Braking rate of 0.15g plus six (6) times TBU/DBU weight in the vertical direction	TBU/DBU: BCP @ maximum MRP w/ μ=1.0 at wheel-rail	Loads acting on brake brackets at 1.5x AW2 weight	TBU/DBU: BCP @ maximum MRP w/ μ=1.0 at wheel-rail	Emergency braking plus six (6) times TBU/DBU weight in the vertical direction
Dampers		-	-	2x Fatigue load	-	-	-	-	-	Maximum possible damper force, from manufacturer	Determined by Contractor	-	Maximum possible damper force, from manufacturer
Exceptional Test		> 75 strain gauges All loads applied simultaneously	-	> 100 strain gauges All loads applied simultaneously	All loads applied simultaneously	Combination of loads to produce the maximum stresses	Case 1 - Vertical load combined with lateral/longitudinal load Case 2 - Reduced vertical load of 1g combined with twist	> 100 strain gauges (50 truck frame, 50 truck bolster) All loads applied simultaneously	75-100 strain gauges All loads applied simultaneously	> 75 strain gauges Lateral and longitudinal loads applied separately	> 75 strain gauges All loads applied simultaneously	75-100 strain gauges All loads applied simultaneously	-
Allowable Stress		No permanent deformation	No permanent deformation	Yield strength of the material	No permanent deformation	90% Yield strength base material, AWS D1.1 allowable static stress for welds	Yield strength of the material	Yield strength of the material	Yield strength of the material	Yield strength of the material	No permanent deformation	Yield strength of the material	Yield strength of the material

Truck FATIGUE Design/Test Loads

Agency/Standard		STANDARDS								ELECTRIC MULTIPLE UNITS			
		BRB CP/DDE/115	UIC Code	UIC Code	EN 13749:2011	PRIIA 305-001	PRIIA 305-003	PRIIA 305-007	PRIIA 305-009	METRA	SEPTA	Denver	LIRR
Spec No. Date		Issue 2 March 1988	515-4 1st Ed. 1993	615-4 2nd Ed. 2003	Issue 2 March 2011	Amtrak 962 Rev C.4 Aug 2016	Amtrak 964 Rev B.1 June 2018	Amtrak 979 Rev- Aug 2011	Amtrak 995 Rev- Sept 2012	Spec No. M-01042 Jan 2004	Conformed Version April 2006	Revision 01 April 2009	Contract 929 September 2013
Vehicle Type Model No. Vehicle Name		Suburban MU	Trailer Bogies	Motive Power Units Bogies	Category B-I, B-II	Bi-Level	Single Level	Trainset	DMU	EMU Gallery Type Highliner	EMU Silverliner V	EMU Eagle	EMU M-9
Builder		NA	NA	NA	NA	NA	NA	NA	NA	Nippon Sharyo	Hyundai Rotem - Car CSC (Amsted) - Truck	Hyundai Rotem - Car CSC (Amsted) - Truck	Kawasaki
Vertical Load per truck	Fz	Laden	$(m_{vom}+1.2*C_2-2m^+)*g/2$	$(m_v+1.2*C_2-2m^+)*g/2$	$(Mv + 1.2\ P_2 - 2m^+)g/2$	AW1 highest loaded truck	AW1 highest loaded truck	AW1 highest loaded truck	AW1 highest loaded truck	0.675x AW1 carbody weight	AW1 highest loaded truck	AW2 highest loaded truck	AW1 highest loaded truck
Bounce Factor	β	± 0.30	0.20	0.20	0.20	± 0.20	± 0.20	± 0.20	± 0.20	± 0.175	± 0.20	± 0.15	± 0.15
Roll Factor	α	Derive from CoG and lateral load	0.10	0.10	± 0.10 Not applied at CG	Derive from CoG and lateral load	Derive from CoG and lateral load	Derive from CoG and lateral load	Derive from CoG and lateral load	-	Derive from CoG and lateral load	Derive from CoG and lateral load	Derive from CoG and lateral load
Pitch Factor	Φ	-	-	-	-	Derive from CoG and longitudinal load	Derive from CoG and longitudinal load	Derive from CoG and longitudinal load	Derive from CoG and longitudinal load	-	-	-	Derive from CoG and longitudinal load
Vertical Load per Side Frame	Fz1 Fz2	(Fz/2)*(1 ± β ± α)	(Fz/2)*(1 ± β ± α)	(Fz/2)*(1 ± β ± α)	(Fz/2)*(1 ± β ± α)	(Fz/2)*(1 ± β ± α ± Φ)	(Fz/2)*(1 ± β ± α ± Φ)	(Fz/2)*(1 ± β ± α ± Φ)	(Fz/2)*(1 ± β ± α ± Φ)	(Fz/2)*(1 ± β)	(Fz/2)*(1 ± β ± α)	(Fz/2)*(1 ± β ± α)	(Fz/2)*(1 ± β ± α ± Φ)
Lateral per truck	Fy	± 0.30 Fz	$0.25*((Fz/2)+0.5*m^+g)$ Quasi-static 0.25*((Fz/2)+0.5*m+g) Dynamic	$0.25*((Fz/2)+0.5*m^+g)$ Quasi-static 0.25*((Fz/2)+0.5*m+g) Dynamic	0.063*(Fz + m+g) Quasi-static 0.063*(Fz + m+g) Dynamic Applied to each axle	± 0.15 Fz	± 0.15 Fz	± 0.15 Fz	± 0.15 Fz	± 0.25 Fz	± 0.15 Fz	± 0.15 Fz	± 0.15 Fz
Longitudinal per truck	Fx	± 0.20 Fz	-	-	-	± 0.15 Fz	± 0.15 Fz	± 0.15 Fz	± 0.15 Fz	± 0.15 Fz	± 0.15 Fz	± 0.15 Fz	± 0.15 Fz
Lozenging each wheel	Fx1	-	0.1*((Fz/2)+0.5*m+g)	0.1*((Fz/2)+0.5*m+g)	$0.05 * (Fz + m^+g)$	-	-	-	-	-	-	-	-
Twist		25% WUL of diagonal wheels	0.50%	0.50%	0.50%	-	-	-	-	-	-	-	1.25% over wheelbase
Accessory Loads		-	-	-	-	Determined by Customer ±100% of maximum steady state or harmonic dynamic conditions	Determined by Customer ±100% of maximum steady state or harmonic dynamic conditions	Determined by Customer ±100% of maximum steady state or harmonic dynamic conditions	Determined by Customer ±100% of maximum steady state or harmonic dynamic conditions	-	Determined by Customer ±100% of maximum steady state or harmonic dynamic conditions	Determined by Customer ±100% of maximum steady state or harmonic dynamic conditions	±100% of maximum steady state or harmonic dynamic conditions
Traction		-	-	-	1.1x Normal acceleration or deceleration	-	-	-	-	Motor torque reaction force	Maximum specified motor current	-	Maximum dynamic braking torque
Braking		-	-	-	1.1x Service braking	-	-	-	-	Full service braking at AW2 w/ μ=0.245 at brake shoe	Maximum brake force under full cylinder pressure	-	Emergency braking
Dampers		-	-	-	Reference force (definition speed)	-	-	-	-	-	-	-	2x Force at nominal service velocity
Fatigue Test No. Cycles		10 M cycles	6M cycles 2M + 20% 2M + 40% Twist load every 10th cycle	10 M cycles 6M 2M + 20% 2M + 40% Twist load every 10th cycle	10 M cycles 6M 2M + 20% 2M + 40% Twist load every 10th cycle	APTA RP-M-009-98 Casting 2M Fabricated up to 14M	APTA RP-M-009-98 Casting 2M Fabricated up to 14M	APTA RP-M-009-98 Casting 2M Fabricated up to 14M	APTA RP-M-009-98 Casting 2M Fabricated up to 14M	179 strain gauges	Truck frame and bolster subjected to 6M cycles	See 1975 Silverliner IV Test 2M@ 60kips 1M@ 72.5kips +21% 1M@ 85.0kips +42% 1M@ 97.5kips +63% 1M@ 110 kips +83%	10.4M total 6.4M 1M +15% 1M +30% 1M + 45% 1M + 60%
Allowable Stress		-	Fatigue strength per ERRI Report B12/RP17, Appendix 6	Fatigue strength per ERRI Report B12/RP17, Appendix 6	Material limits based on European or national standards, either endurance limit or cumulative damage approach	Determined by Contractor for 40 year life, submit to Customer for approval	Determined by Contractor for 40 year life, submit to Customer for approval	Determined by Contractor for 40 year life, submit to Customer for approval	Determined by Contractor for 40 year life, submit to Customer for approval	60% Modified Goodman diagram endurance limit for base material, AWS D1.1 fatigue threshold for welds	< 60% Modified Goodman diagram endurance limit for base material, AWS D1.1 fatigue threshold for welds	Modified Goodman diagram endurance limit for cast material	AWS D1.1 fatigue threshold for base material and welds, 50% of endurance limit for cast material

