Association of American Railroads

SAFETY AND OPERATIONS

MANUAL OF STANDARDS

AND

RECOMMENDED PRACTICES
SECTION C-II

DESIGN, FABRICATION, AND CONSTRUCTION OF FREIGHT CARS

ISSUED NOVEMBER 2023

Compiled under the direction of the Committees responsible for the subjects shown herein.

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MSRP A-I TABLES OF CONTENT
The Section A-I Tables of Content lists the standards, specifications, and recommended practices that are contained in each volume of the AAR *Manual of Standards and Recommended Practices (MSRP).* This information is updated every week, and is available for no cost at https://aar.com/standards/MSRPs/MSRP-A1.pdf.

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TECHNICAL QUESTIONS
For technical questions regarding this manual, use the email address appropriate to this manual from the following list.

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TO THE USER

AAR *Manual of Standards and Recommended Practices (MSRP)* Section C-II—Design, Fabrication, and Construction of Freight Cars contains Specification M-1001. This specification covers minimum requirements for the design and construction of new freight cars that are mandatory for equipment used in interchange service.

It is recommended that Specification M-1001 be applied whenever practicable in repairing freight cars. Any structural subassembly or component of the underframe or superstructure of a car must meet these requirements and be so warranted by suppliers. The car builders must ensure that freight cars meet both AAR and Federal Railroad Administration (FRA) requirements.

USER'S GUIDE

Section C-II contains the following:

- **Preface**: A listing of the subjects covered in the individual volumes making up this manual. This preface is part of each section.
- **Table of Contents:** Identification of the chapters in numerical sequence.
- **Specification M-1001:** The body of this volume is a single specification setting forth mandatory requirements for virtually all phases of freight car design.

RELATED SECTIONS

Other sections in the *MSRP* that contain pertinent information are as follows:

- Section B—Couplers and Freight Car Draft Gear Components
- Section C-I—Car Construction Fundamentals and Details
- Section D—Trucks and Truck Details
- Section E-I—Brakes and Brake Equipment
- Section G—Wheels and Axles
- Section H—Journal Bearings and Lubrication

RESPONSIBILITY

The coverage of Section C-II is the responsibility of the AAR Equipment Engineering Committee.

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PREFACE

The *Manual of Standards and Recommended Practices* of the Safety and Operations Department, Association of American Railroads, is issued by authority of the Management Committee and includes all regularly adopted specifications, standards, and recommended practices of the Association of American Railroads.

The manual is composed of the following sections:

- Section AS—Administrative Standards Supplement serves as a supplement to all *MSRP* sections (this is available as a free download at https://aarpublications.com/msrp.html)
- Section A-I—Table of Contents, Alphabetical and Numerical Index of Sections B through S inclusive (this is available as a free download at https://aarpublications.com/msrp.html)
- Section B—Freight Car Draft Components (100 Series)
- Section C (C-I)—Car Construction Fundamentals and Details (200 and 2000 Series)
- Section C-II—Design, Fabrication, and Construction of Freight Cars, M-1002
- Section C-III—Specifications for Tank Cars, M-1003
- Section D—Trucks and Truck Details (300 and 3000 Series)
- Section E (E-I)—Brakes and Brake Equipment (300, 400, 3000, and 4000 Series)
- Section E-II—Electronically Controlled Brake Systems (4200 Series)
- Section F—Sensors
- Section G (G-I)—Wheels and Axles (600 Series)
- Section G-II—Wheel and Axle [Shop] Manual (600 and 700 Series)
- Section H (H-I)—Journal Bearings and Lubrication (700 Series)
- Section H-II—Roller Bearing [Shop] Manual (700 Series)
- Section I—Intermodal Equipment Manual
- Section J—Specification for Quality Assurance, M-1003
- Section K-I—Railway Electronics Systems Architecture and Concepts of Operation (9000 Series)
- Section K-II—Locomotive Electronics and Train Consist System Architecture (9100 Series)
- Section K-III—Wayside Electronics and Mobile Worker Communications Architecture (9200 Series)
- Section K-IV—Office Architecture and Railroad Electronics Messaging (9300 Series)
- Section K-V—Electronics Environmental Requirements and System Management (9400 Series)
- Section K-VI—Railway Data Management and Communications (9500 Series)
- Section M—Locomotives and Locomotive Interchange Equipment (500 and 5000 Series)
- Section N—Multi-Level Manual (800 Series)
- Section S (S-I)—Casting Details
- Section S-II—Truck Details and Casting Codes (300 and 3000 Series)
- Section S-III—Coupler and Yoke Details (100 Series)
- Section T-I—Interoperable Fuel Tenders for Locomotives, M-1004
- Section T-II—Interoperable Battery Tenders for Locomotives, M-1005
- Section T-III—Interchange Standards for Fuel Tender Components

Specifications are designated with an “M” prefix (e.g., M-900). Standards are prefixed “S” (e.g., S-900). Recommended Practices carry the prefix “RP” (e.g., RP-900).

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CHAPTER 1
ADMINISTRATIVE PROVISIONS

1.1 Purpose and Scope

1.1.1 Authority

1.1.1.1 These specifications cover minimum requirements for the design and construction of new freight cars. Tank cars are included in Chapter 6 of these specifications and where specific references are made to the AAR *Manual of Standards and Recommended Practices (MSRP)*, Section C-III, Specification M-1002, "AAR Specifications for Tank Cars." These specifications supersede all earlier design requirements and AAR specifications.

1.1.1.2 Adherence to these specifications as minimum requirements in the design and construction of freight cars is mandatory for all cars intended for interchange service. These specifications shall be considered to establish maximum permissible stresses or strains and minimum design criteria, and shall not be construed as establishing standard limitations.

1.1.1.3 While the requirements of these specifications are not retroactive, it is recommended that they be followed insofar as practicable when repairing freight cars. For applicability to modified or rebuilt cars, see the *Office Manual of the AAR Interchange Rules*, Rule 88.

1.1.1.4 Any subassembly or detail design of the superstructure or underframe of a car that forms a part of the structural strength of the car must meet all of the requirements as stipulated in these specifications, and suppliers of such subassemblies or details must so warrant.

1.1.1.5 The car builder or assembler bears the responsibility for ensuring that the car meets all applicable FRA and AAR design and construction requirements.

1.1.1.6 Where a new car design involves the use of new structural materials not foreseen in these specifications, data confirming the suitability of the materials for the intended usage shall be submitted.

1.1.1.7 When methods of fabrication and assembly of materials are unconventional, data confirming the practicability of these methods shall be submitted.

1.1.1.8 Cars with gross weight on rail in excess of 71,500 lb per axle cannot be operated in unrestricted interchange, except under special circumstances as covered in other AAR publications.

1.1.2 Related Specifications and Publications

Portions of the following publications shall form a part of these specifications where applicable:

- AAR *Manual of Standards and Recommended Practices (MSRP)*
- *Field Manual of the AAR Interchange Rules*
- *Office Manual of the AAR Interchange Rules*
- AISC (American Institute of Steel Construction) *Steel Construction Manual*
- AISI (American Iron and Steel Institute) standards
- ANSI/AWS (American Welding Society) D.15.1, "Railroad Welding Specification for Cars and Locomotives"
- ASTM (American Society for Testing and Materials) standards
- Code of Federal Regulations, Title 49

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1.2 Cars of an Untried Type

1.2.1 Definition

1.2.1.1 A car shall be considered of an untried type, whether built new or altered, when it does not fall into a category of conventional car. (For purpose of these specifications, a conventional car is defined as a car currently approved for unrestricted interchange service that has accumulated a significant record of mileage and satisfactory service.)

1.2.1.2 Application of new appurtenances or structural modification to a conventional car shall not render it new and untried, as long as the structural integrity of primary load-carrying members is retained within allowable limits listed in Chapter 4, and modifications to its suspension, draft, and/or brake system do not alter the car's performance adversely.

1.2.2 Procedure for Approval

All cars of an untried type as defined in paragraph 1.2.1 must be approved by the AAR Equipment Engineering Committee. Application should be made in accordance with *MSRP* Standard S-060.

1.2.3 Data To Be Furnished with Application for Approval

1.2.3.1 For the Equipment Engineering Committee

- General design or arrangement drawings, completely dimensioned, including those dimensions listed in paragraph 2.1.5, "Standard Dimensions," with specifications; also such detail or subassembly drawings as may be required to show the proposed construction if this is not fully disclosed in the general design or arrangement drawing and accompanying stress analysis. Materials and welding information covering primary load-carrying members must be included.
- Stress calculations covering all primary load-carrying members must be included. A stress calculation summary covering all primary load-carrying members of car derived from loads and forces listed in paragraph 4.1, "Loads and Forces," also shall be included. The assumptions and resulting stresses used in design of items not covered by Chapter 4; i.e., suspension system, special cargo tiedown, draft gear, etc., shall be included.
- Calculations submitted shall describe loading combinations applied, boundary conditions used, summary of resulting stresses, deflection if significant, and safety margins. If computerized output is used, it must include a full description of the computer program model; structural model; model drawing and assumptions used in modeling of the structure; loading combinations applied; boundary conditions used; and summary of resulting stresses.
- Estimated light weight.
- A clearance diagram shall be superimposed on the critical cross-section and end view of the car. Conformance or nonconformance to a current AAR clearance diagram must be so indicated.
- A completed rating form for a new car design (paragraph 1.3.8). Completion of this form indicates applicants' evaluation of the differences between the design submitted and that of a conventional approved car of a similar type. The Equipment Engineering Committee will consider this information in its determination of the analytical and/or test data required for the approval of the design submitted (paragraph 1.3).
- Center of gravity of car, fully loaded (paragraph 2.1.3).
- General description of truck, including spring travel and snubber type.
- Draft arrangement, including type of coupler, draft gear, draw bar, articulation, and/or other pertinent data.
- Curve negotiability calculations (paragraph 2.1.4).

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1.2.3.2 For the Brake Equipment Committee and Equipment Engineering Committee

- Brake arrangement drawing showing type and location of all items of brake equipment, including a number and size of brake cylinders, ABD or equivalent valve, reservoir, cutout cock, brake cylinder release valve, retaining valve, angle cock and hand brake.
- Gross weight on rails
- AAR Form 600 in accordance with *MSRP*, Section E-I, Standard S-473
- Braking force diagram showing the following:
  - Braking ratio
  - Total lever ratio
  - Total braking force (air)
  - Total braking force (hand brake)
  - Forces in brake rigging
- Statement indicating that the stresses in the brake rigging do not exceed the allowable stresses specified in the AAR Manual, Section E-I
- Location and group type of slack adjuster
- Type of brake shoes

1.2.4 Safety Appliances

The car builder or the design agency, if other than the car builder, shall ensure that the application and details of safety appliances comply with all applicable requirements of the *MSRP*, Section C-I, Standard S-2044, and its associated standards and specifications and with the applicable safety appliance regulations of the Federal Railroad Administration.

1.3 Car Design Approval Procedure

1.3.1 General

1.3.1.1 Upon receipt of the completed data outlined in paragraph 1.2, the Equipment Engineering Committee will evaluate the request for car design approval and advise the applicant of its determination as to the assigned car category (A, B, C, or D) for the submitted design, testing, and/or additional analytical or other data required to achieve car design approval within 90 working days of receipt of the above.

1.3.1.2 In the event the applicant decides to appeal the decision of the Committee, applicant must advise the AAR under the provisions of *MSRP* Standard S-060 within 30 days with the basis for the appeal. After review by the Equipment Engineering Committee and in the event agreement regarding test requirements is not reached, the appeal will be forwarded to the Technical Services Working Committee (TSWC) for a final decision.

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1.3.2 Categories of New Cars

Using the applicant's information (furnished on the rating form in paragraph 1.3.8) as a basis for discussion, the Equipment Engineering Committee shall determine the new car category (A, B, C, or D). The testing required for approval of the car is subject to the Equipment Engineering Committee's discretion, but the following serve as guidelines:

1.3.2.1 Category A cars are those that have major differences from conventional cars and require extensive evaluation of safety and operational aspects. This evaluation shall be carried out in three phases. Phase 1 requires all tests and/or analyses to be carried out in accordance with the requirements provided in Chapter 11. The Equipment Engineering Committee will determine the appropriate mix of tests and analyses required. Upon successful completion of Phase 1, the Equipment Engineering Committee will consider the allocation of cars for field-service testing (Phase 2). The initial field-service test quantity is to be not less than 20 cars nor more than 250 cars.

No increase in the allocation can be granted unless inspection reports are submitted on 15 cars or 10% of the cars in service, whichever is greater, that have accumulated a minimum of 25,000 miles each with at least 10% and not more than 50% of the mileage to be empty. Upon satisfactory completion of Phase 2 tests, the approval cycle can move to Phase 3 (conditional approval) in accordance with paragraph 1.3.4.

1.3.2.2 Category B cars are those with significant differences from conventional cars that raise serious questions regarding safety and/or operation in interchange service. Such cars will be required to complete most of the Phase 1 tests and/or analyses in accordance with Chapter 11 as determined by the Equipment Engineering Committee. The Phase 2 field-service tests will be carried out with a minimum of 10 cars and a maximum of 500 cars. No increase in the allocation can be granted unless inspection reports are submitted on 10 cars or 10% of the cars in service, whichever is the greater, that have accumulated a minimum of 25,000 miles each with at least 10% and not more than 50% of the mileage to be empty. Upon satisfactory completion of Phase 2 tests, the approval cycle can move to Phase 3 (conditional approval) in accordance with paragraph 1.3.4.

1.3.2.3 Category C cars are those with minor differences from conventional cars that raise some questions regarding safety and/or interchange service. Such cars will require a limited number of Phase 1 tests and/or analyses in accordance with Chapter 11, as determined by the Equipment Engineering Committee, depending upon the type of safety questions raised by the design. However, upon satisfactory results from Phase 1 testing, the design may bypass Phase 2 (field service) and move directly to Phase 3 (conditional approval).

1.3.2.4 Category D cars are those with insignificant differences from conventional cars. Minimum or no testing will be required for these cars.

Prototype groups of less than 20 cars may be authorized for developmental testing. Mileage on the final version of the prototype will be applicable to the field-service test requirements.

1.3.3 Field Service Testing

1.3.3.1 For cars subject to inspection requirements in Phase 2 field-service tests, the Equipment Engineering Committee (eec@aar.com) is to be advised of the date and location of the inspection in order that a representative of the Committee can be present. Satisfactory completion of field-service testing shall include evidence that cars exhibit satisfactory operational performance in monitored, over-the-road trains that represent a severe dynamic operational environment.

1.3.3.2 Reporting marks and numbers of the cars placed in service are to be reported to eec@aar.com on a monthly basis and distributed to railroads by circular letter with a request to report operational or mechanical problems. Service reports are to be submitted by the builder at the completion of 6 months of operation.

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1.3.4 Conditional Approval

1.3.4.1 After successful completion of initial field tests (Phase 2), the Equipment Engineering Committee may consider an increase in the initial car quantity for Category A and B cars in Phase 3 of the approval cycle.

1.3.4.2 This increase will be up to 10 times the initial field-service test quantity in service, up to a maximum of 2,000 cars. For Category C cars that are not subject to Phase 2 tests, a minimum of 5 and a maximum of 2,000 cars will be allowed into service for Phase 3 of the approval cycle. In all cases, service reports are to be submitted every 6 months.

1.3.5 Withdrawal of Test Authorization and/or Conditional Approval

1.3.5.1 In the event written validated reports indicate operational or mechanical problems, the committee reserves the right to discontinue the test. The applicant shall be allowed 30 days to provide satisfactory explanation and/or resolution of reported difficulties. In the event the committee does not consider the car satisfactory and withdraws test authorization, the applicant may appeal the decision to the TSWC, Mechanical Division. Such appeal may be accompanied by any comments, explanations, or evidence in support of such appeal that the proponent desires to submit.

1.3.5.2 In the event that the Equipment Engineering Committee considers that the conditional approval should be withdrawn, the recommendation must be confirmed by the TSWC. The applicant may submit for consideration any comments, explanations, or evidence in support of applicant's position.

1.3.6 Approval

The basis of granting approval of the design shall be

- completion of 1 year's service by a minimum of 200 conditionally approved cars with an average accumulated service mileage of 50,000 miles; and
- applicant's submission of a detailed inspection report on a minimum of 20 of these cars, with no validated reports of operational or mechanical problems in need of resolution by the applicant, car owner, or agent for the AAR.

The Equipment Engineering Committee is to be advised by email at eec@aar.com of the date and location of the inspection so that a representative of the Committee can be present.

1.3.7 Summary of Approval Requirements for New and Untried Cars

New Car Category[a/]	Approval Phase 1[b/] Track Worthiness Tests/Analyses	Approval Phase 2 Field Service Test Requirements			Approval Phase 3 Conditional Approval Requirements		
	Types of Test or Analyses Required	Min (car)	Max (cars)	Period (miles per car)	Min (car)	Max (cars)	Period (miles per car)
A	All	20	250	25,000	20	2000[c/]	50,000
B	Most	10	500	25,000	10	2000[c/]	50,000
C	Some		[d/]		5	2000	50,000
D	[d/]		[d/]			[d/]	

[a/] New Car Category based on evaluation by the Equipment Engineering Committee.
[b/] The types of tests and/or analyses required will depend upon the degree of variation from conventional car design, as determined by the Equipment Engineering Committee.
[c/] Maximum number of cars is 2000 or 10 times the number submitted for field service tests, whichever is the least.
[d/] At the discretion of the Equipment Engineering Committee.

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1.3.8 Rating Form—New Freight Car Designs

Car Description: Builder:

Identification/Car No.: Submitted by:

AAR Mechanical Designation Date:

Section I

Instructions: Please indicate your evaluation (0, 1, or 2) as to the degree of variation from conventional freight car characteristics, where

0 = no significant differences
1 = moderate deviations
2 = substantial deviations

Vehicle Dynamics Influences

Truck Design & Suspension		Car Body Properties			Subtotal		Weighing Factor		Total
_____	+	_____		=	_____	×	3	=	_____

Coupler & Draft Gear		Loading			Subtotal		Weighing Factor		Total
_____	+	_____		=	_____	×	2	=	_____

Other Influences

Construction		Commodities		Operations		Subtotal		Weighing Factor		Total
_____	+	_____	+	_____	=	_____	×	1	=	_____

Total Rating Value = _____

New Car Category[a/] = _____

[a/] Rating Value of 12–23 = Category A
Rating Value of 8–11 = Category B
Rating Value of 4–7 = Category C
Rating Value of 0–3 = Category D

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Section II

Instructions: Describe how the design differs from conventional cars of a similar type, and describe the basis for the rating suggested in Section I.