Agency/Standard		LOCOMOTIVES and POWER CARS						COACHES					
		AMTRAK	NJT	AMTRAK	NJT	SFRTA	MBTA	AMTRAK	NJT	SCRRA	CALTRANS	SEPTA	AMT
Spec No. Date		Amtrak 588 May 1996	Contract 07-035 2001	Amtrak 865 Rev 5 2008	Contract 07-062 Aug 2008	ITB No. 10-005 June 2010	Contract 671 July 2010	Amtrak Spec No. 576, Rev. 4, Aug 1994	March 2003	IFB No. EP142-06 2008	PRIIA 305-001 Rev C.1 Sept 2012	Revision 0 March 2017	Revision 6.2 January 2018
Vehicle Type Model No. Vehicle Name		Power Car & Coach Acela	Electric Loco ALP46	Electric Locomotive ACS-64 Cities Sprinter	Dual Mode Loco ALP45	Diesel-Elec Loco BL36PH	Diesel-Elec Loco HSP46	Single Level Viewliner I	Multilevel Coach	Bi-Level Guardian	Bi-Level Intercity	Multi-Level Coach	Bi-Level Coach
Builder		Alstom & Bombardier	Bombardier	Siemens	Bombardier	Brookville - Loco Bradken - Truck	MotivePower - Loco Bradken - Truck	Morrison Knudsen - Car CSC (Amsted) - Truck	Bombardier	Hyundai Rotem - Car Bradken - Truck	Nippon Sharyo - Car CSC (Amsted) - Truck	CRRC - Car Bradken - Truck	CRRC - Car Bradken - Truck
Vertical Load per truck	Fz	AW1 average loaded truck	RtR average loaded truck	RtR highest loaded truck	RtR average loaded truck	(W - m^{+})	RtR average loaded truck	AW3 highest loaded truck	0.55x AW3 car weight (Max) 0.50 AW0 car weight (Min)	AW2 highest loaded truck	AW1 highest loaded truck	AW3 average loaded truck	AW2 highest loaded truck
Bounce Factor	β	± 0.15	± 0.2	± 0.20	± 0.3	± 0.35	± 0.3	± 0.15 Fz	-	± 0.20	± 0.20	± 0.15	± 0.20
Roll Factor	α	Derive from CoG and lateral load	0.10	Derive from CoG and lateral load	0.2	-	± 0.2	Derive from CoG and lateral load	Derive from CoG and lateral load	Derive from CoG and lateral load	Derive from CoG and lateral load	Derive from CoG and lateral load	-
Pitch Factor	Φ	Derive from CoG and longitudinal load	-	Derive from CoG and longitudinal load	-	-	-	-	Derive from CoG and longitudinal load	Derive from CoG and longitudinal load	-	Derive from CoG and longitudinal load	-
Vertical Load per Side Frame	Fz1 Fz2	(Fz/2)*(1 ± β ± α ± Φ)	(Fz/2)*(1 ± β ± α)	(Fz/2)*(1 ± β ± α ± Φ)	(Fz/2)*(1 ± β ± α)	-	(Fz/2)*(1 ± β ± α)	(Fz/2)*(1 ± β ± α)	-	(Fz/2)*(1 ± β ± α ± Φ)	(Fz/2)*(1 ± β ± α)	(Fz/2)*(1 ± β ± α ± Φ)	(Fz/2)*(1 ± β)
Lateral per truck	Fy	± 0.15 Fz	0.25*((Fz/2)+0.5*m^{+}g) Quasi-static 0.25*((Fz/2)+0.5*m+g) Dynamic	± 0.15 Fz	0.25*((Fz/2)+0.5*m^{+}g) Quasi-static 0.25*((Fz/2)+0.5*m+g) Dynamic	± 0.25 Fz	0.25*((Fz/2)+0.5*m^{+}g) Quasi-static 0.25*((Fz/2)+0.5*m+g) Dynamic	± 0.15 Fz	± 0.15 Fz (Max)	± 0.15 Fz	± 0.15 Fz	± 0.15 Fz	± 0.15 Fz
Longitudinal per truck	Fx	± 0.15 Fz	-	± 0.15 Fz	-	± 0.15 Fz	-	± 0.125 Fz	± 0.15 Fz (Max)	± 0.15 Fz	± 0.15 Fz	± 0.15 Fz	± 0.15 Fz
Lozenging each wheel	Fx1	-	-	-	0.1*((Fz/2)+0.5*m+g)	-	-	-	-	-	-	-	-
Twist		-	0.50%	-	1%	-	1% over wheelbase	-	-	-	-	-	-
Accessory Loads		±100% of maximum steady state or harmonic dynamic conditions	-	±100% maximum steady state values	-	-	-	Determined by Customer ±100% of maximum steady state or harmonic dynamic conditions	-	±100% of maximum steady state values	Determined by Customer ±100% of maximum steady state or harmonic dynamic conditions	-	-
Traction		Maximum dynamic braking torque	Max steady state torque	Max steady state torque plus five (5) times the weight supported by the truck frame	Max steady state torque	Acceleration or deceleration at 3.0 mph/sec	-	-	1 Wheel ± 2.0" at AW3 (worst case revenue service)	-	-	Maximum steady state torque	-