Truck Design and Suspension—Is the design AAR approved? Type of material—cast or fabricated? Type of suspension—coil, leaf, elastomeric or air springs? Type of snubbing device—friction, hydraulic etc? Adverse effects of wheels, axles, bearings and brake rigging?
Car Body Properties—Light weight and CG height; any unusual dynamic characteristics, torsional rigidity, etc.?
Coupler and Draft Gear—Are AAR-approved standard or special cushioning devices used? If not, explain.
Loading—What is the gross rail load; potential for irregular load distribution; CG height with maximum uniform or concentrated live load, depending on car type, applied at most critical locations?

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Section II (continued)

Construction—Are methods of assembly, materials, and/or fastenings different from accepted practice? If so, explain.
Commodities—Are unusual ladings carried that require special handling, securement devices, loading or unloading facilities? If so, explain.
Operations—Are special train handling or operational restraints required? What is the estimated annual mileage per car?
Other Remarks:

CHAPTER 2
GENERAL DATA

2.1 Design Data

2.1.1 Scope

The basic design data in this section apply to all cars intended for interchange service.

2.1.2 Dimensions and Weights

2.1.2.1 Limiting Outline

Where cars are intended for unrestricted interchange service, they must comply with the requirements of Plates B and B-1 shown in AAR *Manual of Standards and Recommended Practices (MSRP)* Section C-I, Standard S-2056. Cars having outside dimensions that exceed those of Plate B to a limited extent may be built for limited interchange service provided none of these outside dimensions go beyond those as outlined in the appropriate plates.

2.1.2.2 Weight and Axle Spacing Criteria

Table 2.1

Type Truck	Maximum Track Load (lb) Per Car	Per Axle	Wheel Size (in.)
4 wheel	286,000	71,500	36
6 wheel	315,000	52,500	30
6 wheel	360,000	60,000	33
6 wheel	394,500	65,750	36

2.1.2.3 Cooper Bridge Rating Criteria

The Cooper Bridge Rating is the established technique for evaluating and qualifying the load input into bridge structures for a specific car design.

All new car designs intended for unrestricted interchange shall be compared to the Cooper E Rating criteria established for bridges utilizing Method 1 or meeting dimensional requirements of Method 2 below.

While interchange between railroads is not restricted for cars meeting these limits, individual railroads may have conditions for some routings that are more restrictive than these requirements. Acceptance by a railroad of a railcar for certain routings may be subject to additional requirements. Any questions regarding the ability of a specific railcar configuration to be used over a given routing should be directed to the railroad company(ies) responsible for the infrastructure on any portion of the proposed service.

2.1.2.3.1 Method 1 - Calculated Limits for Cooper Bridge Rating Criteria

Cooper Bridge Ratings for the car are calculated using AAR Bridge software for span lengths in the range of 0 ft to 400 ft. The ratings are calculated for bending moment, shear, and floor beam/pier reaction. The loading assumes a unit-train of the proposed railcars at maximum axle weights. The maximum for all three ratings is compared to the Cooper E Rating Limits for span lengths of 0 ft to 50 ft. For span lengths between 50 ft and 400 ft, the maximum is determined from bending moment and shear ratings only. The proposed car shall not produce a Cooper rating in excess of the values shown in Table 2.2.

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Table 2.2

Span Length (ft)	Cooper E Rating Guideline
0 to 9	E71.5
9 to 11	E71.5 to E65[a/]
11 to 25	E65
25 to 45	E65 to E60[a/]
45 to 225	E60
225 to 300	E60 to E63.5[a/]
300 to 400	E63.5 to E65.5[a/]

[a/] For those span length ranges that display a range of values for the Cooper E Rating Limit, the limit is linearly interpolated from the span length.

2.1.2.3.2 Method 2 - Dimensional Limits for Cooper Bridge Rating Criteria

Cars having the characteristics listed in Table 2.3 shall comply with the dimensional limits listed in Table 2.4.

Table 2.3

Gross Rail Load (lb)	Truck Wheelbase
220,000	5 ft 8 in.
263,000	5 ft 10 in.
286,000	5 ft 10 in.

Table 2.4

Gross Rail Load (lb)	Bolster Centerline to Striker Face (Overhang)	Truck Centers (Minimum)	Length Over Pulling Faces (Minimum)
286,000 and 263,000	4 ft 11 ½ in.	29 ft 5 in.	41 ft 11 ½ in.
	5 ft 5 ½ in.	28 ft 4 ½ in.	41 ft 11 in.
	5 ft 11 ½ in.	27 ft 4 in.	41 ft 10 ½ in.
220,000	4 ft 11 ½ in.	22 ft 6 in.	35 ft 0 ½ in.
	5 ft 5 ½ in.	21 ft 6 in.	35 ft 0 ½ in.
	5 ft 11 ½ in.	20 ft 6 in.	35 ft 0 ½ in.

2.1.2.3.3 Restricted Interchange - Cooper Rating

If neither Method 1 nor Method 2 are met, the car's interchange is restricted and calculated Cooper E Ratings must be provided to the host railroad for acceptance.

2.1.3 Vertical Center of Gravity

Height of center of gravity of a fully loaded car (including weight of trucks and spring deflection) shall not exceed 98 in. above top of rail.

Notes for calculating the center of gravity of a loaded car:

1. Box cars are to be loaded to the eaves and to their load limit capacity. If the center of gravity exceeds 98 in. when fully loaded to the top of the eaves, then the empty car center of gravity and the floor height(s) must be marked in accordance with *MSRP* Section C-I, Standard S-910.
2. Covered hopper cars are to be loaded to the eaves and to their load limit capacity. Cars without clearly definable eaves are to be loaded to 97% of the total shell volume and to their load limit capacity.
3. Open top hopper cars and gondola cars are to be loaded with 10-in. average heap and to their load limit capacity. (The heap shall be calculated as a uniform rectangular load 10 in. above the sides of the car.)
4. Center of gravity of flatcars is to be determined for empty cars only.
5. The cubic capacity of covered hopper cars shall be calculated as that volume below the lowest portion at the hatch frame.
6. Bulkhead flatcars, including center partition cars, having centers of gravity exceeding 98 in. when fully loaded evenly to the top of the bulkhead be marked in accordance with *MSRP* Section C-I, Standard S-910.
7. Center partition (center beam) cars are to be loaded to the underside of the overhanging flanges of the top cross beam to load limit capacity. For cars designed without an overhanging cross-beam flange, design the car with a loaded center of gravity based on it being loaded to the load limit when fully loaded evenly to the top of the bulkhead.

2.1.4 Horizontal and Vertical Curves

2.1.4.1 General

2.1.4.1.1 Cars shall be built to negotiate the minimum curves possible with the use of standard or alternate standard couplers, yokes, and strikers, but in no event shall cars exceed minimum radius curves specified in paragraphs 2.1.4.2 and 2.1.4.3. When cars are required to operate over curves of lesser radii than provided for in paragraphs 2.1.4.2 and 2.1.4.3, it shall be the responsibility of the customer to notify the car builder of the requirements.

2.1.4.1.2 Calculations are to be made considering standard draft and sliding sill cars with couplers and underframes in normal position and end-of-car cushioning devices in either buff, normal, or draft positions, whichever are the most critical.

2.1.4.2 Horizontal Curve and Tangent

Cars shall be designed to operate over the respective horizontal curves without interference between trucks, car body, and brake rigging. In addition, cars shall be designed to negotiate a No. 7 crossover having 13-ft track centers when coupled to a like car or AAR base car. All combinations for buff and draft plus the bolster offset must be investigated. In all cases, the calculated knuckle contour angling must not exceed the values shown in Tables 2.5 and 2.6 of paragraph 2.1.4.4.

2.1.4.2.1 Minimum Curve Coupled to Base Car and Like Car

Length Over Pulling Faces of Couplers	Minimum Radius Base and Like Car
Less than 50 ft	185 ft

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Length Over Pulling Faces of Couplers	Minimum Radius Base and Like Car
Over 50 ft to 56 ft	215 ft
Over 56 ft to 63 ft	250 ft
Over 63 ft to 70 ft	275 ft
Over 70 ft to 75 ft	300 ft
Over 75 ft	350 ft

On multi-unit cars, the length over pulling face of couplers is defined as follows:

End Unit	Intermediate, Drawbar-Connected Units	Articulated Cars
Twice the distance from the pulling face to the midpoint between trucks	Twice the distance from the centerline of the draw bar to the midpoint between trucks	Truck centers

2.1.4.2.2 Minimum Curve for Cars Uncoupled

- For cars having truck centers of 46 ft 3 in. or less–150-ft radius.
- For cars having truck centers greater than 46 ft 3 in.–180-ft radius.
- For multi-unit cars, the truck centers of the longest individual unit will be used.

2.1.4.2.3 Base Car

The base car is equipped with SBE60C design couplers and Y40A design yokes, applied to the car as in *MSRP* Section C-I, Standard S-239, and has the following dimensions:

Length over pulling faces of couplers	44 ft 7 7/8 in.
Length over strikers	42 ft 0 3/8 in.
Truck centers (T.C.)	31 ft 1 3/8 in.
Overhang (centerline of bolster to striker face)	5 ft 5 1/2 in.
Horn clearance	0 ft 3 3/4 in.

2.1.4.2.4 Curve Negotiation—Standard Method of Calculating Minimum Radius

This method may be used to determine the minimum horizontal curve and tangent any two coupled cars can negotiate. Tables are included giving dimensions pertinent to the standard coupler applications when draft gears are in normal position. The formulae presented are simplified empirical equations derived from studies of exact methods and actual car service data.

2.1.4.2.5 Placement of Cars

The most critical design condition is the curve and tangent with no easement. In this condition, one car is on tangent and the other on curve. If different length cars are involved, the critical condition usually occurs with the shorter car on the tangent, but the long car on the tangent should be analyzed.

Two like long cars coupled, each equipped with Type F couplers, may be more restrictive than one long car equipped with Type F couplers while coupled to a base car.

The cars are placed with the coupling line at the point of tangency when Equation (1) is used. Generally, this is the critical positioning when couplers with large amounts of contour angling are used, such as E and E/F couplers.

The cars are placed with the coupler pivot point of the car on the tangent at the point of tangency when Equation (2) is used. Generally, this is the critical positioning when couplers with small amounts of contour angling are used, such as F couplers.

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In all cases, like cars and the car being studied coupled to the base car must be checked.

Equation (1)

Start with Equation (1). If Equation (1) results in a calculated horizontal contour angle exceeding the limits of Table 2.6, Equation (2) must be used. The horizontal contour angle (a) is calculated using Equation (1.4).

$$R = \frac{(B_2)^2 - (D_2)^2 - E^2}{2E} \qquad \text{Equation 1}$$

$$E = \frac{Y_1 + C_1 + Y_2 + C_2}{12} \qquad \text{Equation 1.1}$$

$$Y = 0.625\frac{B}{D} + 0.625 \text{ (use appropriate subscripts)} \qquad \text{Equation 1.2}$$

$$\beta = \text{asin}\left(\frac{B_2}{R+E}\right) \qquad \text{Equation 1.3}$$

$$\alpha = \theta_1 + \beta - \theta_2 \qquad \text{Equation 1.4}$$

where in Fig. 2.1,

R	=	Minimum radius of curve (ft)
B	=	½ of distance over coupling lines (ft)
D	=	½ of distance over truck centers (ft)
E	=	Total lateral displacement between car center lines at tangent point (ft)
C	=	Maximum allowable lateral coupler displacement at coupling line relative to car centerline (inches). See Table 2.5.
Y	=	Car centerline lateral displacement at coupling line (in.)
α	=	Horizontal contour angle between coupler centerlines (degree). See Table 2.6 for maximums.
θ	=	Maximum allowable lateral coupler angle relative to car centerline (degree). See Table 2.5.
β	=	Horizontal angle between car centerlines (degree)

NOTES:
1. SUBSCRIPT (1) APPLIES TO CAR ON TANGENT AND SUBSCRIPT (2) APPLIES TO CAR ON CURVE. REFER TO FIG. 2.1.
2. THE CONSTANT 0.625 IS EMPIRICAL AND REPRESENTS TRUCK LATERAL AND OTHER CURVE FACTORS, SUCH AS SPREAD OF RAILS AND EFFECT OF SUPERELEVATION.
3. IN THE FORMULA FOR R, THE TERM $-E^2$ IS MATHEMATICALLY INSIGNIFICANT AND CAN BE DISREGARDED.

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Equation (2)

If Equation (1) results in calculated horizontal contour angle exceeding the limits of Table 2.6, Equation (2) must be used.

$$R = \frac{(B_2 + L_1)^2 - (D_2)^2 - E^2}{2E} \qquad \text{Equation 2}$$

$$E = L_2 \sin\theta_2 + L_1 \sin(\theta_2 + \alpha) + X_1 + X_2 \qquad \text{Equation 2.1}$$

$$X_1 = Y_1\left(\frac{D_1 + B_1 - L_1}{D_1 + B_1}\right) \qquad \text{Equation 2.2}$$

$$X_2 = Y_2\left(\frac{D_2 + B_2 + L_1}{D_2 + B_2}\right) \qquad \text{Equation 2.3}$$

$$Y = \frac{1}{12}\left(0.625\frac{B}{D} + 0.625\right) \text{(Use appropriate subscripts)} \qquad \text{Equation 2.4}$$

$$\beta = \text{asin}\left(\frac{B_2 + L_1}{R + E}\right) \qquad \text{Equation 2.5}$$

$$\theta_1 = \theta_2 + \alpha - \beta \qquad \text{Equation 2.6}$$

where in Fig. 2.2,

R	=	Minimum radius of curve (ft)
L	=	Effective coupler length (ft). See Table 2.5.
B	=	½ of distance over coupling lines (ft)
D	=	½ of distance over truck centers (ft)
E	=	Total lateral displacement between car centerlines at tangent point (ft)
X	=	Car centerline lateral displacement at tangent point (ft)
Y	=	Car centerline lateral displacement at coupling line (ft)
θ	=	Maximum allowable lateral coupler angle relative to car centerline (degree). See Table 2.5.
α	=	Maximum horizontal contour angle between coupler centerlines (degree). See Table 2.6.
β	=	Horizontal angle between car centerlines (degree)

NOTES:
1. SUBSCRIPT (1) APPLIES TO CAR ON TANGENT AND SUBSCRIPT (2) APPLIES TO CAR ON CURVE. REFER TO FIG. 2.2.
2. THE CONSTANT 0.625 IS EMPIRICAL AND REPRESENTS TRUCK LATERAL AND OTHER CURVE FACTORS, SUCH AS SPREAD OF RAILS AND EFFECT OF SUPERELEVATION.
3. IN THE FORMULA FOR R, THE TERM $-E^2$ IS MATHEMATICALLY INSIGNIFICANT AND CAN BE DISREGARDED.

The lateral coupler angle relative to the car centerline in the car on tangent must not exceed the values shown in Table 2.5 and is calculated using Equation (2.6).

2.1.4.3 Vertical Curves

Cars shall be designed to operate over the respective vertical curves without interference between (a) car body, including attached parts, and trucks; (b) car body, including attached parts, and track structure or retarders; (c) trucks and track structure or retarders. The design shall include allowance for truck springs deflected to 75% of their total travel, and where applicable, maximum wear of 2 in., plus body deflection, except $\frac{3}{4}$ in. wear allowance is to be used between axle and center sill attachments.

The 2 $\frac{3}{4}$-in. clearance above top of rails specified in the equipment diagrams in *MSRP* Section C-I, Standard S-2056 does not apply in calculating clearance for vertical curves.

2.1.4.3.1 Minimum Curve (Concave and Convex) Coupled to Base Car and Like Car

Truck Centers	Minimum Curve
Up to 41 ft 3 in. *and* overhangs of 5 ft 6 in. or less	785 ft
All others less than 63 ft	1,650 ft
63 ft or greater	2,000 ft

2.1.4.3.2 Minimum Curve for Cars Uncoupled

Truck Centers	Minimum Radius
Up to 41 ft 3 in.	500 ft
41 ft 3 in. to 54 ft 0 in.	850 ft
Over 54 ft 0 in.	1,250 ft

For multi-unit cars, the truck centers of the longest individual unit will be used. For single unit cars to be unloaded by rotary car dumpers, the minimum radius is 300 ft.

2.1.4.3.3 Base Car

The same base car as for horizontal curves is to be used, except equipped with F70C design couplers and Y45A design yokes applied to the car as in *MSRP* Section C-I, Standard S-245. The dimensions then become as follows:

Length over pulling faces of couplers	44 ft 8 $\frac{3}{8}$ in.
Length over strikers	42 ft 0 $\frac{7}{8}$ in.
Truck centers (T.C.)	31 ft 1 $\frac{3}{8}$ in.
Overhang (T.C. to striker face)	5 ft 5 $\frac{3}{4}$ in.
Horn clearance	0 ft 3 $\frac{3}{4}$ in.

If it is required to determine the minimum vertical curve radius for cars equipped with Type E couplers, the base car will be as given in paragraph 2.1.4.2.3.

2.1.4.3.4 Curve Negotiation—Standard Method of Calculating Minimum Radius

2.1.4.3.4.1 This method may be used to determine the minimum vertical curve that any two coupled cars can negotiate. Tables are included giving dimensions pertinent to the standard coupler applications when draft gears are in normal position. The formulae presented are simplified empirical equations derived from studies of exact methods and actual car service data.

2.1.4.3.4.2 Cars are to be considered vertically offset 3 in. maximum difference between light and loaded cars.

2.1.4.3.4.3 In all cases, like cars and the car being studied coupled to the base car must be checked.

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2.1.4.3.5 Vertical Curve and Tangent—Placement of Cars

2.1.4.3.5.1 Usually when both cars have F Couplers, the curve and tangent with no easement is the most critical condition. Here one car is on the curve and the other on the tangent.

2.1.4.3.5.2 When Equation (3) is used, the cars are placed with the coupling line at the point of tangency. When Equation (4) is used, cars are placed with the coupler pivot point of the car on the tangent at the point of tangency.

Two Identical Cars	Two Different Cars
On a convex curve, the critical condition is when the car on curve is 3 in. higher than the car on tangent. In the concave condition, the reverse condition is critical. If allowed vertical angling is the same upward and downward, then only one calculation is required, as the convex and concave condition will be identical. Where there is a difference in upward and downward vertical angling, it is necessary to calculate all four conditions (i.e., concave and convex with ± 3 in. offset), although the critical conditions stated above hold true in most cases.	The most critical condition generally arises with the short car on tangent and the long car on curve. On a convex curve, the critical condition is when the car on curve is 3 in. higher than the car on tangent. In the concave condition, the reverse condition is critical. If Type F Coupler application is used, the condition where the long car on a convex curve is 3 in. higher generally is the most critical. Where there is a difference in upward and downward vertical angling, it is necessary to calculate all four conditions (i.e., concave and convex with ±3 in. offset), although the critical conditions stated above hold true in most cases.

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Equation (3)

Start with Equation (3). If Equation (3) results in a calculated vertical contour angle exceeding the limits of Table 2.6, Equation (4) must be used. The vertical contour angle (a) is calculated using Equation (3.3).

$$R = \frac{(B_2)^2 - (D_2)^2 - E^2}{2E}$$ Equation 3

$$E = L_2 \sin\theta_2 + L_1 \sin\theta_1 + Z - Y$$ Equation 3.1

$$\beta = \text{asin}\left(\frac{B_2}{R+E}\right)$$ Equation 3.2

$$\alpha = \theta_1 + \beta - \theta_2$$ Equation 3.3

where in Figs. 2.3 and 2.4,

R	=	Minimum radius of curve (ft)
B	=	½ of distance over coupling lines (ft)
D	=	½ of distance over truck centers (ft)
L	=	Effective coupler length (ft). See Table 2.7.
Z	=	Vertical displacement of coupler centerlines at coupling line (ft). See Table 2.8.
Y	=	Vertical displacement of coupler centerlines at coupling line (ft) due to difference between light and loaded cars.
E	=	Vertical displacement between car centerlines due to curving (ft)
α	=	Maximum vertical contour angle between coupler centerlines (degree). See Table 2.6.
θ	=	Maximum coupler vertical angle relative to car centerline (degree). See Table 2.7.
β	=	Vertical angle between car centerlines (degree).

NOTES:

1. SUBSCRIPT (1) APPLIES TO CAR ON TANGENT, AND SUBSCRIPT (2) APPLIES TO CAR ON CURVE. REFER TO FIG. 2.3 OR FIG. 2.4.
2. IN THE FORMULA FOR R, THE TERM $-E^2$ IS MATHEMATICALLY INSIGNIFICANT AND CAN BE DISREGARDED.
3. Y MAX = 0.250 FT (3 IN.) FOR CURVE NEGOTIABILITY CALCULATIONS. Y = 0 FOR NO VERTICAL OFFSET BETWEEN CARS.
4. A CALCULATED NEGATIVE VALUE FOR α IS ACCEPTABLE PROVIDED THE MAGNITUDE OF α DOES NOT EXCEED THE MAXIMUM ALLOWABLE IN TABLE 2.6. A NEGATIVE VALUE FOR α MEANS THE α IS ON THE OPPOSITE SIDE OF THE REFERENCE LINE SHOWN IN THE FIGURESS, AND THE FORMULA FOR α THEN BECOMES $\alpha = \theta_1 + \theta_2 - \beta$.

The lateral coupler angle relative to the car centerline in the car on tangent must not exceed the values shown in Table 2.5 and is calculated using Equation (2.6).

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Equation (4)

If Equation (3) results in a calculated vertical contour angle exceeding the limits of Table 2.6, Equation (4) must be used.

$$R = \frac{(B_2 + L_1)^2 - (D_2)^2 - E^2}{2E} \qquad \text{Equation 4}$$

$$E = L_2 \sin\theta_2 + L_1 \sin(\theta_2 + \alpha) + Z - Y \qquad \text{Equation 4.1}$$

$$\beta = \text{asin}\left(\frac{B_2 + L_1}{R + E}\right) \qquad \text{Equation 4.2}$$

$$\theta_1 = \theta_2 + \alpha - \beta \qquad \text{Equation 4.3}$$

NOTES:
1. THE DEFINITIONS OF THE VARIABLES ARE THE SAME AS FOR EQUATION (3).
2. SUBSCRIPT (1) APPLIES TO CAR ON TANGENT, AND SUBSCRIPT (2) APPLIES TO CAR ON CURVE. REFER TO FIG. 2.5 OR FIG. 2.6.
3. IN THE FORMULA FOR R, THE TERM $-E^2$ IS MATHEMATICALLY INSIGNIFICANT AND CAN BE DISREGARDED.
4. FOR E AND E/F COUPLERS IN FIG. 2.6, $\theta = 0$ BECAUSE THESE COUPLERS CANNOT ANGLE DOWN. FOR THIS CONDITION, EQUATION (4.3) BECOMES $\beta = \theta_2 + \alpha$. USING THIS β IN EQUATION (4.2), CALCULATE R USING E AS CALCULATED IN EQUATION (4.1).

The vertical coupler angle relative to the car centerline for the car on tangent (θ_1) must not exceed the value in Table 2.7 and is calculated using Equation (4.3).

2.1.4.4 Tables and Diagrams for Computing Curve Negotiability

2.1.4.4.1 Horizontal Curve Data for Standard Coupler Arrangements

The data in this chart are for cars with standard draft gears. For cars having EOCC units, the engineer must refer to the application drawing for the intended coupler and cushioning unit. The maximum coupler lateral angle from this application drawing, in both the full draft and full buff positions, must be used in curving calculations. The maximum coupler lateral displacement at the coupling line must be calculated, for the full draft and full buff conditions, based on this maximum coupler lateral angle.

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Table 2.5 Horizontal curve data for standard coupler arrangements

Coupler Design Yoke Design Arrangement Standard[a/]	Maximum Coupler Lateral Angle[b/]	Maximum Coupler Lateral Displacement at Coupling Line C (in.)[b/]	Effective Coupler Length L[c/] (in.)	Length of Shank[d/] (in.)
E60C, Y40A, S-239	7°	3.47	28.46	21.50
E61B, Y30A, Striker S-2018	9°	4.53	28.94	16.94
E67B, Y41A, S-241	8°	4.63	33.28	25.00
E68B, Y45A, S-243	13°	9.67	43.00	31.00
E68B, Y49A, 13° Striker	13°	9.67	43.00	31.00
E69A, Y45A, S-244	15°	15.53	60.00	48.00
F70C, Y45A, Striker S1C (former)	10°	5.08	29.25	17.25
F70C, Y45A, S-245	13°	6.58	29.25	17.25
F79C, Y45A, S-247	13°	9.67	43.00	31.00
F79C, Y49A, 13° Striker	13°	9.67	43.00	31.00
F73A, Y45A, 15° Striker	15°	15.53	60.00	48.00

NOTES:

[a/] Where no arrangement standard exists, strikers are shown.
[b/] Lateral tabular values shown are maximum coupler displacements. Lateral values may be reduced, providing cars can negotiate the required curves specified in paragraph 2.1.4.2.1.
[c/] Length from coupling line to intersection of coupler centerline with car centerline.
[d/] Length from coupler horn to butt or pivot point of coupler.

2.1.4.4.2 AAR Standard Couplers Maximum Contour Angle Between Coupler Centerlines

Table 2.6 AAR standard couplers
Maximum contour angle between coupler centerlines

Coupler Combination	Contour Angle α	
	Horizontal	Vertical
E and E	13.50°	4.75°
E and F	8.00°	3.25°
E and H	6.50°	2.00°
F and F	3.75°	2.00°
F and H	1.75°	0.75°
H and H	0°	0°

2.1.4.4.3 Vertical Curve Data for Standard Coupler Arrangements

The data in this table are for cars with standard draft gears, with the coupler horn pushed against the striker. For cars with EOCC units, the engineer must calculate θ and the maximum coupler vertical displacement with the coupler in the full draft and full buff positions. These values must be used in curving calculations.

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Table 2.7 Vertical curve data for standard coupler arrangements[a/]

Coupler Design Yoke Design Clearance Over Coupler Shank	Effective Coupler Length[b/] (in.)	Maximum Coupler Vertical Angle θ° [c/,d/]		Maximum Coupler Vertical Displacement at Coupling Line (in.)[c/]	
		Upward	Downward	Upward	Downward
E60C, Y40A, 1/2 in.	28.46	1.77°	0°	0.88	0
E61B, Y30A, 1/2 in.	28.94	1.74°	0°	0.88	0
E67B, Y41A, 1/2 in.	33.28	1.52°	0°	0.88	0
E68B, Y45A, 5/8 in.	43.00	1.17°	0°	0.88	0
E68B, Y49A, 5/8 in.	43.00	1.17°	0°	0.88	0
E69A, Y45A, 11/16 in.	60.00	0.84°	0°	0.88	0
F70C, Y45A, 1 5/16 in.	29.25	5.50°	7.00°	2.81	3.56
F79C, Y45A, 2 9/16 in.	43.00	5.50°	7.50°	4.00	5.50
F79C, Y49A, 2 9/16 in.	43.00	5.50°	7.50°	4.00	5.50
F73A, Y45A, 4 1/4 in.	60.00	5.50°	7.50°	5.75	7.83

NOTES:

a/ Data applies to couplers with or without top and/or bottom shelves.
b/ Length from coupling line to intersection of coupler centerline with car centerline.
c/ Actual vertical angling capability may be less depending on car design.
d/ E and E/F couplers with and without shelves are assumed to angle up in all cases where indicated by figures. E and E/F couplers cannot angle down due to rigid coupler carriers.

Table 2.8 Allowable vertical displacements at coupling line between coupled couplers[a/, b/]

Coupler Combination		Maximum Vertical Displacement Z[c/]	
Coupler 1	Coupler 2	Coupler 1 Higher than Coupler 2	Coupler 2 Higher than Coupler 1
E, SE, or SBE	E, SE, or SBE	0.531 ft (6 3/8 in.)	0.531 ft (6 3/8 in.)
E, SE, or SBE	F	0.531 ft (6 3/8 in.)	0.292 ft (3 1/2 in.)
F	E, SE, or SBE	0.292 ft (3 1/2 in.)	0.531 ft (6 3/8 in.)
E, SE, or SBE	SF	0.375 ft (4 1/2 in.)	0.292 ft (3 1/2 in.)
SF	E, SE, or SBE	0.292 ft (3 1/2 in.)	0.375 ft (4 1/2 in.)
F or SF	F or SF	0	0

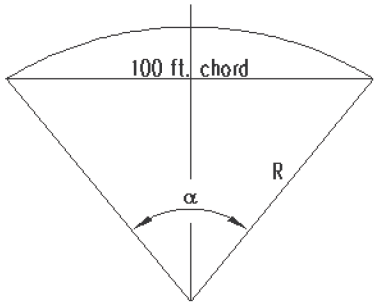
NOTES:

a/ Data applies to couplers with or without top and/or bottom shelves.
b/ For combinations of E, SE, or SBE couplers, the Z displacement 6 3/8 in. in Table 2.8, plus maximum vertical displacement 0.88 in. in Table 2.7 equals 7 1/4 in., which is maximum displacement allowed for calculations. For shelf couplers, values are for contact between knuckle and shelf.
c/ Actual vertical angling capability may be less depending on car design.

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Table 2.9 Degree and radius of curve (based on 100-ft chord)

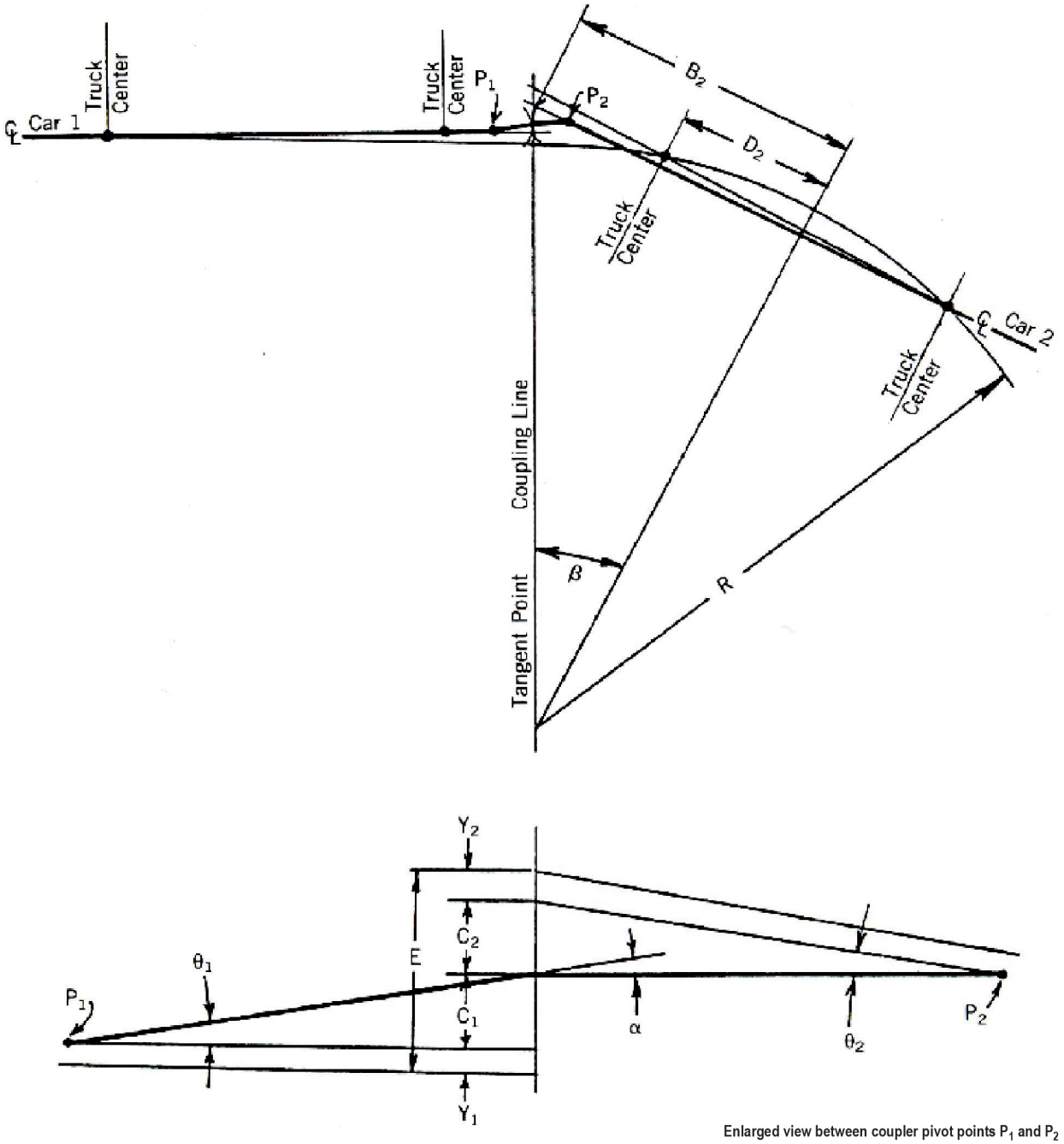
Degree	Radius	Degree	Radius	Degree	Radius	Degree	Radius
10	574	16	359	22	262	28	207
11	522	17	338	23	251	29	200
12	478	18	320	24	240	30	193
13	442	19	303	25	231	31	187
14	410	20	288	26	222	32	181
15	383	21	274	27	214	33	176



$$R = \frac{50}{\sin\frac{1}{2}\alpha}$$

$$\text{Approximate } R = \frac{5730}{\alpha}$$

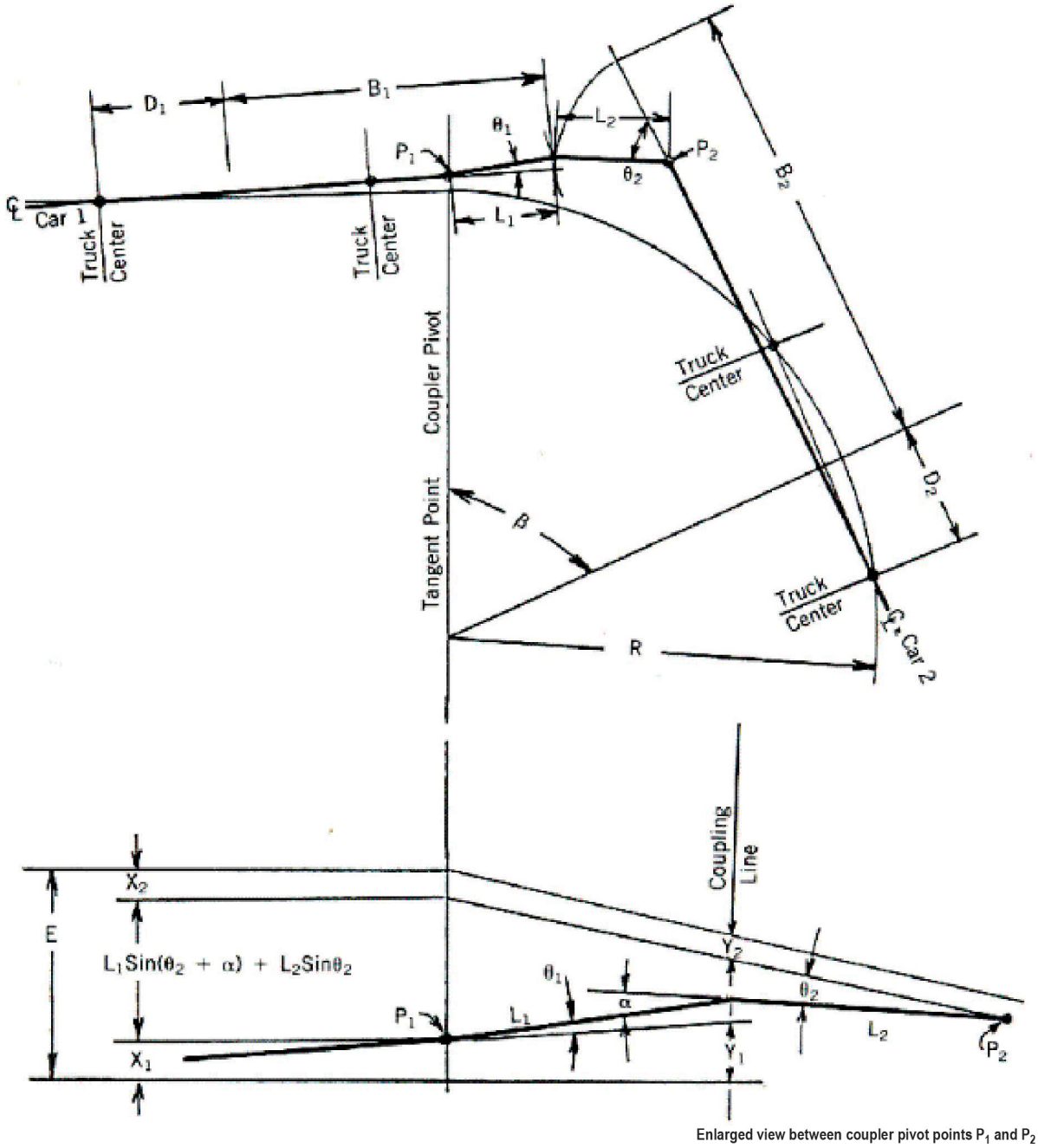
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Minimum radius of curvature as determined by maximum horizontal displacement of couplers. Coupling line is at point of tangency. For definition of terms see Equation (1).