Braking		Maximum brake force under full cylinder pressure w/ μ=0.1	Max steady state torque	Full service BCP w/ μ=0.2, plus four (4) times the TBU/DBU weight	1.1x Emergency braking	Deceleration 3.5 mph/sec	-	40% of TBU with 2.75 mph/sec deceleration 60% of DBU with 2.75 mph/sec deceleration	Braking rate of 0.15g	Full service braking w/μ=0.20 plus six (6) times TBU/DBU weight in the vertical direction	Full service braking @ AW2 load	Full service braking plus six (6) times TBU/DBU weight in the vertical direction	Emergency braking plus six (6) times TBU/DBU weight in the vertical direction
Dampers		-	Force at 0.3 m/s damper velocity	Force when operating at maximum velocity	Derived by vehicle dynamic calculations	-	-	-	2x Max expected Service Load	Maximum possible damper force, from manufacturer	-	Force when operating at maximum velocity	Maximum possible damper force, from manufacturer
Fatigue Test No. Cycles		6M total 2M no cracks 1M + 15% 1M + 30% 1M + 45% 1M + 55%	10 M cycles 6M 2M + 20% 2M + 40% Twist load every 10th cycle UIC 615-4	10 M (no crack) 2M +10% 2M +20%	10 M cycles 6M 2M + 20% 2M + 40% Twist load every 10th cycle	Option 1 - 10M cycle lab test Option 2 - On-track stress testing using instrumented trucks	10M total 6M 2M +20% 2M +40% UIC 615-4	2M cycles	6M total 2M 2M +10% 2M +20%	6M total 2M Base 2M +10%, 2M +20% for cast Increase base cycles to highest weld detail endurance limit for fabricated design	2M cycles* 1M +10% 500k +20% *only 2M cycles was completed before contract cancelled	10M total 6M Base 2M +20%, 2M +40% for cast Increase base cycles to highest weld detail endurance limit for fabricated design	-
Allowable Stress		AWS D1.1 fatigue threshold for base material and welds	Determined for the infinite life according to DS 952 of Deutsche Bahn AG	Modified Goodman diagram endurance limit for base material, AWS D1.1 fatigue threshold for welds	Determined for the infinite life according to DS 952 of Deutsche Bahn AG	Determined by Contractor, submit to Customer for approval. Shall not exceed AWS D1.1 allowables for welded structures	Modified Goodman diagram endurance limit for base (cast) material	Satisfactory if no critical crack is developed after 2 million cycles.	AWS D1.1 fatigue threshold for base material and welds	AWS D1.1 fatigue threshold for base material and welds	Determined by Contractor for 40 year life, submit to Customer for approval	Modified Goodman diagram endurance limit for base material, AWS D1.1 fatigue threshold for welds	AWS D1.1 fatigue threshold for base material and welds