Fig. 2.1 Horizontal curve and tangent—Equation (1)

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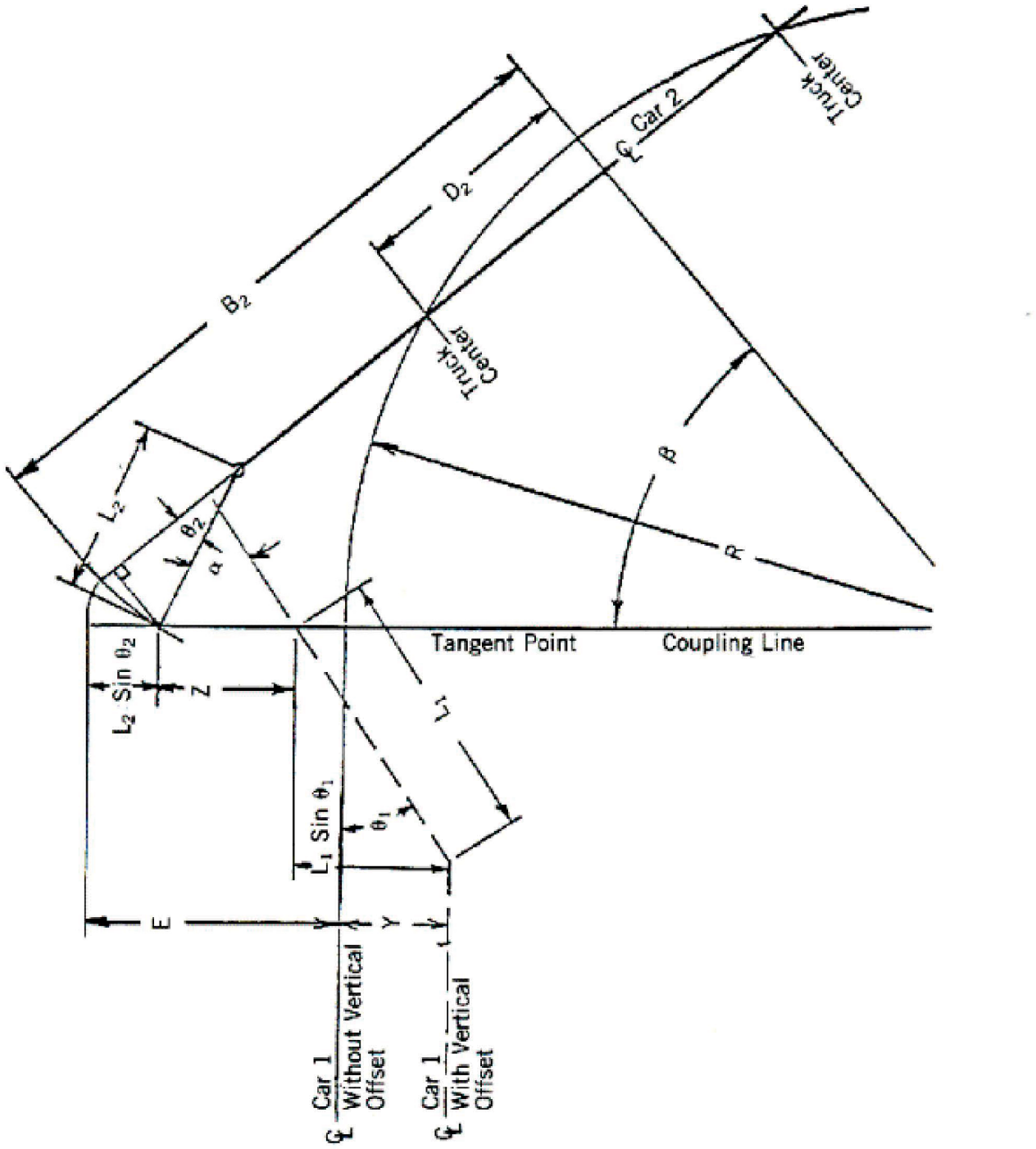


Enlarged view between coupler pivot points P_1 and P_2

Minimum radius of curvature as determined by maximum horizontal contour angle. Coupler pivot is at point of tangency. For definition of terms see Equation (2).

Fig. 2.2 Horizontal curve and tangent—Equation (2)

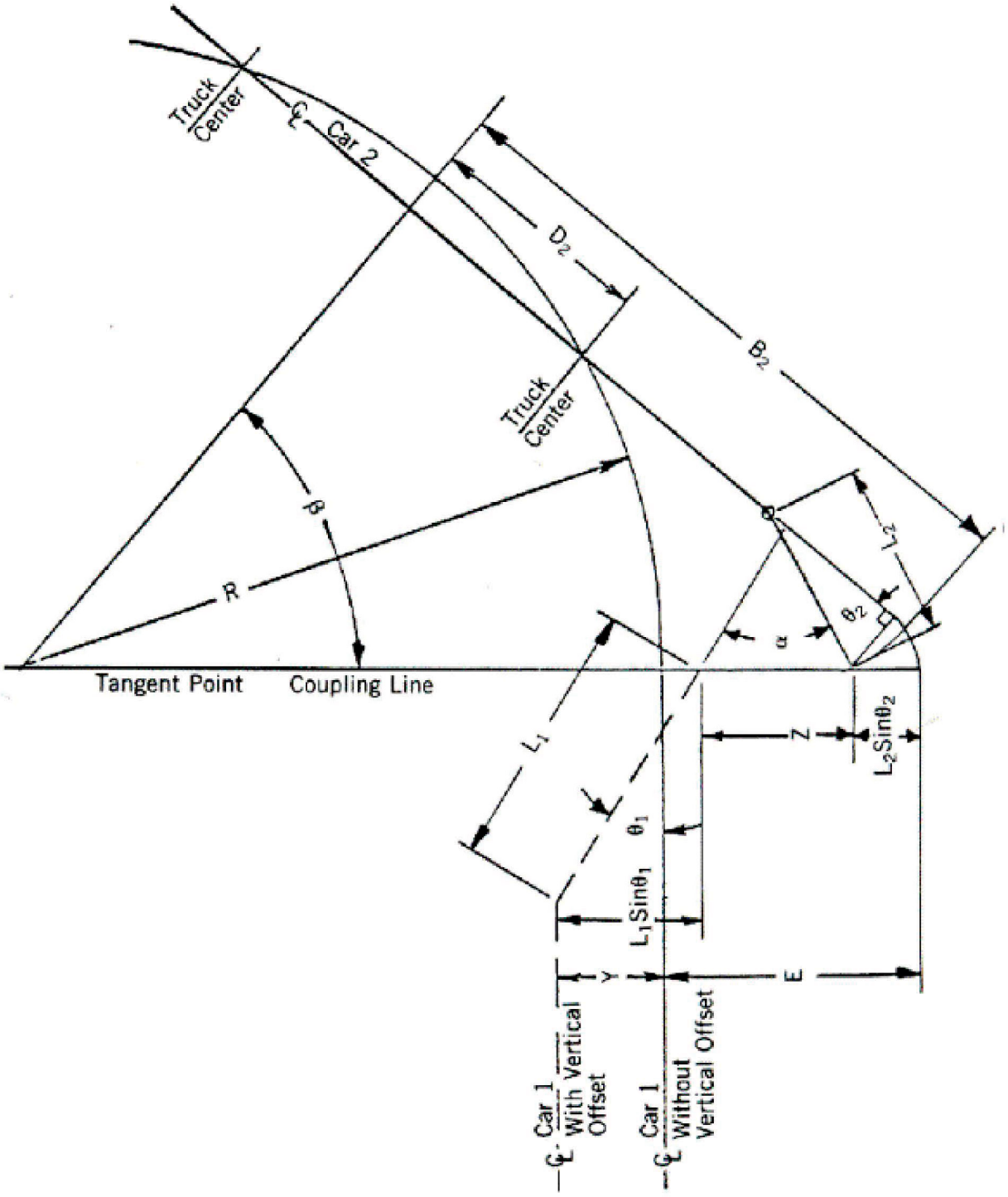
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Minimum radius of curvature as determined by maximum vertical displacement of couplers. Coupling line is at point of tangency. For definition of terms see Equation (3). Note that coupler in car on curve angles down while coupler in car on tangent angles up.

Fig. 2.3 Vertical convex curve and tangent—Equation (3)

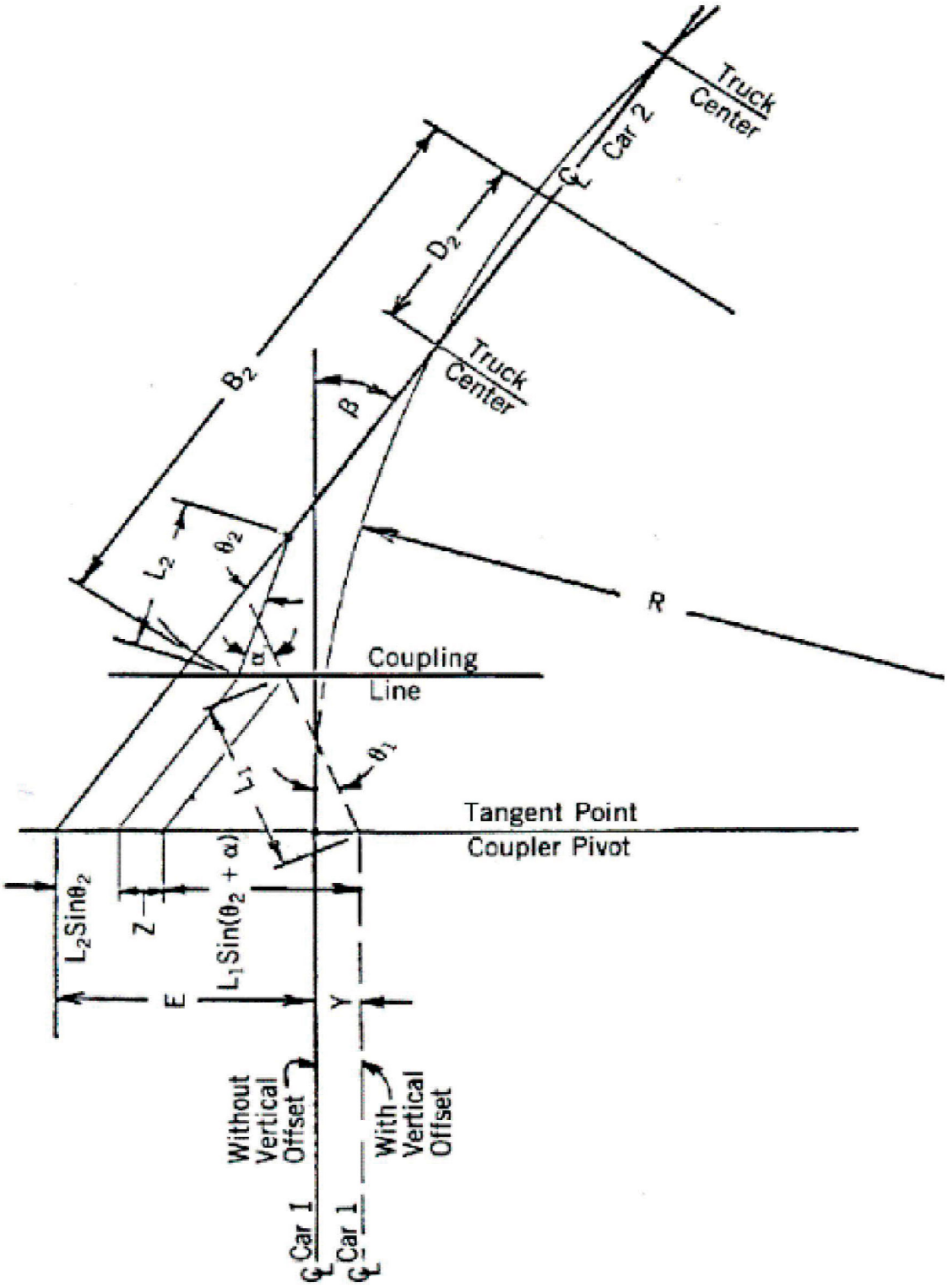
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Minimum radius of curvature as determined by maximum vertical displacement of couplers. Coupling line is at point of tangency. For definition of terms see Equation (3). Note that coupler in car on curve angles up while coupler in car on tangent angles down.

Fig. 2.4 Vertical concave curve and tangent—Equation (3)

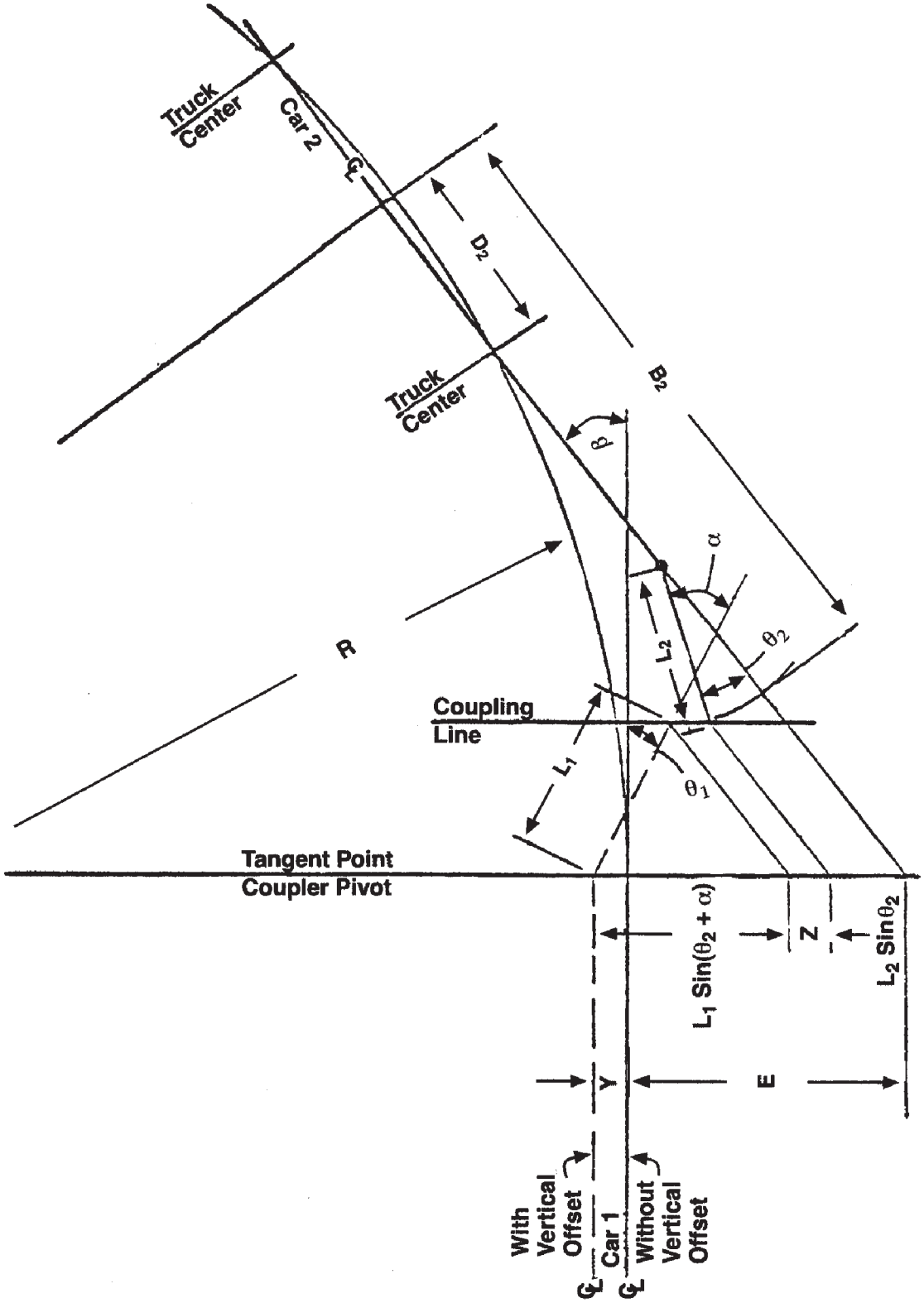
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Minimum radius of curvature as determined by maximum vertical contour angle. Coupler pivot is at point of tangency. For definition of terms see Equation (4). Note that coupler in car on curve angles down and coupler in car on tangent angles up.

Fig. 2.5 Vertical convex curve and tangent—Equation (4)

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Minimum radius of curvature as determined by maximum vertical contour angle. Coupler pivot is at point of tangency. For definition of terms see Equation (4). Note that coupler in car on curve angles up and coupler in car on tangent angles down.

Fig. 2.6 Vertical concave curve and tangent—Equation (4)

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2.1.5 Standard Dimensions

Nominal dimensions are theoretical design points. All other dimensions are to be measured on level tangent track.

2.1.5.1 Height from Top of Rails to Lowest Part of Trucks

Height from top of rails to lowest part of trucks, with truck springs solid and maximum wear conditions, must not be less than 2 3/4 in.

2.1.5.2 Height from Top of Rails to Center of Coupler

Coupler height (top of rails to center of coupler) on new empty cars must not exceed 35 in.

Note: On new empty flatcars with overhang from truck center to striker of 10 ft or greater, the coupler height may be 33½ in. This variance in coupler height compensates for the upswing at the end of the car due to the flexibility of these types of cars when loaded.

2.1.5.3 Height from Top of Rails to Body Center Plate Wearing Surface

Nominal height from top of rails to body center plate wearing surface (empty car):

- For cars with 13 3/4-in.-diameter body center plate—25 1/2 in.
- For cars with 15 3/4-in.-diameter body center plate—25 1/16 in.

2.1.5.4 Nominal Distance from Centerline of Car to Centerline of Body Side Bearing

Nominal distance from centerline of car to centerline of body side bearing shall be 25 in.

2.1.5.5 Height from Body Center Plate Wearing Surface to Body Side Bearing Wearing Surface

- 70-ton cars —4 5/16 in.
- 100-ton and 125-ton cars—5 in.

2.1.5.6 Height of Side Bearing

Height of side bearing shall be 5 1/16 in. ± 1/16 in.

2.1.5.7 Vertical Clearance

Nominal vertical clearance between top of truck unsprung part (side frame, etc.) and vertically located adjacent part of car structure (body bolster etc.) and between top of wheel flange and vertically located adjacent part of car structure for empty car body must be as follows:

	Truck Part	Wheel Flange
Cars having 2 1/2-in. travel truck springs	3 1/2 in.	4 in.
Cars having 3 1/16-in. travel truck springs	4 in.	4 1/2 in.
Cars having 3 11/16-in. travel truck springs	4 1/2 in.	5 in.
Cars having 4 1/4-in. to 4 1/2-in. travel truck springs	5 1/4 in.	5 1/2 in.
Cars using low conveyance truck and 3 1/16-in. travel truck springs.	3 1/8 in.	3 7/16 in.
Cars using low conveyance trucks and 3 11/16-in. travel truck springs.	4 1/2 in.	4 3/16 in.

2.1.5.8 Vertical Clearance Between Top of Coupler Shank and Striker

E60C, E61B, E67B design couplers	7/8 in. maximum, 1/2 in. minimum
E68B, E69A design couplers	7/8 in. maximum, 11/16 in. minimum
F70C design coupler	1 5/16 in.
F79C design coupler	2 9/16 in.
F73A design coupler	4 1/4 in. maximum

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2.1.5.9 Striker Opening Width

E coupler	E60C Design, AAR S-2017-91 striker	11 1/2 in.
	E67B Design, AAR S-2019-86 striker for repairs (rivited application) or S-2021-91 (welded application)	11 1/2 in.
	E68B Design, 13° maximum lateral striker	20 3/4 in.
	E69A Design, 15° maximum lateral striker	30 3/4 in.
F coupler, 13° lateral, S-236 striker	F70C	13 7/16 in. (Note: S-236 striker has a 14 1/4 in. opening)
F coupler, 13° maximum lateral striker	F79C	31 15/16 in.
F coupler, 15° maximum lateral Striker	F73A	19 13/16 in.

2.1.5.10 Coupler Horn Clearance

For cars with 24 5/8 in. draft gear pockets	3 3/4 in.
For cars with 10-in. end-of-car cushioning	11 in.
For cars with 15-in. end-of-car cushioning	16 in.

2.1.5.11 Inside Width of Strikers at Front of Key Slot

Inside width of strikers at front of key slot, minimum	11 5/8 in.

2.1.5.12 Nominal width between Rear Draft Lugs

Nominal width between rear draft lugs	6 7/8 in.

2.1.5.13 Dimensions of Draft Gear Pocket

Standard	Length	24 5/8 in. ± 1/16 in.
	Width	12 7/8 in. + 1/4 in. – 0 in.
	Depth	12 1/2 in. nominal

2.1.5.14 Distance from Front Draft Lugs to Striker Face

For cars with 24 5/8-in. draft gear pocket	With Type E coupler	17 3/4 in.
	With F70C design coupler and 10° lateral	18 in.
	With F70C design coupler and 13° lateral	18 in.
	With F79C design coupler and 13° lateral	31 3/4 in.
	With F73A design series coupler and 15° lateral	49 in.

2.1.5.15 Nominal Distance from Center of Truck to Striker Face for Cars Having Four Wheel Trucks

With 24 5/8-in. draft gear pocket and draft gears having up to 3 1/4-in. nominal travel in buff and E or F couplers	Hopper cars	4 ft 11 1/2 in. (E couplers)
	Other cars with vertical yoke attachments	5 ft 11 1/2 in. (E couplers)
		5 ft 11 3/4 in. (F couplers)
	For box and gondola cars without end platforms	5 ft 5 1/2 in. (E couplers)
		5 ft 5 3/4 in. (F couplers)

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Note: The dimensions shown above may be modified to suit conditions, particularly where long cars are involved and it is desired to obtain the maximum width of car possible by maintaining the truck centers to a minimum and adjusting the overhang accordingly. However, the overhang must not exceed 13 ft 0 in.

2.1.5.16 Draft Gear Follower Thickness

Draft gear follower thickness	2 1/4 in.

2.1.5.17 Axles Sizes, Capacities, and Body Center Plate Diameters

Capacity Nominal	Axle Designation and Journal Size (in.)	Maximum Gross Weight on Rails (lb) [a/]	Diameter of Body Center Plate (in.)
70 ton	E—6 x 11	220,000	13 7/8 [b/]
	L—6 x 8	220,000	13 7/8 [b/]
100 ton	F—6 1/2 x 12	286,000	15 7/8 [b/]
	K—6 1/2 x 9	286,000	15 7/8 [b/]
125 ton[c/]	G—7 x 12	315,000	15 7/8 [b/]
	M—7 x 9	315,000	15 7/8 [b/]

[a/] The total weight on rails shown in the table above is for four-wheel trucks, four axles per car. For cars having six-wheel, eight-wheel, etc., trucks, the total weight on rails is proportional to the number of axles under car.

[b/] +0 / –1/8 in.

[c/] Cars of this weight can be operated only in limited interchange when complying with the requirements of the *Field Manual of the AAR Interchange Rules*, Rule 70 for 315K cars.

2.1.5.18 Center Pin Diameter

For 30- to 125-ton cars, inclusive	1 3/4 in.

2.1.5.19 Jacking Pads

Height of jacking pads—*all cars*	29 in. minimum to 45 in. maximum

2.1.6 Length of Coupler—Long Cars

Lightweight cars on curves can experience coupler angles combined with draft forces (drawbar pull) to produce lateral loads on the rail that can potentially cause wheel lift on the outer rail or wheel flange climb on the inner rail. To provide a margin of safety from these derailment types, the car design should minimize the L/V ratio of the truck sum forces.

2.1.6.1 Length of Coupler—Design Criteria

The limit is derived from the ratio of the truck sum lateral forces over the truck sum vertical forces in empty car conditions $\left(\frac{\text{truck sum lateral}}{\text{truck sum vertical (lightweight)}}\right)$. The maximum allowable L/V ratio is 0.82.

2.1.6.2 Length of Coupler—Analysis Configuration

The analysis shall be conducted with the car on a curve of 10° while subjected to a draft force of 200,000 lb. The test car shall be coupled on both sides to adjacent cars that are equipped with E60 couplers and have a length of 42 ft 3/8 in. over strikers.

2.1.6.3 Length of Coupler—Special Instructions

Cars 85 ft over end sills (85 ft 8 in. over strikers) or longer must be equipped with 60-in. E69 or F73 type couplers in accordance with *Office Manual of the AAR Interchange Rules*, Rule 88C.

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2.1.6.3.1 For Cars Other Than Those Using Span Bolsters Having Integral Draft Arms

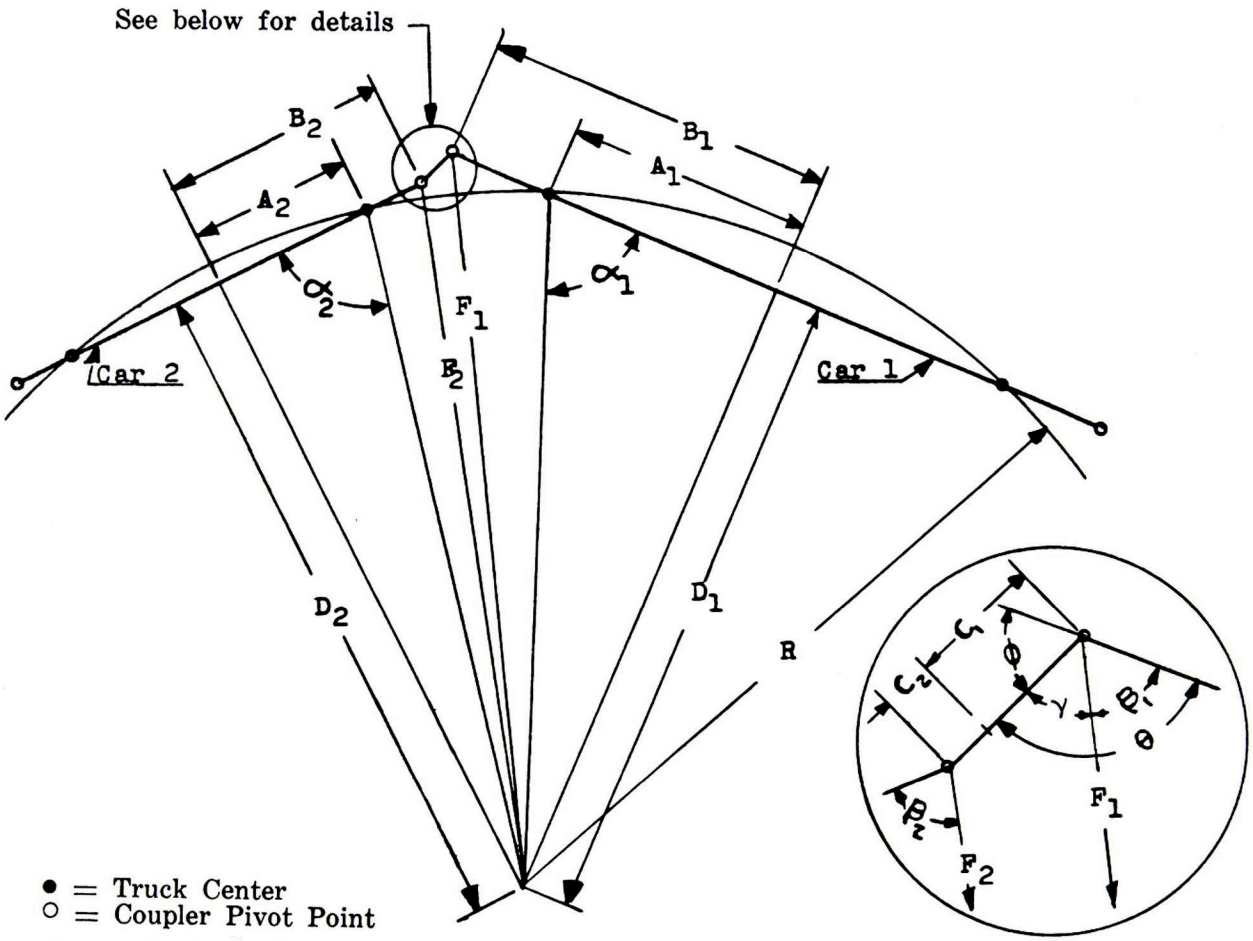
- The method of computing the coupler angles shall be in accordance with paragraph 2.1.6.4.
- The method of computing the truck lateral force shall be in accordance with paragraph 2.1.6.5.

2.1.6.3.2 For Cars Using Span Bolsters Having Integral Draft Arms

- The method of computing the coupler angles shall be in accordance with paragraph 2.1.6.6.
- The method of computing the truck lateral force shall be in accordance with paragraph 2.1.6.7.

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2.1.6.4 Geometry of Coupled Cars on Curve



● = Truck Center
○ = Coupler Pivot Point

C = Coupler length

$A = \frac{\text{Truck center distance}}{2}$

$R = \text{Radius of curve} = \frac{50}{\sin\frac{\text{track curvature in degrees}}{2}}$

$\alpha = \operatorname{acos}\frac{A}{R}$

$D = R\sin\alpha$

R–D = Offset between center of car and center of track

$B = \frac{\text{Length over pulling faces of couplers} - 2(\text{coupler length})}{2}$

$\gamma = \operatorname{acos}\frac{(C_1 + C_2)^2 + F_1^2 - F_2^2}{2(C_1 + C_2)F_1}$

$F = (B^2 + D^2)^{1/2}$ or $F = \frac{B}{\cos\beta}$

$\beta = \operatorname{acot}\frac{B}{D}$

$\theta = \beta_1 + \gamma$

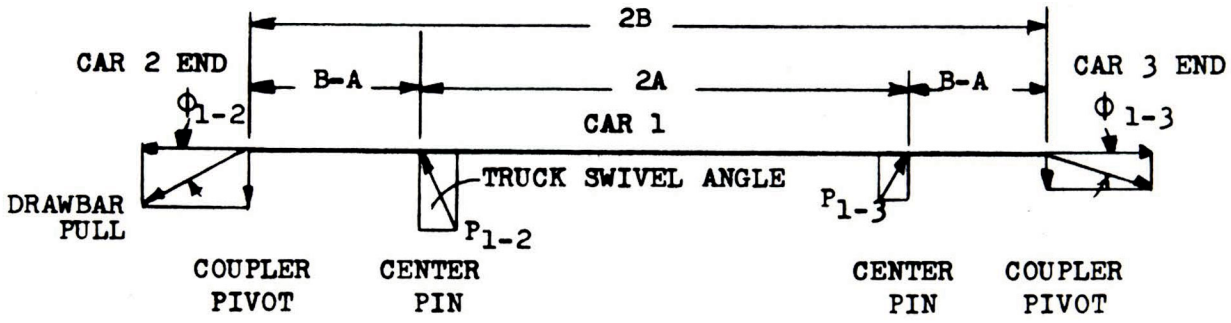
ϕ = Coupler Angle = 180° - θ

[Lateral force on the coupler of Car 1 due to the angularity of its drawbar = (Drawbar force) sin ϕ]

Car 2 = Base car (see paragraph 2.1.4.2.3)

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2.1.6.5 Lateral Forces on Car Due to Angularity of Couplers



Neglecting unbalanced elevation and truck swivel angle, the moment balance becomes this:

$$P_{1\text{-}2} = \frac{\text{Drawbar pull} \cos\phi_{1\text{-}2}[\tan\phi_{1\text{-}2}(A+B) - \tan\phi_{1\text{-}3}(B-A)]}{2A}$$

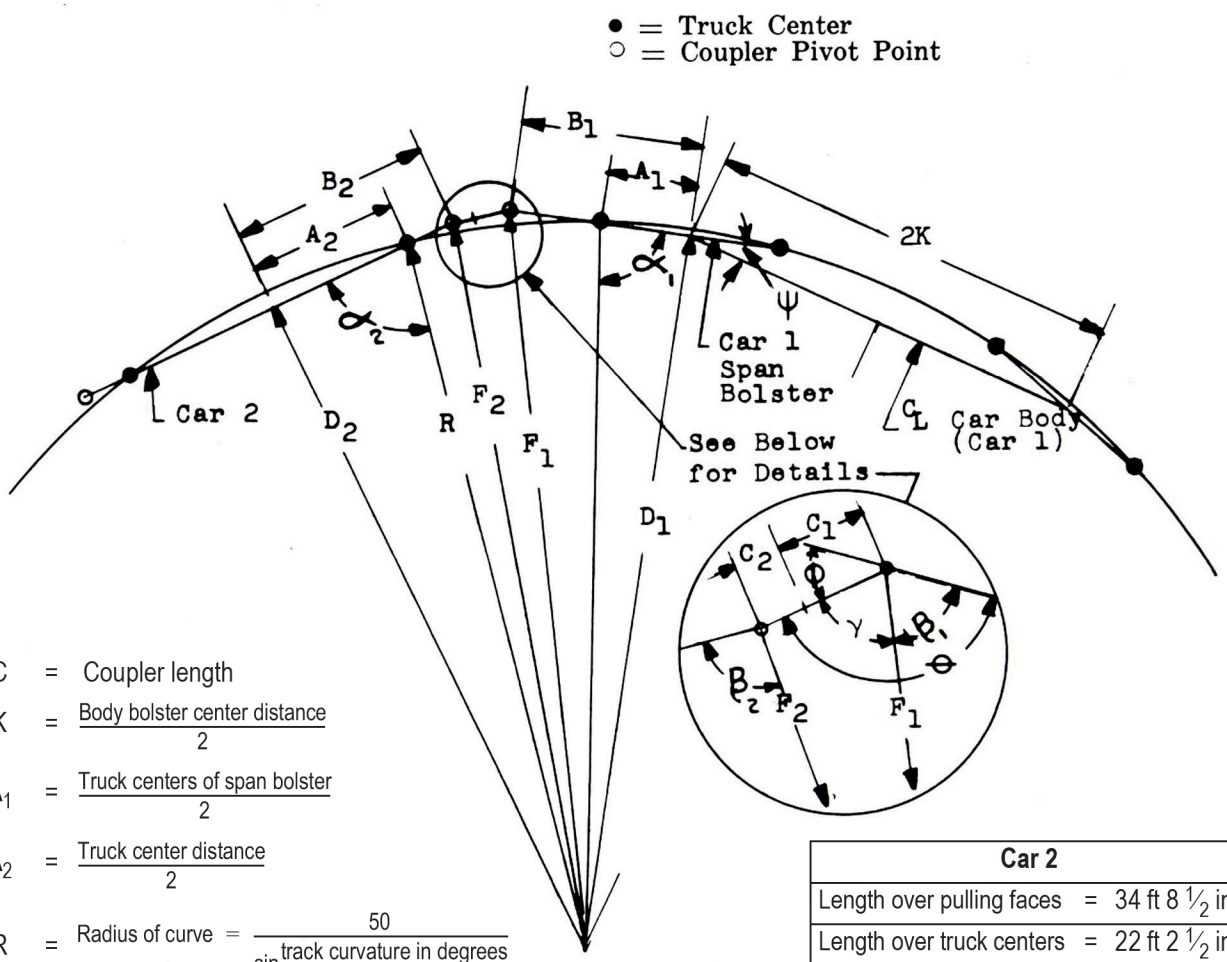
$$P_{1\text{-}3} = \frac{\text{Drawbar pull} \cos\phi_{1\text{-}2}[\tan\phi_{1\text{-}3}(A+B) - \tan\phi_{1\text{-}2}(B-A)]}{2A}$$

Note: $P_{1\text{-}2}$ will be largest when $\phi_{1\text{-}2}$ is large and $\phi_{1\text{-}3}$ is small. This will occur for a long car when Car 2 is short and Car 3 is another long car.

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2.1.6.6 Geometry of Coupled Cars on Curve

(For cars using span bolsters having integral draft arms.)



Car 2		
Length over pulling faces	=	34 ft 8 ½ in.
Length over truck centers	=	22 ft 2 ½ in.
Coupler length	=	28.46 in.