Truck structural design criteria – Variable list

Specification	Variable	Definition
GENERAL	**CoG**	Center of gravity for carbody
	Fz	Vertical load per truck
	Fz1	Vertical load per side frame, quasi-static loaded side
	Fz2	Vertical load per side frame, quasi-static unloaded side
	β	Bounce dynamic vertical load factor (track surface)
	α	Roll quasi-static vertical load factor (curving)
	Φ	Pitch quasi-static vertical load factor (traction and braking)
	m^+	Truck (bogie) mass or weight
	μ	Coefficient of friction
	TBU	Tread brake unit
	DBU	Disk brake unit
	AW0	Passenger car weight, empty and ready for passenger service
	g	Gravitational constant, 9.81 m/s^2
AMT	**AW2**	Passenger car weight, AW0 plus fully seated plus 2.4 standees per m^2
Bi-Level Coach	**AW3**	Passenger car weight, AW0 plus fully seated plus 6.0 standees per m^2
	p.w	Passenger mass 155 lb (70 kg) each
Amtrak	**BCP**	Brake cylinder pressure
Elec Loco	**MRP**	Main reservoir pressure
	RtR	Ready-to-run locomotive weight
Acela	**AW1**	Passenger car weight, AW0 plus fully seated passenger load.
PC & Coach		
AMTRAK	TBD	TBD when tech spec is provided
Viewliner I		
BRB CP/DDE/115	**BRB**	British Railways Board
Suburban MU	**Fymax**	Maximum load in lateral direction. Defined as the maximum net lateral track force (kN) exerted on the track by a wheelset which is sustained for any 2m of wheelset movement. Peak forces, if not sustained for 2m are not considered
	WUL	Wheel unloading
	Laden	Normal full passenger load, approximately equal to AW2
CALTRANS	**AW1**	Passenger car weight, AW0 plus seated load plus one crew
Bi-Level	**AW2**	Passenger car weight, AW1 plus one standee per 3 ft^2 (0.3 m^2)
Denver	**AW1**	Passenger car weight, AW0 plus seated load plus one crew
Eagle	**AW2**	Passenger car weight, AW1 plus four standees per m^2
	AW4	Passenger car weight, AW1 plus eight standees per m^2
	p.m	Passenger mass 165 lb each

Specification	Variable	Definition
EN 13749:2011	**Mv**	Vehicle mass, empty in running order
Category B-I, B-II	**P_1**	Exceptional design payload, fully seated plus four standees per m^2, see EN 15663
	P_2	Nominal service payload, all normal seats occupied, see EN 15663 for details
	p.m	Passenger mass 80 kg each
GO Transit	**m.v**	Empty weight of vehicle in running order
Diesel-Elec. Loco.	**w.tm**	Traction motor weight
	w.us	Unsprung weight per truck
	w.wh	Wheelset weight
LIRR	**AW1**	Passenger car weight, AW0 plus seated load plus one crew
M-9	**AW3**	Passenger car weight, AW1 plus one standee per 1.8 ft^2
	p.m	Passenger mass 165 lb each
MBTA	**Fy**	Load in lateral direction
Diesel-Elec. Loco.	**Fz**	Load in vertical direction
	m.v	Empty weight of vehicle in running order
	m^+	Bogie mass or weight
	n.b	Number of bogies
	α	Roll factor
	β	Bounce factor
METRA	**AW3**	Passenger car weight, AW0 plus fully seated plus 100 standees
Gallery Car	**p.w**	Passenger mass 155 lb each
METRA	**AW1**	Passenger car weight, AW0 plus fully seated
EMU Two Level	**AW2**	Passenger car weight, AW0 plus fully seated plus 50 standees
	AW3	Passenger car weight, AW0 plus fully seated plus 100 standees
	GU	Gear unit weight, 1080 lb
	p.w	Passenger mass 155 lb each
NJT Dual Mode Loco	**RtR**	Ready-to-run locomotive weight
NJT Electric Loco	**RtR**	Ready-to-run locomotive weight
NJT	**AW3**	Passenger car weight, AW0 plus fully seated plus 177 standees (car car)
Multilevel Coach	**p.w**	Passenger mass 165 lb each
PRIIA 305-001	**AW1**	Passenger car weight, AW0 plus seated load plus one crew
Bi-Level	**AW2**	Passenger car weight, AW1 plus one standee per 3 ft^2 (0.3 m^2)
	AW3	Passenger car weight, AW1 plus one standee per 1.5 ft^2 (0.14 m^2)
	p.m	Passenger mass 180 lb (82 kg) each
PRIIA 305-003	**AW1**	Passenger car weight, AW0 plus seated load plus one crew
Single Level	**AW2**	Passenger car weight, AW1 plus one standee per 3 ft^2 (0.3 m^2)
	AW3	Passenger car weight, AW1 plus one standee per 1.5 ft^2 (0.14 m^2)

Specification	Variable	Definition
	p.m	Passenger mass 180 lb (82 kg) each
PRIIA 305-007 Trainset	**AW1** **AW2** **AW3** **p.m**	Passenger car weight, AW0 plus seated load plus one crew Passenger car weight, AW1 plus one standee per 3 ft^2 (0.3 m^2) Passenger car weight, AW1 plus one standee per 1.5 ft^2 (0.14 m^2) Passenger mass 180 lb (82 kg) each
PRIIA 305-009 DMU	**AW1** **AW2** **AW3** **p.m**	Passenger car weight, AW0 plus seated load plus one crew Passenger car weight, AW1 plus one standee per 3 ft^2 (0.3 m^2) Passenger car weight, AW1 plus one standee per 1.5 ft^2 (0.14 m^2) Passenger mass 180 lb (82 kg) each
SCRRA Bi-Level Guardian	**AW2** **AW3** **BCP** **MRP** **p.w**	Passenger car weight, AW0 plus fully seated plus 80 standees Passenger car weight, AW0 plus fully seated plus 215 standees Brake cylinder pressure Main reservoir pressure Passenger mass 165 lb each
SEPTA Multilevel Coach	**AW3** **p.w**	Passenger car weight, AW0 plus fully seated plus 6.0 standees per m^2 Passenger mass 165 lb each
SEPTA Silverliner V	**AW1** **AW3** **p.m**	Passenger car weight, AW0 plus 18,204 lb Passenger car weight, AW0 plus 36,900 lb Passenger mass 164 lb each
SFRTA Diesel-Elec. Loco.	**A** **W**	Weight of axle and axle-mounted components Weight per truck at rail with fully loaded locomotive +5%
UIC Code Motive Power Units	$\mathbf{C_1}$ $\mathbf{C_2}$ $\mathbf{m_v}$ **p.m**	Passenger load, fully seated plus four standees per m^2 Passenger load, fully seated plus two standees per m^2 Empty weight of vehicle in running order Passenger mass 80 kg each
UIC Code Trailer Bogies	$\mathbf{m_{vom}}$ $\mathbf{C_1}$ $\mathbf{C_2}$ **p.m**	Vehicle mass, empty in running order Passenger load, fully seated plus four standees per m^2 Passenger load, fully seated plus two standees per m^2 Passenger mass 80 kg each

Appendix B (informative): P_2 dynamic track forces

This appendix provides background information related to the evaluation of P_2 forces on rail vehicles. According to D.R. Ahlbeck [Ref. 4], the following forces occur when a rail vehicle encounters a sharp track defect or rail joint:

> The first impact force, called the P_1 force peak, results from the wheel impacting the end of the rail onto which it is running. This P_1 force occurs 1/4 to 1/2 millisecond after the wheel crosses the gap in the rail ends. The second load impulse, called P_2, occurs 5 to 10 milliseconds later in the vicinity of the first running-on tie. The P_1 force has substantial high-frequency content in the range of 1000 to 2000 Hz and results primarily from the wheel/rail Hertzian contact stiffness and the rail mass. The P_2 forces are of lower frequency content in the range of 20 to 100 Hz and can be transmitted readily to the ties and ballast. Consequently, the P_1 force is associated with rail end batter, while the P_2 force is associated with the development of a depressed joint profile due to tie, ballast and subgrade deterioration.

This appendix gives guidance on calculation and evaluation of P_2 forces. This appendix is informative and does not contain requirements that must be evaluated for demonstrating compliance to this recommended practice.

B.1. Background

Jenkins created the first P_2 equation in 1974, followed later by Ahlbeck's interpretation in 1980. Amtrak documents from as late as 1994 show the Ahlbeck equation still in use at that time. However, starting in 1993 British Rail began using its own form of the P_2 equation as shown below [Ref. 1]:

The P2 force shall be calculated using the following formula:

$$P_2 = Q + (A_z . V_m . M.C.K)$$

where

$$M = \left[\frac{M_v}{M_v + M_z}\right]^{0.5}$$

$$C = 1 - \left[\frac{\pi . C_z}{4[K_z(M_v + M_z)]^{0.5}}\right]$$

$$K = \left(K_z . M_v\right)^{0.5}$$

Q = maximum static wheel load (N)

V_m = maximum normal operating speed (m/s)

M_v = effective vertical unsprung mass per wheel (kg)

A_z = 0.020 rad
(total angle of vertical ramp discontinuity)

M_z = 245 kg
(effective vertical rail mass per wheel)

$C_z = 55.4 \times 10^3$ Ns/m
(effective vertical rail damping rate per wheel)

$K_z = 62.0 \times 10^6$ N/m
(effective vertical rail stiffness per wheel)

B.2 Current best practice for P_2 forces

A difficulty with the rendering of the original British Rail formula is that each section of the equation is written on a different line, so it is necessary to refer to each line before completing the P_2 equation at the top. Starting in 2010, this formula was consolidated (but the inputs and output not altered) into the single-line formula shown below:

$$P_2 = P_0 + 2\alpha v \sqrt{\frac{m_u}{m_u + m_t}} \left(1 - \frac{\pi c_t}{4\sqrt{k_t(m_u + m_t)}}\right) \sqrt{k_t m_u}$$

British Rail P_2 force equation from "AMTRAK - Allowable Speeds for P_2 force of 82k_M.Trosino_10-11-2010 - Vehicle Summary" – October 2010

This formula is suitable for use with either metric or English units. The terms used are listed below with example units.

Term	Description	Metric Unit	English Unit
P_2	Dynamic vertical rail force	kN	lbf
P_0	Maximum static wheel load	kN	lbf
2α	Total dip angle	radians	radians
v	Maximum normal operating speed	m/s	in/sec
m_u	Unsprung mass per wheel	kg	lbf·sec^2/in
m_t	Effective vertical rail mass per wheel	kg	lbf·sec^2/in
c_t	Effective vertical rail damping per wheel	kN·s/m	lbf·sec/in
k_t	Effective vertical rail stiffness per wheel	MN/m	lbf/in

Consistent units are critical to proper use of the P_2 force expression. The principal issue comes with mass values in English units. At standard gravity (9.80665 m/s^2), a per wheel weight of 'Q' lb-force (lbf) means a mass of Q lb-mass (lb). With English units, the P_2 equation requires mass values expressed as a weight divided by standard gravity. The units of the gravity term are dictated by the length unit used in the operating speed and the rail stiffness and damping values. For inches, the gravity term is 386.09 in/sec^2. As an example, this means a value of 5.180 (i.e., 2,000/386.09) for a per wheel weight of 2,000 lbf (or mass of 2,000 lb). The corresponding units are lbf·sec^2/in as listed in the example units shown above. This adjustment is not necessary when metric units are used.