C = Coupler length

$K = \frac{\text{Body bolster center distance}}{2}$

$A_1 = \frac{\text{Truck centers of span bolster}}{2}$

$A_2 = \frac{\text{Truck center distance}}{2}$

$R = \text{Radius of curve} = \frac{50}{\sin\frac{\text{track curvature in degrees}}{2}}$

$\alpha = \operatorname{acos}\frac{A}{R}$

$D = R\sin\alpha$

$B_1 = \frac{(\text{Length over pulling faces of couplers} - 2K) - 2(\text{coupler length})}{2}$

$B_2 = \frac{\text{Length over pulling faces of couplers} - 2(\text{coupler length})}{2}$

$ß = \operatorname{acot}\frac{B}{D}$

$F = (B^2 + D^2)^{1/2}$

$\gamma_{1\text{-}2} = \operatorname{acos}\frac{(C_1 + C_2)^2 + F_1^2 - F_2^2}{2(C_1 + C_2)F_1}$

$\theta_{1\text{-}2} = \beta_1 + \gamma_{1-2}$

$\phi_{1\text{-}2}$ = Coupler angle = $180° - \theta_{1\text{-}2}$

ψ = Angle between span bolster and centerline of car body (equivalent to ϕ_{1-2} of paragraph 2.1.6.4

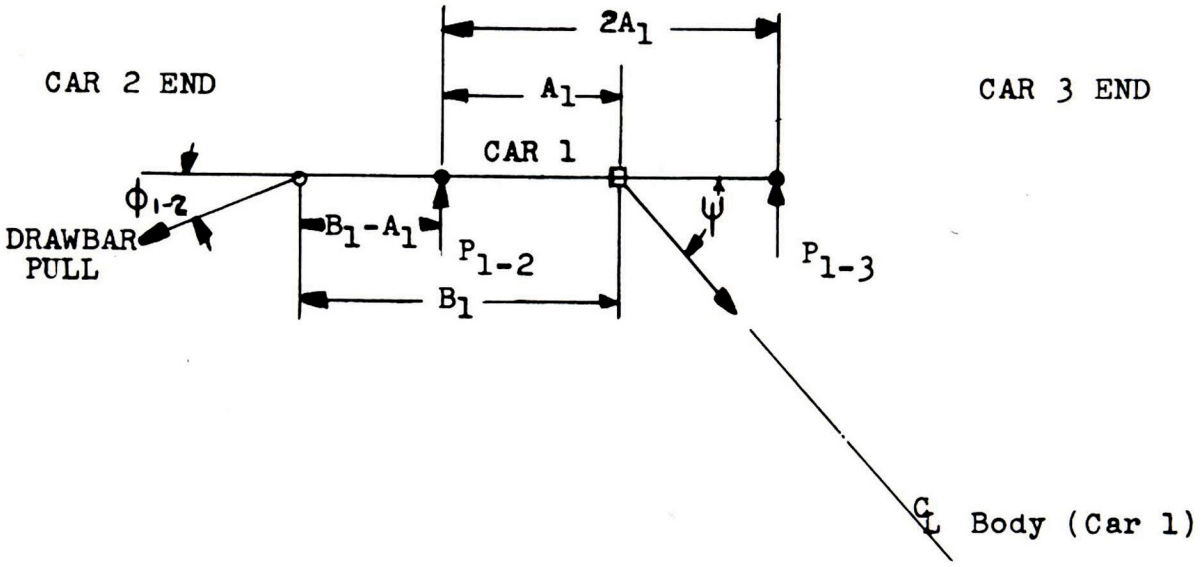
$\psi = 90° - \operatorname{acos}\frac{K}{D_1}$

[Lateral force on the coupler of Car 1 due to the angularity of its drawbar = (drawbar force) sin ϕ]

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2.1.6.7 Lateral Forces on Car Due to Angularity of Couplers

(For cars using span bolsters having integral draft arms)



● = Truck Center Pin
○ = Coupler Pivot
□ = Body Center Pin

Neglecting unbalanced elevation and truck swivel angle,

$$P_{1\text{-}2} = \frac{(\text{Drawbar pull})[(A_1)(\cos\phi_{1\text{-}2})(\tan\psi) + (B_1 + A_1)(\sin\phi_{1\text{-}2})]}{2A_1}$$

$$P_{1\text{-}3} = \frac{(\text{Drawbar pull})[(A_1)(\cos\phi_{1\text{-}2})(\tan\psi) - (B_1 - A_1)(\sin\phi_{1\text{-}2})]}{2A_1}$$

Note: $P_{1\text{-}2}$ will always be the largest lateral force.

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2.2 Definitions

2.2.1 Maximum gross weight on rails	Light weight of car (paragraph 2.2.5) plus the maximum weight of the lading.
2.2.2 Dead load	Weight of car body structure, all fixtures permanently attached thereto, and all appurtenances considered part of the car. For stress analysis, it shall be permissible to use the estimated light weight of the complete car less the weight of the trucks.
2.2.3 Load limit	Maximum weight of lading; and the difference between the marked light weight of car and the maximum gross weight on rails.
2.2.4 Live load	Equal to the load limit, or any percentage thereof, specified for design load.
2.2.5 Light weight	Total weight of the empty car including trucks and all appurtenances considered part of the car.
2.2.6 Nominal capacity	Any capacity, in multiples of 1000 lb, based on its light weight, journal size, and gross weight on rails. In no event may the nominal capacity as shown in the equipment register exceed the load limit. See paragraph 2.1.5.17.
2.2.7 Cubic capacity	Available loading space of the car. The cubic capacity shall be calculated in cubic feet based on the inside dimensions shown on the design drawings. For box, refrigerator, and stock cars, the cubic capacity shall be based on the length between end walls and an inside height equal to the distance from the top of floor or floor racks to the lowest point of the carline, roof, side plate, or other structure, whichever is lower. The inside width should be at the narrowest point. For open top hopper cars and gondola cars, the cubic capacity shall be calculated as that volume below top of sides. For covered hopper cars, the cubic capacity shall be calculated as that volume below the lowest portion of the hatch frames.

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CHAPTER 3
MATERIALS—BODY STRUCTURE

3.1 General

3.1.1 The materials used in freight car construction shall be of structural and/or car building quality as defined in general by the specification listed hereafter.

3.1.2 All metals, polymers, wood products, fasteners, and welding materials used in freight car construction shall be specified in accordance with generally accepted standards or grades such as AAR or other specifications.

3.1.3 The allowable stresses of paragraph 4.2 shall be compatible with the mechanical properties of the material used.

3.1.4 The term "other specifications" as used herein shall include specifications of the American Society for Testing and Materials (ASTM), the Association of American Railroads (AAR), other recognized agencies', manufacturers' and car builders' specifications for proprietary grades.

3.1.5 Where materials are specified to AAR, ASTM, or other specifications, the latest revision thereof shall apply.

3.1.6 Where material is to be welded, its suitability for welding by the proposed method and procedure shall be documented.

3.1.7 All steels defined by the specifications listed hereafter shall be manufactured by one or more of the following processes: Open hearth, electric furnace, or basic oxygen. Bessemer steel will not be acceptable.

3.1.8 Where improved corrosion resistance is desired in carbon steel, copper bearing steel may be specified. Copper content shall be a minimum of 0.20% and where weldability is a factor, it is preferable that copper content does not exceed 0.30%.

3.1.9 The following car body structural members must be made from steel having a minimum Charpy V-notch value as obtained under specification ASTM A673 of 20 ft·lb at 0 °F on a heat lot basis:

- Fabricated body bolster top cover plates, (excluding hopper cars and cars that have plate steel deck forming top cover plate)
- Center sill webs and bottom cover plates from bolster to end of car made of plate or bar steels
- Entire center sill bottom cover plate of flatcars of all types
- Shear plates on cars not equipped with continuous center sill
- Hot rolled center sill sections outboard of the body bolster

3.1.10 Protective Coatings

In cases where coating layers are specified from different suppliers, the compatibility of the system shall be satisfactorily established.

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CHAPTER 4
DESIGN

4.1 Loads and Forces

4.1.1 General

The car structure shall be designed for the following loads and forces singly and in combination as applicable:

- Dead load (paragraph 4.1.2)
- Live load (paragraph 4.1.3)
- Lift truck wheel loads (paragraph 4.1.4)
- Vertical loads on coupler (paragraph 4.1.5)
- Jacking load (paragraph 4.1.6)
- Roof loads (paragraph 4.1.7)
- Buff and draft load (paragraph 4.1.8)
- Compressive end load (paragraph 4.1.9)
- Impact load (paragraph 4.1.10)
- Vertical forces induced by horizontal impact (paragraph 4.1.11)
- Longitudinal force on ends of box and gondola cars (paragraph 4.1.12)
- Longitudinal force on bulkheads of flatcars (paragraph 4.1.13)
- Lateral forces (paragraph 4.1.14)
  - Bulging pressure of bulk material on sides of cars (paragraph 4.1.14.1)
  - Lateral pressure from leaning loads on sides of gondola cars (paragraph 4.1.14.2)
  - Effects on centrifugal force (paragraph 4.1.14.3)
- Adjustable bulkheads—box cars (paragraph 4.1.15)
- Lateral and longitudinal forces on fixed side wall fillers—box cars (paragraph 4.1.16)
- Roping staples (paragraph 4.1.17)

4.1.2 Dead Load

The dead load as defined in paragraph 2.2.2 may be distributed to the underframe members by either of the following methods:

4.1.2.1 Uniformly distributed over the area of the underframe bounded by the side sills and the end sills.

4.1.2.2 Apportioned in accordance with the estimated dead weight of the components, considered as the superimposed dead load on the respective members.

4.1.2.3 For cars intended for bulk loading, the dead load may be added to the live load and distributed to the supporting members in accordance with the configuration of the supporting structure.

4.1.3 Live Load

Cars intended for bulk loading shall be considered loaded to full cubic capacity, and distribution of the load to supporting members shall be in accordance with the configuration of the supporting structure and the load.

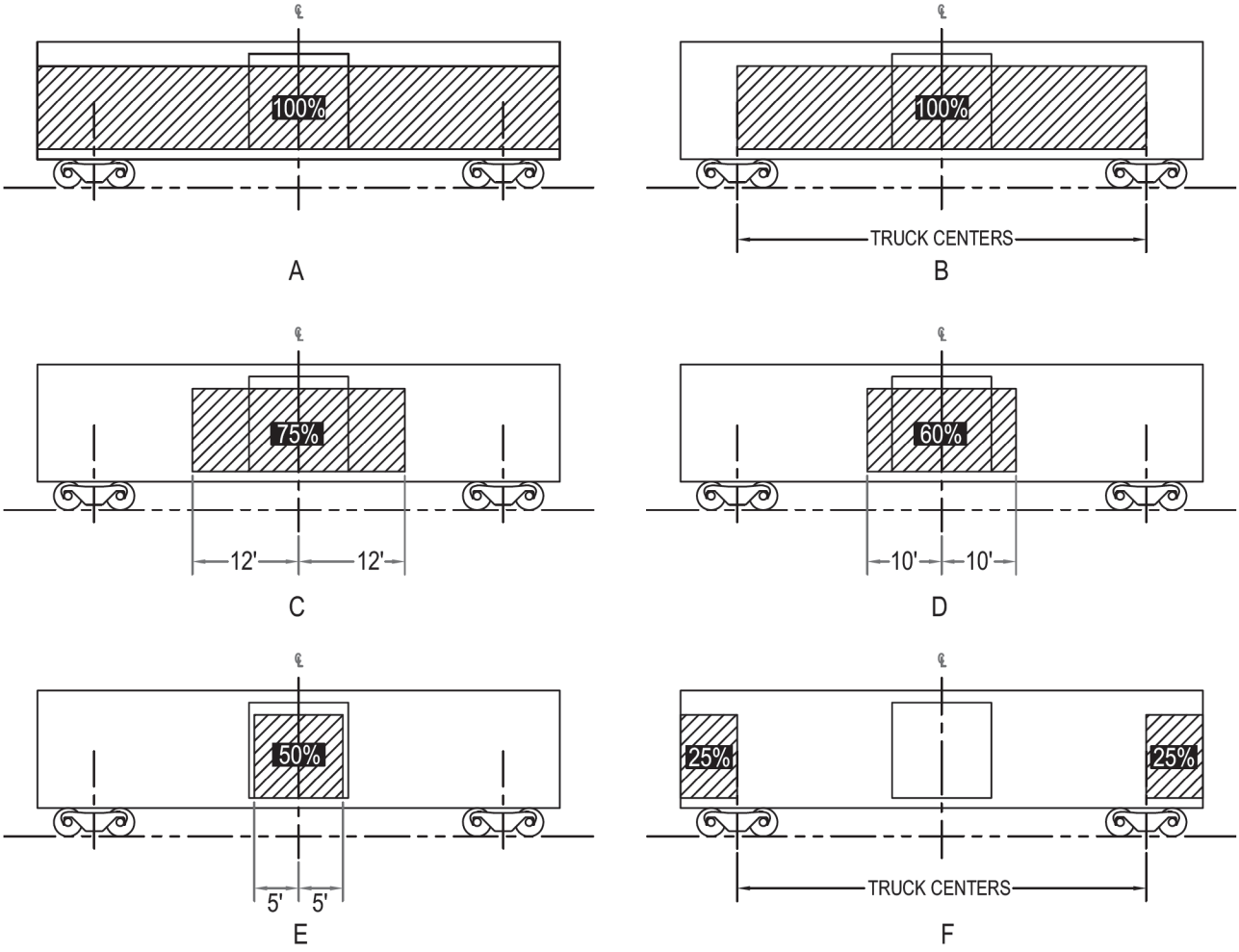
The live load as defined in paragraph 2.2.4 shall be distributed to box, flat, gondola, and center beam cars in accordance with paragraphs 4.1.3.1, 4.1.3.2, 4.1.3.3, 4.1.3.4, and 4.1.3.5.

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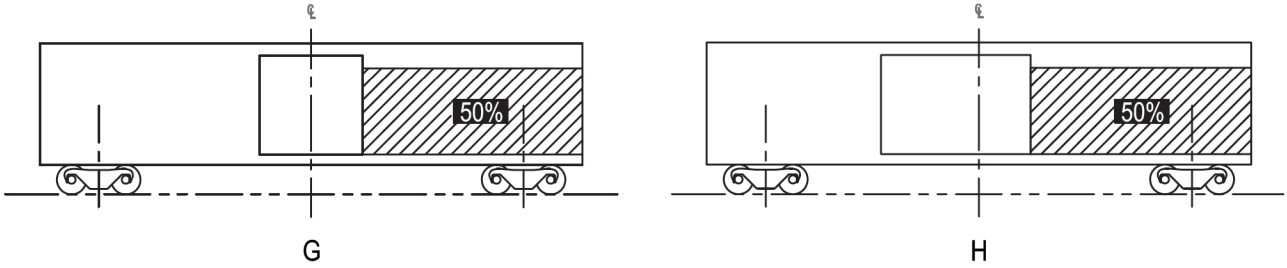
Special type cars designed for specific commodities only, where the live load as defined in paragraph 2.2.4 is not distributed in accordance with paragraphs 4.1.3.1, 4.1.3.2, 4.1.3.3, 4.1.3.4, and 4.1.3.5 must be marked in accordance with AAR *Manual of Standards and Recommended Practices (MSRP)*, Section C-I, Standard S-910 Marking of Freight Cars. An appropriate note shall be included in the equipment register giving a general description of the cars, the type of lading that can be handled, and an indication of the distribution of the live load.

In cases where the load is shown as concentrated on a primary structural member (for example, center sill or side sill), the distribution of the load to support members shall be in accordance with the configurations of the support members and the load.

4.1.3.1 Box Cars with Inside Length Less than 61 ft



Note: The live load as indicated on diagram (e) maybe placed at any location between truck centers that produces the critical loading condition on the center sill or any other member.

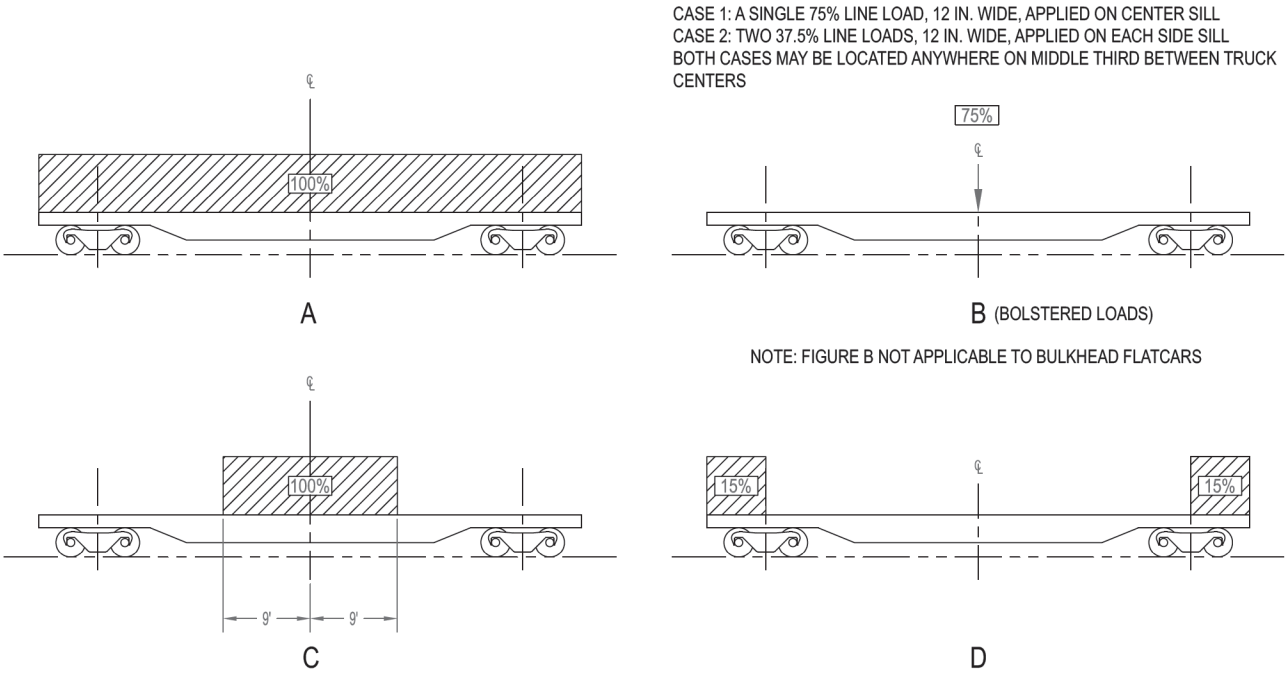


Note: For cars with inside length 61 ft or greater, only Figures A and B apply.

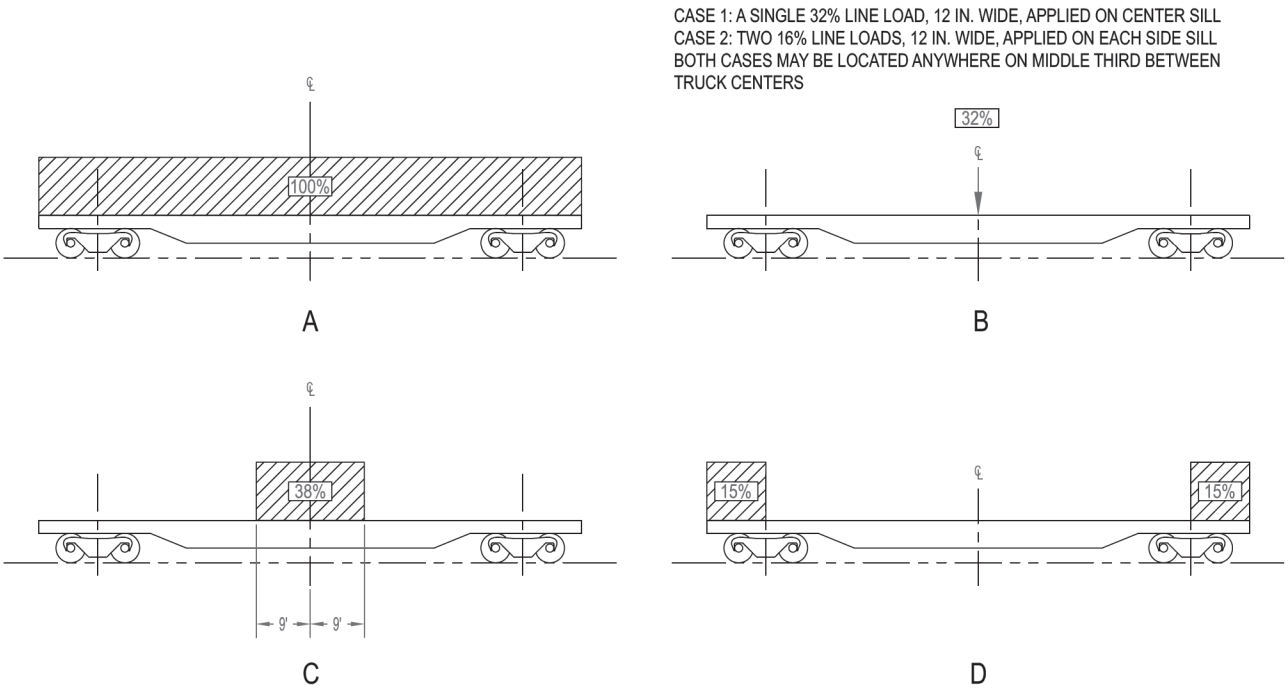
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4.1.3.2 Flatcars

4.1.3.2.1 Flatcars with Deck Length of 68 ft or Less

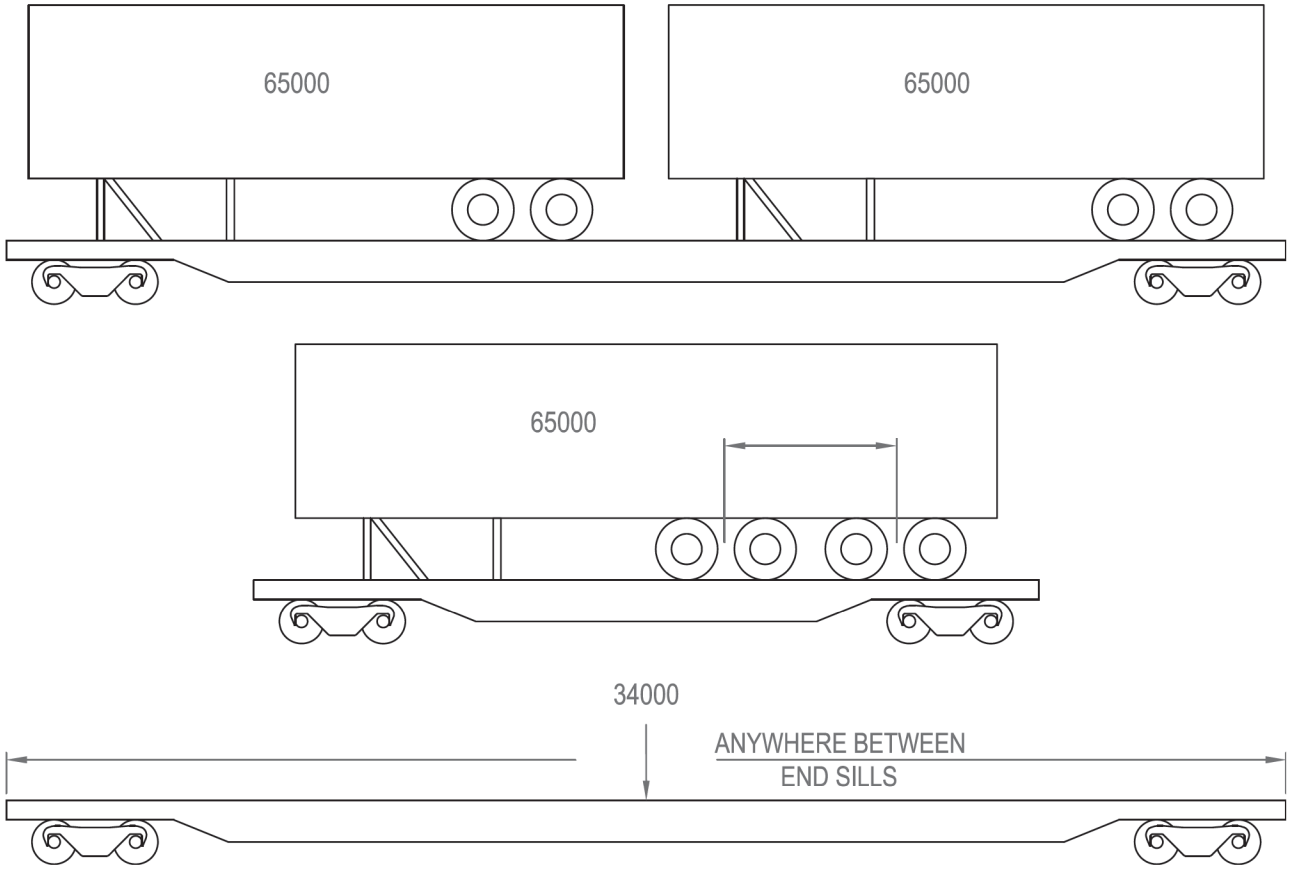


4.1.3.2.2 Flatcars with Deck Length Longer than 68 ft

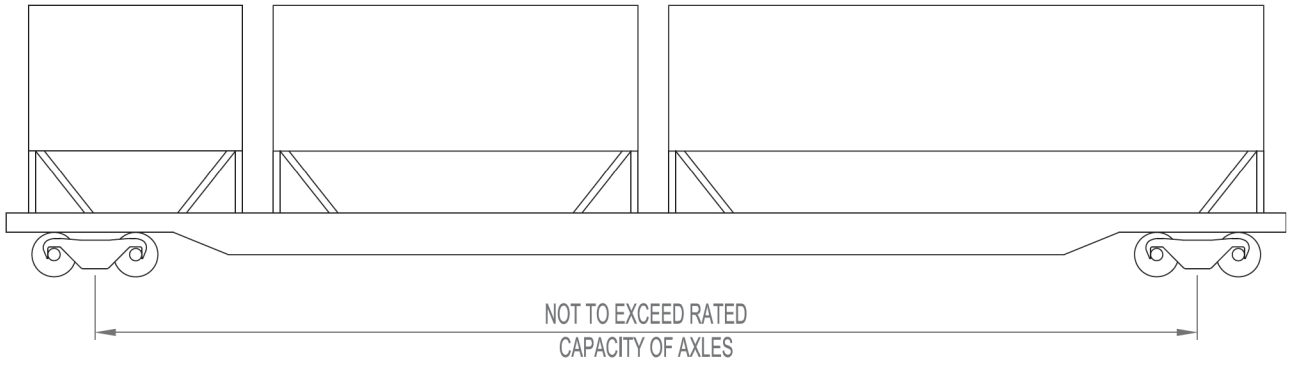


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4.1.3.2.3 Live Load Distribution—Trailer Transport Cars

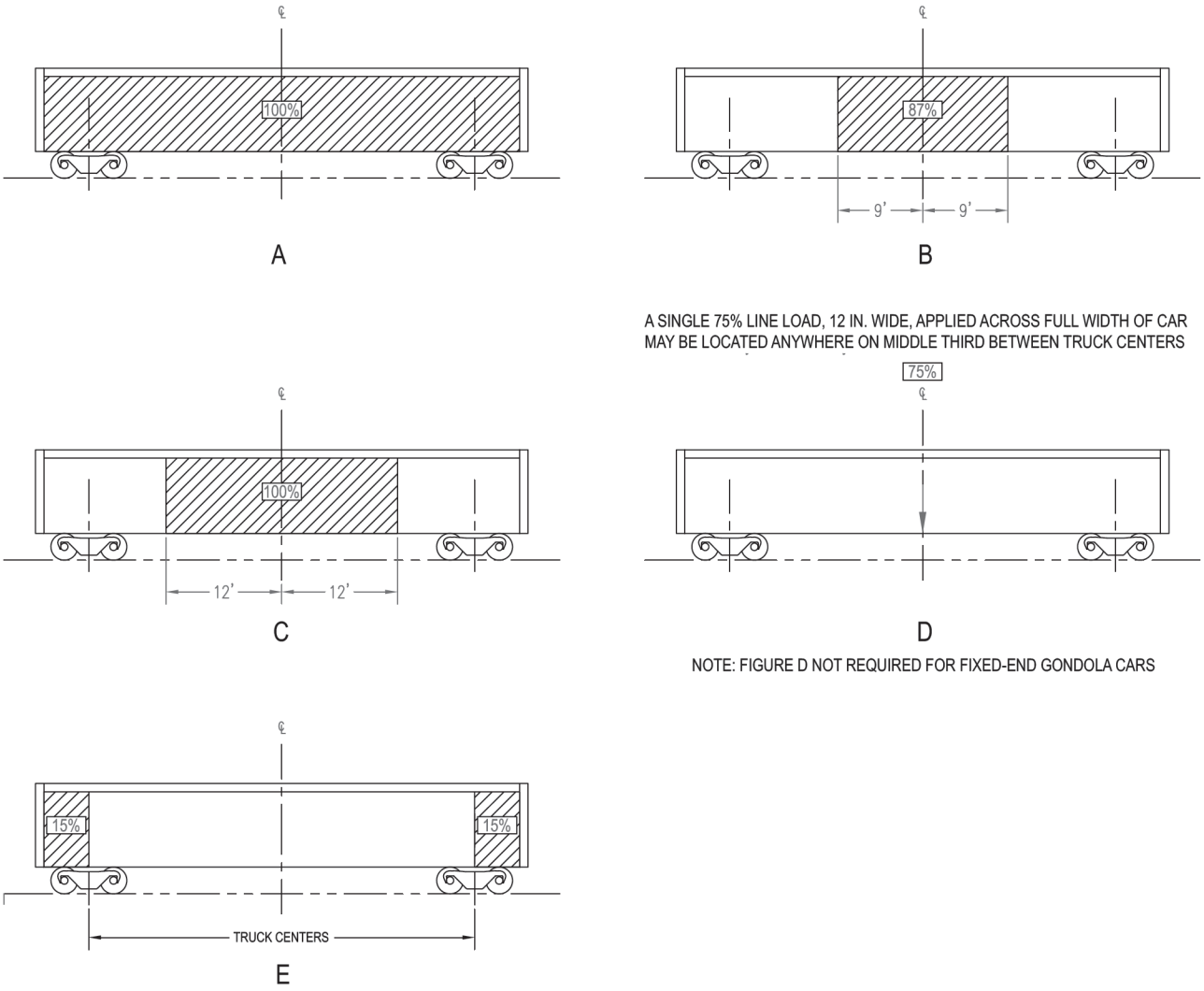


4.1.3.2.4 Live Load Distribution—Container Transport Cars



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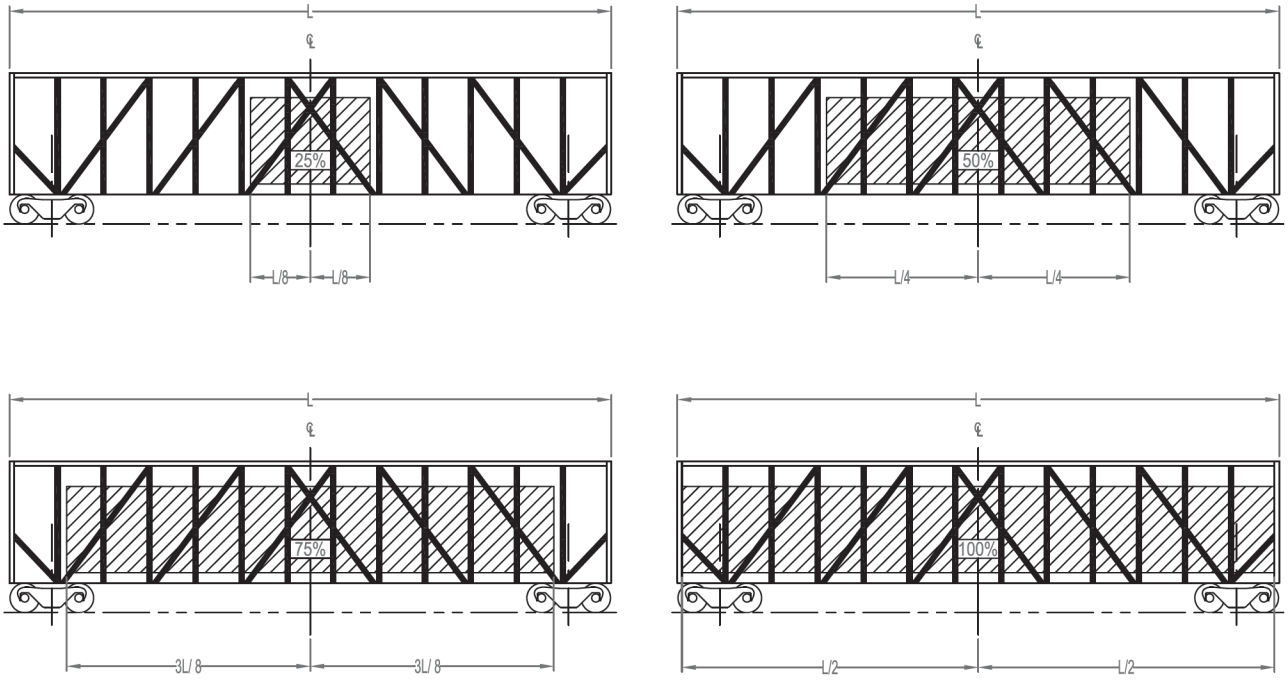
4.1.3.3 Gondolas



Note: For high-side gondolas with inside height greater than 6 ft 6 in., only Figure A applies.

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4.1.3.4 Center Beam Cars



LOAD TO BE EQUALLY DISTRIBUTED ON BOTH SIDES OF THE CENTER BEAM

4.1.3.5 Trailer/Container Cars

4.1.3.5.1 Trailer transport car structure shall be designed to sustain the static vertical load imposed by the most critical combinations of 65,000-lb gross weight trailers for which the car is designed with the most critical tandem settings as listed in paragraph 4.1.3.5.1.1 The calculated axle and kingpin loads shall be based on a uniformly distributed load of 61,000 lb and a concentrated bogie load of 4,000 lb. *MSRP* Section I, Specification M-931 is referenced as information.

4.1.3.5.1.1 Critical Tandem Settings

Table 4.1 Critical tandem settings

Trailer length overall (ft)	53	48	45	40	28
Kingpin-to-nose distance (in.)	36	36	36	36	30
Rear of trailer to rear axle (in.)	36 to 150	36 to 115	27 to 115	27 to 115	36
Axle centers (in.)	49	49	49	49	N/A
Maximum load per axle (lb)	22,000	22,000	22,000	22,000	22,000
Maximum kingpin loads (lb)	30,000	30,000	30,000	30,000	30,000

4.1.3.5.1.2 For cars subjected to roll-on/roll-off type of trailer loadings, the car structure shall also be designed for loads imposed by tires of tractors used in loading operations. The tractors shall be considered to have a two-wheel rear axle, each wheel supporting 17,000 lb on a 5-in.-long by 8-in.-wide tire imprint applied to the car deck at any location along the normal path followed by the tractor tires.

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4.1.3.5.2 The container transport car structure designed for transporting a dry cargo van-type container as outlined in *MSRP*, Section I, Specification M-930, shall be designed to sustain loads imposed by that specification.

Cars designed for shipping multiple container sizes shall be designed to sustain loads imposed by the following container sizes and weights:

Table 4.2 Container sizes and weights

Nominal Length (ft)	Gross Weight (lb)
53	67,200
48	67,200
45	67,200
40	67,200
20	52,900

and/or any combination of the above where the combined overall lengths do not exceed the design capability of the container support system configuration, and weight supported by any car axle is not greater than its rated capacity.

4.1.3.5.3 Flatcars designed to be used in combination with auto rack superstructure shall meet the requirements of *MSRP*, Section N, Specification M-956.

4.1.4 Lift Truck Wheel Loads

4.1.4.1 General service box cars and flatcars shall be designed for loading and unloading by forklift trucks with front axle loads of 50,000 lb minimum or a wheel load of 25,000 lb minimum. The treads of the truck wheels shall be assumed to be on 32-in. centers and the tire print to be 13 1⁄2 × 5 3⁄8 in. with 16-in. width of tire. The wheel loads shall be located to provide critical design load for the member under consideration.

4.1.4.2 Refrigerated and insulated box cars can be designed with front axle loads of 25,000 lb.

4.1.5 Vertical Loads on Coupler

The car structure shall be designed to withstand the following loads:

4.1.5.1 The car shall be considered to be loaded such that its weight is 125% of its normal gross rail load under the most severe loading condition in paragraph 4.1.3. A force shall be applied to the coupler shank immediately adjacent to either the striker face or the face of the cushion unit body sufficient to lift that end of the car, including its truck and wheel sets. The opposite end of the car shall be supported at the center plate. Maximum allowable stress shall be as defined in paragraph 4.2.2.5. Span bolster cars or cushion underframe cars having a sliding sill shall be excluded from the requirements of this paragraph.

4.1.5.2 For cushion underframe cars having sliding sills, a load representing 125% of the upward force required to lift the fully loaded car body and nearest truck, including wheel sets, shall be applied to the sliding center sill in a plane at the ends of the fixed center sills. Maximum allowable stress shall be as defined in paragraph 4.2.2.5.

4.1.5.3 All cars shall have the capability to sustain a vertical up and down load of 50,000 lb at the pulling face of the coupler.

4.1.6 Jacking Load

Car structure shall be designed to sustain 40% of gross rail load applied to each jacking pad, with stresses not to exceed yield strength. The jacks shall be placed under the portion of the car extending outside of rails, preferably under the side sills at the body bolster or under the ends of the body bolster. Suitable jacking pads shall be provided for the jacks. Lifting provisions shall meet the requirements of the *MSRP*, Section C-I, Standard S-234.

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4.1.7 Roof Loads

The roof of box cars or other house-type cars shall be capable of sustaining a uniformly distributed load of 15 lb/ft^2 of projected area plus a concentrated load of 300 lb.

4.1.8 Buff and Draft Loads

The car structure shall be designed to sustain a 390,000-lb draft (tensile) or 350,000-lb buff (compression) drawbar and/or train-action load applied on the front or rear draft lugs at each end of the car along the nominal centerline of the couplers. In addition, front draft lugs and their attachment shall be designed to 500,000 lb with an additional 1.8 load factor applied. The maximum allowable stress shall be yield. The attachment stress for this requirement may be calculated by dividing the force by the effective area of the attachment to the sill.

4.1.8.1 Buff and Draft Load Distribution

4.1.8.1.1 Buff and Draft Load Distribution for Cars with Conventional Draft Gears

The buff or draft load for cars in this group shall be distributed to all longitudinal underframe members between bolsters based on the relative flexibility of the bolster and all the longitudinal underframe members. The attachment of such members to all transverse members of the underframe must be sufficient to provide the necessary restraint and develop the required strength for the increment of the buff or draft load distributed to each member.