Today the above formula, which is simply referred to as the British Rail Equation, can be found in the most recent versions of the PRIIA (locomotive, trainset, DM loco and DMU) specifications. Additionally, the British Rail Equation is currently in use by multiple agencies, such as Metro North Railroad, New Jersey Transit, EXO (Montreal), VIA Rail, and Amtrak. Given the extent of its use, it is considered best practice to use the British Rail Equation for any P_2 force calculations.

B.3 Guidance to calculate unsprung mass for P_2 forces

The following discussion considers unsprung mass per wheelset. The resulting values need to be halved for use in the British Rail Equation which views the calculation as being "per wheel".

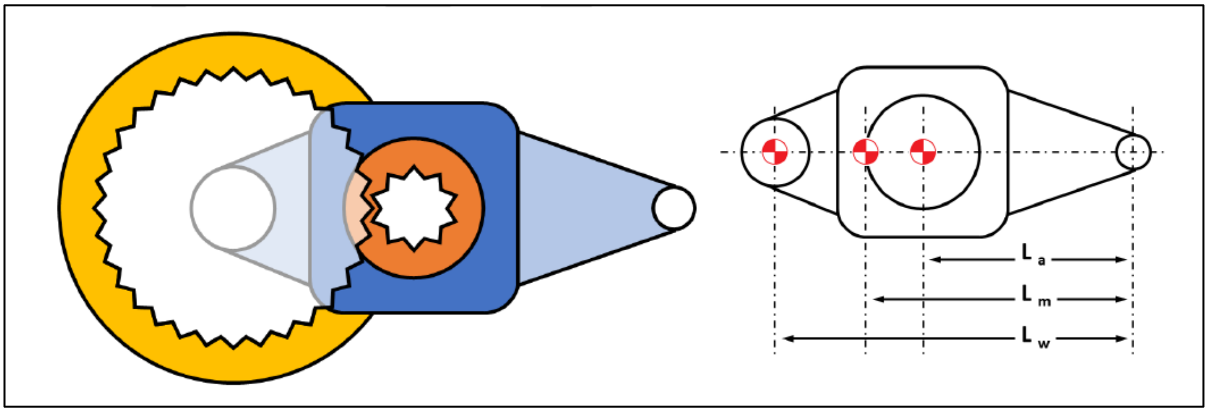
1. For the general unsprung mass of a wheelset as well as the unsprung mass of a rigid frame undriven bogie with primary suspension, the following unsprung mass equation rules apply:
    a. These parts have 100 percent of their mass included in the wheelset unsprung mass calculations: axles, non "swing arm type" axle boxes, axle-mounted brake discs, wheel-mounted (cheek) brake discs, axle bearings, wheels, and other miscellaneous axle-mounted equipment such as speed sensors, axle box-mounted current collectors and grounding brush assemblies.
    b. These parts have 50 percent of their mass unsprung for the wheelset unsprung mass calculations: the primary suspension springs, the primary suspension dampers, and other miscellaneous symmetrical geometry connecting from the wheelset to the truck frame or the body (such as rods, links and cables).
2. For a traction motor that is "axle-hung" (also referred to as axle-mounted, i.e., primarily supported by the axle), the effective unsprung mass of the wheelset can be calculated using the equation below. This equation takes into consideration motor and armature rotational inertia which increase the active unsprung mass. The equation assumes the classic, single stage gear arrangement consisting of a wheelset-mounted drive gear and an armature-mounted pinion. The effective unsprung mass will be less than that calculated by the equation if there is significant drivetrain torsional flexibility between the axle and the armature. More complex arrangements, a multi-stage gearbox for example, will change the influence of the armature pitch inertia.

$$m_u = m_w + \frac{1}{{L_w}^2}\left[{L_m}^2.m_m + I_m + {L_a}^2.m_a + \left(n_p + 1\right)^2.I_a\right]$$

This equation may be used with any consistent set of units. The terms used are listed below with example units.

Term	**Description**	**Metric Unit**	**English Unit**
I_a	Armature pitch rotational inertia	kg·m^2	lb·in^2
I_m	Traction motor case pitch rotational inertia	kg·m^2	lb·in^2
L_a	Armature center of gravity longitudinal offset relative to truck frame support (see **Figure 3**)	m	in
L_m	Traction motor case center of gravity longitudinal offset relative to truck frame support (see **Figure 3**)	m	in
L_w	Wheelset center of gravity longitudinal offset relative to truck frame support (see **Figure 3**)	m	in
m_a	Armature mass	kg	lb
m_m	Traction motor case mass	kg	lb
m_u	Effective unsprung mass per wheelset	kg	lb
m_w	Wheelset mass	kg	lb
n_p	Pinion gear ratio (wheelset drive gear teeth / motor armature pinion gear teeth)	[-]	[-]

FIGURE 3
Wheelset with axle-hung traction motor



This equation for effective unsprung mass was derived by a member of the recommended practice task group and independently second checked. This equation differs from that provided in Australian Standard AS 7508:2017 in that:

a. It allows for the possibility that the center of gravity of the armature may be longitudinally offset from that of the motor case.
b. It separates the influence of the armature mass from that of the motor case.
c. It expresses the pinion gear ratio, n_p, as wheelset drive gear teeth / motor armature pinion gear teeth.
d. The form of the equation separates the influence of the wheelset mass from that of the traction motor.

3. The above equation for a traction motor can also be used for further "axle-hung" items also connected to the truck frame. Examples are a swing arm type axle box or a gearbox or quill drive for a frame-mounted traction motor. The contribution of these items, based on mass and geometry considerations, is given by the following expression:

$$m_{u+} = \frac{1}{{L_s}^2}\left[{L_c}^2 . m_c + I_c\right]$$

This equation may be used with any consistent set of units. See below for a definition of terms with example metric and English units.

Term	Description	Metric Unit	English Unit
I_c	Component pitch rotational inertia	kg m^2	lb in^2
L_c	Component center of gravity longitudinal offset relative to truck frame support	m	in
L_s	Axle center offset relative to truck frame support for component	m	in
m_{u+}	Effective additional unsprung mass per wheelset	kg	lb

Further guidance is as follows.

a. For swing arm type axle boxes, the offset L_c is the longitudinal offset of the swing arm center of gravity to the attachment bushing on the truck frame. The dimension L_s is the longitudinal offset between the axle center and the attachment bushing on the truck frame.
b. For gearboxes or axle drives, the offset L_c is the longitudinal offset of the component center of gravity to the attachment bushing on the truck frame or reaction link attaching to truck frame. The dimension L_s is the longitudinal offset between the axle center and the attachment to the truck frame. Gearbox mass and center of gravity calculation should include 50 percent of the mass of the coupling to the traction motor and/or the mass of any reaction rod between the gearbox and truck frame. Note that gearboxes and axle drives may contribute further influences arising from referred inertia of the motor armature reacted through the drivetrain. These influences may further increase the effective unsprung mass, even if the motor unit is mounted directly to the truck frame.

B.4 P_2 Forces for PRIIA and transit agencies

For illustration purposes, the following tables present a selection of input parameters and P_2 force limits taken from a range of sources, including:

1. PRIIA specifications
2. British and Australian standards
3. Practice adopted by various transit agencies
4. Values used for evaluation of P_2 forces for a selection of example vehicles

Where applicable, source values have been converted from metric to English units in the following tables.

TABLE 2
Specifications and Standards

	PRIIA			**Other**		
Spec / Standard Revision & Date	**Locomotive** Rev B June 2017	**Dual-Mode Locomotive** Rev A January 2015	**DMU** Initial Release Sept 2012	**British Rail Standard GM/TT0088** Rev A October 1993	**RISSB Australian Standard AS 7508** Edition 2017	**Units**
Total dip angle (2α)	**0.017**	**0.017**	**0.017**	**0.020**	**0.010 - 0.014**	**radians**
Maximum speed (v)	**Per Vehicle Characteristics**					**mph**
						in/sec
Axle load						**lbf**
Static wheel load (P_0)						**lbf**
Unsprung mass per axle						**lbf·sec²/in**
Unsprung mass per wheel (m_u)						**lbf·sec²/in**
Eff. rail mass per wheel (m_t)	**1.1335**	**1.1335**	**1.1335**	**1.399**	**0.6681 - 1.9300**	**lbf·sec²/in**
	438	**438**	**438**	**540**	**258 - 745**	**lbf**
Eff. rail damping per wheel (c_t)	**671**	**671**	**671**	**317**	**274 - 320**	**lbf·sec/in**
Eff. rail stiffness per wheel (k_t)	**392900**	**392900**	**392900**	**354331**	**548175 - 668088**	**lbf/in**
Basis of P2 Calculation	**British Railways Board Group Standard GM/TT0088 Issue 1, Rev. A**					**-**
P2 Limit	**82000**	**82000**	**82000**	**72389**	**39342 - 66319 depending on rail network/segment**	**lbf**

TABLE 3

P₂ Force Limits

	Transit Agencies					
Agency	**Metro North Railroad**	**Metrolinx Go Transit**	**New Jersey Transit & AMT (Quebec)**	**Via Rail (Canada)**	**Amtrak**	**Units**
Total dip angle (2α)	**0.017**	**0.020**	**0.017**	**0.020**	**0.017**	**radians**
Maximum speed (v)	**125**	**90**	**125**	**100**	**150**	**mph**
	2200	**1584**	**2200**	**1760**	**2640**	**in/sec**
Axle load	**Per Vehicle Characteristics**					**lbf**
Static wheel load (P_0)						**lbf**
Unsprung mass per axle						**lbf·sec²/in**
Unsprung mass per wheel (m_u)						**lbf·sec²/in**
Eff. rail mass per wheel (m_t)	**1.1335**	**1.5447**	**1.1335**	**1.5464**	**1.1335**	**lbf·sec²/in**
	438	**597**	**438**	**597**	**438**	**lbf**
Eff. rail damping per wheel (c_t)	**671**	**205**	**671**	**205**	**671**	**lbf·sec/in**
Eff. rail stiffness per wheel (k_t)	**330000**	**226042**	**330000**	**226040**	**392900**	**lbf/in**
Basis of P2 Calculation	**British Railways Board Group Standard GM/TT0088 Issue 1, Rev. A**					**-**
P2 Limit	**71000**	**75000**	**75513**	**80000 lbs (356KN) on Class 5 Track at 100 mph (160 km/h).**	**82000**	**lbf**

TABLE 4

Example Vehicles

	Amtrak & NJT					
Vehicle Vehicle Type	**AEM-7** Electric Locomotive	**Acela** Power Car	**ALP46** Electric Locomotive	**P42** Diesel Locomotive	**ACS-64** Electric Locomotive	**Units**
Total dip angle (2α)	**0.017**	**0.017**	**0.017**	**0.017**	**0.017**	**radians**
Maximum speed (v)	**125**	**150**	**125**	**110**	**125**	**mph**
	2200	**2640**	**2200**	**1936**	**2200**	**in/sec**
Axle load	**50000**	**53000**	**50000**	**66000**	**53000**	**lbf**
Static wheel load (P_0)	**25000**	**26500**	**25000**	**33000**	**26500**	**lbf**
Unsprung mass per axle	**7300**	**5960**	**5300**	**8000**	**7200**	**lb**
Unsprung mass per wheel (m_u)	**9.45**	**7.72**	**6.86**	**10.36**	**9.32**	**lbf·sec²/in**
Eff. rail mass per wheel (m_t)	**1.1335**	**1.1335**	**1.1335**	**1.1335**	**1.1335**	**lbf·sec²/in**
	438	**438**	**438**	**438**	**438**	**lbf**
Eff. rail damping per wheel (c_t)	**671**	**671**	**671**	**671**	**671**	**lbf·sec/in**
Eff. rail stiffness per wheel (k_t)	**392900**	**392900**	**392900**	**392900**	**392900**	**lbf/in**
P2 Force	**75513**	**78857**	**64982**	**80409**	**76521**	**lbf**
P2 Limit	**82000**	**82000**	**82000**	**82000**	**82000**	**lbf**

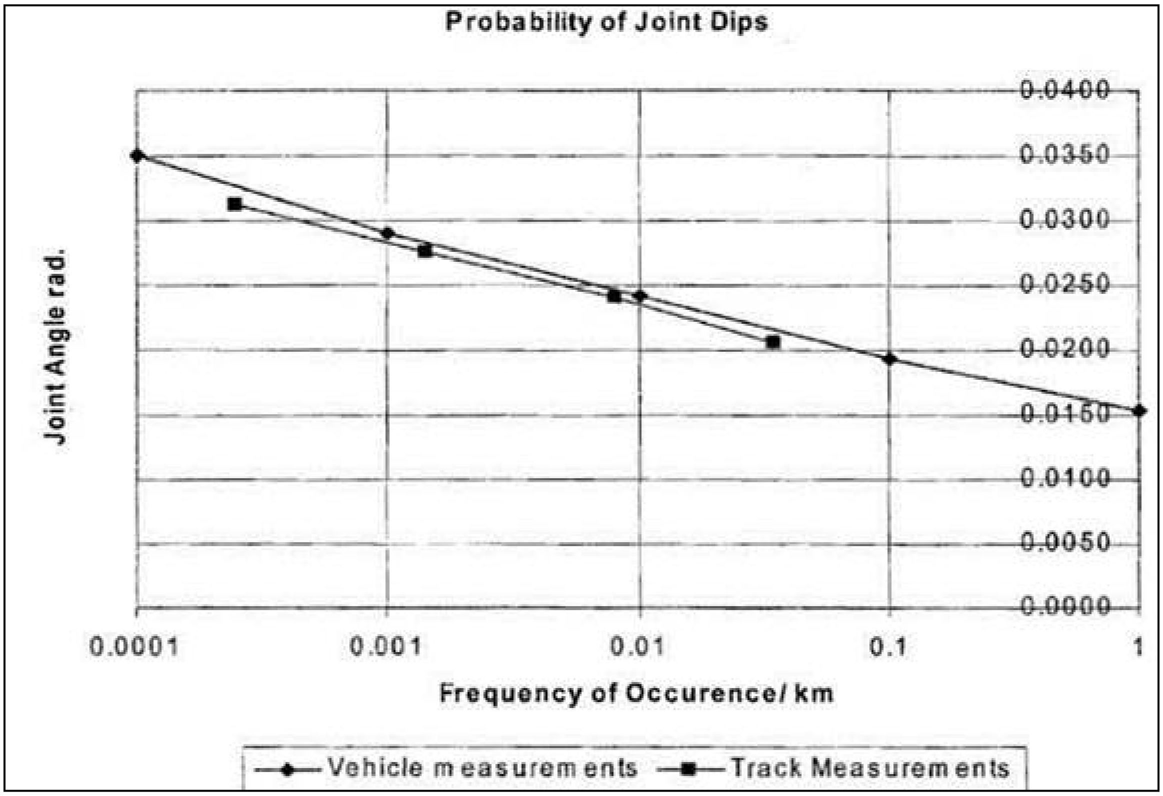
Notes:

1. Unsprung mass in the table above has been calculated by the respective agencies and may not have been calculated using the guidance in Section B.3.
2. The track damping value of 671 lbf·sec/in was assumed for NEC track conditions and corresponds closely with the value cited in Esveld [Ref. 2] of 120,000 N·s/m.
3. The track stiffness value of 392,900 lbf/in is for track with concrete ties and comes from assuming a track deflection under load of 0.25 in, and a track modulus of 4000 lbf/in², and nominal rail cross-section producing the EI value. From this, the track stiffness (P/y) is obtained from the track modulus equation and solving for track stiffness (P/y):

$$u = 4000\frac{lbf}{in^2} = \frac{\left(\frac{P}{y}\right)^{4/3}}{(64EI)^{1/3}}$$

4. The size of the rail dip of 0.017 radians comes from British Rail research [Ref. 3] where they measured the size of rail joint dip angles (2α) and found the frequency of occurrence shown in **Figure 4**.

FIGURE 4
Probability of Joint Dips from British Rail Research



At the time of measurement (during the 1960s and early 1970s) there were still many rail joints that have since been removed when continuously welded rail (CWR) was installed. Therefore, the current distribution would have smaller dip angles than shown in **Figure 4**. It is noted that a dip angle (2α) of 1.0 deg. (0.017 radians, or 0.0085 radians on each side) was measured approximately every 3 km and was therefore relatively common. A dip angle of 2.0 deg. (0.035 radians) was very rare, being measured once every 10,000 km, and so was considered excessive for use in the P_2 force equation. Therefore, it will be assumed for purposes of calculation that a typical battered rail joint with de-flection under loading provides a dip angle (2α) of 1.0 deg.

References

1. British Rail document GM/TT0088 “Permissible Track Forces for Railway Vehicles,” October 1993.
2. Esveld, C., “Modern Railway Track,” Second Edition, Published by MRT-Productions, 2001
3. The Effect of Track and Vehicle Parameters on Wheel/Rail Dynamic Forces,” The Railway Engineering Journal, Vol. 3, No. 1, I.Mech.E. January 1974
4. D.R. Ahlbeck, “An Investigation of Impact Loads Due to Wheel Flats and Rail Joints,” ASME 80-WA/RT-1, 1981