4.1.8.1.2 Draft Load Requirements for Buff and Draft Load Distribution for Cars with Special Cushioning Devices

4.1.8.1.2.1 For cars equipped with sliding sills, the entire buff or draft load shall be sustained by the sliding sill.

4.1.8.1.2.2 For cars equipped with fixed center sills and special cushioning devices mounted in or at the end of the center sills, the buff or draft load shall be distributed as specified in paragraph 4.1.8.1.1.

4.1.9 Compressive End Load

4.1.9.1 The car structure shall be designed to sustain a compressive columnar load of 1,000,000 lb applied at the rear draft lugs at each end of the car on the nominal centerline of couplers. The car structure shall be so proportioned that the induced stresses resulting from the compressive columnar load shall not exceed the allowable unit stresses given in paragraph 4.2.

4.1.9.2 Compliance with this requirement may be demonstrated analytically or by tests.

4.1.9.3 When tests are made to demonstrate capability, they shall be conducted in accordance with requirements of paragraph 11.3.3.1 using appropriate equipment to register loads and strains.

4.1.9.4 It is recognized that the proportionality of stress to strain ceases at the proportional limit. The conversion of test strain to stress may be made by multiplying the strain by the appropriate modulus of elasticity or, when a permanent set exists, by subtracting this value from the maximum strain and multiplying by the appropriate modulus of elasticity, provided a subsequent loading at the same value causes no appreciable increase in permanent set.

4.1.9.5 Compressive End Load Distribution

4.1.9.5.1 Compressive End Load Distribution for Cars with Conventional Draft Gears or End-of-Car Cushioning

The compressive end load for cars in this group shall be distributed to all longitudinal underframe members between bolsters based on the relative flexibility of the bolster and all the longitudinal underframe members. The attachment of such members to all transverse members of the underframe must be sufficient to provide the necessary restraint and develop the required strength for the increment of the compressive end load distributed to each member.

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4.1.9.5.2 Compressive End Load Distribution for Cars with Sliding Sills

For cars equipped with sliding sills, the entire compressive end load shall be sustained by the sliding sill.

4.1.10 Impact Load

The car structure shall be designed to sustain the reaction and inertia forces resulting from a single-ended impact. The impact strength capability must be demonstrated analytically and/or by test of a prototype car or car component at the option of the Equipment Engineering Committee. When prototype car tests are made to demonstrate this capability, they shall be conducted in accordance with paragraph 11.3.4.1 using appropriate instruments to record loads and strains.

4.1.10.1 Impact Load for Cars with Conventional Draft Gears

Cars in this group shall be shown to have the structural capability of withstanding a coupler force of 1,250,000 lb applied to one end of the car.

4.1.10.2 Impact Load for Cars with Special Cushioning Devices

4.1.10.2.1 For cars in this group having a sliding sill, the car underframe (excluding the sliding sill) shall have a structural capability of withstanding a coupler force of 500,000 lb applied at one end of the car. The sliding sill, rear draft stops, and their connections shall have a structural capability of withstanding a static compressive end load of 1,000,000 lb.

4.1.10.2.2 For cars in this group using end-of-car cushioning, the car shall have a structural capability of withstanding the following coupler forces applied at one end of the car:

Table 4.3 Structural requirements for EOC-equipped cars

Length of Travel in Buff (in.)	Coupler Force (lb)
Less than 6	1,250,000
6 but less than 9	1,000,000
9 but less than 14	750,000
14 and greater	600,000

4.1.10.2.3 The center sill, rear draft stops, and their connections shall have a structural capability of withstanding a static compressive end load of 1,000,000 lb.

4.1.10.3 Impact Load Distribution

The design coupler force loads specified in paragraphs 4.1.10.1 and 4.1.10.2 shall be considered to result in an underframe load of 100% of said specified value applied to the center sills at the rear draft lugs.

The above underframe loads shall be distributed to the underframe structure as follows:

4.1.10.3.1 Box Cars Utilizing Conventional Longitudinal Stringer Construction with Transverse or Other Flooring Not Capable of Transmitting Longitudinal Forces

4.1.10.3.1.1 For cars utilizing conventional construction, the distribution of the underframe load to the longitudinal members shall be based on the relative flexibility of the bolster and all longitudinal members when subjected to applied longitudinal loading and car body inertial forces.

4.1.10.3.1.2 For cars utilizing special cushioning devices with through sliding sills, the sliding sill shall sustain 100% of the underframe load. The underframe load is applied to the fixed center sills by the cushioning device, and the distribution shall be on the relative flexibility of the bolster, fixed center sills, side sills, and stringers when subjected to applied longitudinal loading and car body inertial forces.

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4.1.10.3.2 Box Cars Utilizing Plate Floor Construction Capable of Transmitting Longitudinal Load

Such cars shall be designed so that the underframe load is distributed in accordance with paragraph 4.1.10.3.1 An effective width of floor may be considered acting with each longitudinal member.

4.1.10.3.3 Gondola Cars

The underframe load shall be distributed in accordance with paragraphs 4.1.10.3.1 and 4.1.10.3.2.

4.1.10.3.4 Hopper Cars—Open and Covered

For conventionally constructed cars, the distribution of the underframe load to the longitudinal members shall be based on the relative flexibility of the bolster, center sill, and side sills when subjected to applied longitudinal loading and car body inertial forces.

4.1.10.3.5 Conventional Flatcars—General Purpose

The underframe load shall be distributed in accordance with paragraphs 4.1.10.3.1 and 4.1.10.3.2.

4.1.10.3.6 Flatcars—Trailer and Container Transport

4.1.10.3.6.1 For cars not utilizing cushioning in the trailer tie-down device, it shall be assumed that the horizontal load at the trailer kingpin is the design coupler force multiplied by the ratio of gross trailer weight to gross weight on rails. However, this horizontal load need not exceed 240,000 lb. Gross weight of trailers shall be as defined in paragraph 4.1.3.5.

4.1.10.3.6.2 For cars utilizing auxiliary cushioning in the trailer tie-down device, the maximum horizontal kingpin force is to be determined by test in accordance with paragraph 8.2.4. The end load from impact should be distributed to the underframe in accordance with paragraph 4.1.9.5.

4.1.10.3.6.3 For cars designed for transporting dry cargo van-type containers as described in *MSRP*, Section I, Specification M-930, the container pedestal reaction loads shall be assumed to be not greater than loads stated in *MSRP*, Section I, Specification M-952.

4.1.10.3.7 Bulkhead Flatcars

The underframe load shall be distributed in accordance with paragraphs 4.1.10.3.1 and 4.1.10.3.2.

4.1.10.3.8 Center Beam Car

The distribution of the impact load to the longitudinal members of the underframe shall be based on the relative flexibility of the structure.

4.1.11 Vertical Forces

The car body bolster shall be designed for forces resulting from vertical acceleration induced by the horizontal impact force.

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4.1.11.1 A dynamic factor to be applied to the dead and live load bending moments can be computed by summation of moments about the center of gravity of the loaded car as defined in paragraph 2.1.3. The vertical load amplification factor at the body bolster on the impact end then becomes the following:

$$a = 1 + \frac{2hH}{bW}$$

where
- a = amplification factor
- b = distance between truck centers (ft)
- h = vertical distance, centerline of coupler to maximum center of gravity height (ft)
- W = rail load limit less weight of trucks (lb)
- H = horizontal impact force (lb) (from paragraph 4.1.10)

Total bending moment at bolsters then becomes

a × the static dead and live load bending moment

Center beam cars are exempt from this requirement.

4.1.11.2 Dynamic load factors for other members in the structure may be assumed to vary linearly from the centerline of car to the impacted end and the amplification factor may be considered as follows:

$$a = 1 + \frac{12Hhx}{WL^2}$$

where a, h, H, and W are described in paragraph 4.1.11.1
- L = inside length of car (ft)
- x = distance from centerline of car to point in question(ft)

4.1.11.3 Vertical acceleration force for trailers to be used in conjunction with peak horizontal kingpin loads of paragraph 4.1.10.3.6 may be computed by considering a longitudinal trailer acceleration of a magnitude equal to the design kingpin load divided by the trailer gross weight for trailer configurations of paragraph 4.1.3.5. The vertical force is then computed by taking moments about the trailer center of gravity, which may be assumed as 83 in. above the car deck.

4.1.12 Longitudinal Force on Ends of Box Cars and Gondola Cars

4.1.12.1 The ends of box cars and gondola cars shall be designed to withstand a horizontal force induced by the lading as indicated below without exceeding the yield strength of the material.

Table 4.4 Longitudinal force on ends of box cars

Nominal Capacity of Car (ton)	Total Force on Ends (lb)	Percentage of Total Force Uniformly Distributed	
		Top Half	Bottom Half
70 (220K)	220,000	35–45	65–55
100 (263K) 110 (286K)	260,000 284,000	35–45	65–55

Table 4.5 General purpose gondola cars

Nominal Capacity of Car (ton)	Total Force Uniformly Distributed Over Entire End (lb)
70 (220K)	195,000
100 (263K) 110 (286K)	275,000 300,000

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4.1.12.2 For cars designed for special service, such as wood chips, cellulose acetate, coal or other commodities that do not produce longitudinal forces comparable to those specified above, the longitudinal forces should be reduced or increased accordingly as determined by calculations or tests.

4.1.12.3 For cars equipped with special cushioning devices, the above longitudinal forces may be reduced commensurate with the reduction obtained in body force with the type of cushioning device to be used.

4.1.12.4 Compartmentalized cars having permanent, immovable, interior partitions or similar structure may have the longitudinal forces reduced in proportion to the number of compartments. This does not apply to box cars with movable interior load dividers or similar devices.

4.1.13 Longitudinal Force on Bulkheads of Flatcars and Center Beam Cars

4.1.13.1 The bulkheads on flatcars and center beam cars shall be designed for a longitudinal force induced by the lading under end impact. This force shall be the design coupler force multiplied by 55% of the ratio of load limit to gross weight on rails, and it shall be considered as a uniformly distributed load over the face of the bulkhead with the resultant force at mid height. Members that are not continuous through the body bolster shall be disregarded. The induced stresses resulting from this force shall not exceed the allowable unit stresses given in paragraph 4.2.2.6.

4.1.13.2 The bulkheads on flatcars and center beam cars used exclusively in pulpwood service and having permanently canted decks are to be designed for a longitudinal force induced by the lading under end impact. This force shall be the design coupler force multiplied by 50% of the ratio of load limit to gross weight on rails, and it shall be considered as a uniformly distributed load over the face of the bulkhead with the resultant force at mid height. The induced stresses resulting from this force shall not exceed the allowable unit stresses given in paragraph 4.2.2.6.

4.1.14 Lateral Forces

4.1.14.1 Lateral pressure of granular, lump, or pulverized bulk material shall be considered in the design of the containing structure of all cars in which such pressures my be active, e.g., box, hopper and gondola cars.

For closed-top cars, the lateral pressure may be computed by the following Rankine formula:

$$P = \frac{1}{2} W h^2 \left(\frac{1 - \sin\phi}{1 + \sin\phi} \right)$$

where P = lateral force (lb/ft of length)
W = weight of lading (car fully loaded—see paragraph 2.1.3) (lb/ft^3)
h = depth of lading (not including heap)(ft)
ϕ = static angle of repose (25°)

For open top cars:

$$P = \frac{1}{2} W (h + 1)^2 \left(\frac{1 - \sin\phi}{1 + \sin\phi} \right)$$

The lateral force is to be distributed vertically so that it is at maximum at the floor line decreasing uniformly to zero at the top surface of the lading.

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4.1.14.2 The effects of leaning loads, such as diagonally stacked plates resting against the sides of gondola cars with inside height of 6 ft 6 in. or less, shall be provided for in the car design.

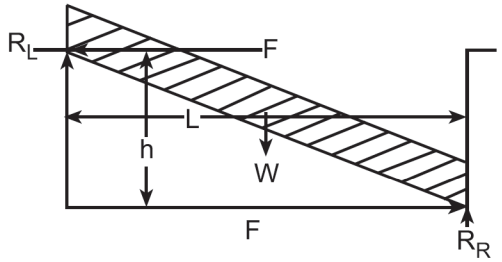
The loading condition may be assumed to be 75% of the load limit uniformly distributed. The total lateral force may be computed by the formula:

$$F = \frac{WhL}{2(h^2 + L^2)}$$

$$R_L = \frac{WL^2}{2(h^2 + L^2)}$$

$$R_R = \frac{W(2h^2 + L^2)}{2(h^2 + L^2)}$$

where F = Total lateral force (lb)
W = 75% of load limit (lb)
L = Inside width of car (ft)
h = Inside height of car (ft)
R_R = Reaction at right side
R_L = Reaction at left side



The lateral force determined by the above formula shall be resisted by the bolster and crossbearer posts only.

The total lateral force to be resisted by each corner post of drop end gondola cars shall be 50% of the total force as determined above.

4.1.14.3 The effects of centrifugal force and/or acceleration shall be considered. For this computation, a lateral acceleration of 0.3 G shall be used.

4.1.15 Adjustable Bulkheads—Box Cars

4.1.15.1 Bulkheads, fixtures and attachments, and the car structure shall be designed to withstand the forces specified in paragraphs 4.1.15.1.1 and 4.1.15.1.2 without exceeding the yield or critical buckling stress.

4.1.15.1.1 Longitudinal Force on Cars with Conventional Draft Gears

4.1.15.1.1.1 Bulkheads, Tracks and Supporting Structure, and Car Structure

Force—Uniformly Distributed (lb)			
Nominal Capacity of Car (ton)	Total Force Over Entire Face of Bulkhead	Over Lower Half of Bulkhead	Over Portion of Bulkhead 12 in. from Bottom
70	190,000	152,000	114,000
100 (263K) (286K)	220,000 240,000	176,000 191,000	132,000 144,000

4.1.15.1.1.2 Tracks and Supporting Structure and Brackets for Bulkheads (Car Empty—Bulkhead Not Locked in Place)

Longitudinal force	15 × bulkhead weight
Vertical force	10 × bulkhead weight
Lateral force	10 × bulkhead weight

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4.1.15.1.2 Longitudinal Force on Cars with Special Cushioning Devices

4.1.15.1.2.1 Bulkheads, Tracks and Supporting Structure, and Car Structure

Force—Uniformly Distributed (lb)			
Nominal Capacity of Car (ton)	Total Force Over Entire Face of Bulkhead	Over Lower Half of Bulkhead	Over Portion of Bulkhead 12 in. from Bottom
70	95,000	76,000	57,000
100 (263K) (286K)	110,000 120,000	88,000 96,000	66,000 72,000

4.1.15.1.2.2 Tracks and Supporting Structure, and Brackets for Bulkheads (Car Empty—Bulkhead Not Locked in Place)

Longitudinal force	7.5 × bulkhead weight
Vertical force	5.0 × bulkhead weight
Lateral force	5.0 × bulkhead weight

4.1.16 Lateral and Longitudinal Forces on Fixed Side Wall Fillers—Box Cars

Fixed side wall fillers and related fixtures and attachments shall be designed to withstand the lateral and longitudinal forces indicated below without exceeding the yield or critical buckling stress.

4.1.16.1 Lateral force equivalent to 25% of the weight of lading across the width of the car. The force shall be considered uniformly distributed over the entire face of the filler.

4.1.16.2 Longitudinal force produced by the friction of the lateral force paragraph 4.1.16.1. For this computation, .25 shall be used as the coefficient of friction.

4.1.17 Roping Staples

4.1.17.1 Roping staple, if used, shall be designed to pull six fully loaded cars equipped with roller bearings on tangent track with a 1% grade. For design purposes, the cable load shall be 22,000 lb and the cable shall be considered to be angled 10° horizontally and 10° vertically from roping staple. The maximum stresses shall conform to requirements of paragraph 4.2.2.4.

Note: This is based on the fact that the cable should never be shorter than one half the length of the car (length over pulling faces of couplers) from the roping staple to the car puller located 10 ft from the centerline of track on the ground.

4.1.17.2 The roping staple should provide 3 in. to 4 in. of longitudinal and vertical opening with a smooth transition section to prevent wedging of the hook.

4.1.17.3 The car structure in the roping staple area shall be designed to withstand the forces from paragraph 4.1.17.1. The application should facilitate direct cable access. The preferred location is on the end of the bolster.

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4.2 Allowable Stresses

4.2.1 Letter Symbols for Stress Analysis

A	=	Area
a	=	Length of a plate
b	=	Breadth or width
c	=	Distance from neutral axis to extreme fiber
$d; D$	=	Diameter
e	=	Eccentricity of applied load
E	=	Modulus of elasticity
f	=	Coefficient of friction
G	=	Modulus of elasticity in shear
h	=	Height; depth; thickness
I	=	Moment of inertia (area)
J	=	Polar moment of inertia (area)
k	=	Spring constant
K	=	Stress concentration factor; critical buckling constant; beam stiffness constant
$L; l$	=	Lengths; spans
M	=	Moment of force; bending moment
$M.S.$	=	Margin of safety
γ	=	Poisson Ratio
$n; N$	=	Number, factor of safety

p	=	Pressure, normal force per unit area
$P; F$	=	Concentrated loads
Q	=	Static moment
w	=	Load per unit length
q	=	Shear flow
R	=	Reaction; radius; stress ratio
r	=	Radius; radius of gyration
S	=	Section modulus
σ	=	Normal stress
τ	=	Shear stress
T	=	Torque
t	=	Thickness
V	=	Shearing force in beam section
W	=	Work; total load
ε	=	Normal strain
γ	=	Shear strain; weight per unit volume
y_m	=	Maximum beam deflection
δ	=	Total elongation

Subscripts and superscripts may be used to denote special values of the letter symbols. These are some commonly used subscripts:

t	=	Tension
c	=	Compression
b	=	Bending

br	=	Bearing
y	=	Yield
u	=	Ultimate

4.2.2 Load Factors and Allowable Design Stresses

4.2.2.1 Each member in the car structure shall be investigated for its most critical loading condition. Such critical loading conditions may result from loads applied singly or in combination, provided such combination can rationally occur.

4.2.2.1.1 When submission of stress analysis is required in accordance with the provision of paragraph 1.1, it shall contain complete computations covering the analysis of each important load-carrying member of the car for its respective most critical loading condition only and shall include a summary sheet, tabulating each critical member, its most critical loading condition, the margin of safety for each member, and reference to the page in the analysis where each computation is performed.

4.2.2.1.2 The margin of safety may be computed as follows for the conditions of simple stress:

$$\text{M.S.} = \frac{\text{allowable stress}}{\text{actual stress}} - 1$$

For other conditions, use the interaction formulas of paragraph paragraph 4.2.2.9.

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4.2.2.2 The design loads applied to the structure shall be those loads defined in paragraph 4.1 multiplied by the respective load factors defined below.

4.2.2.3 The allowable design stresses shall be based upon nationally or internationally recognized standards or by similarly recognized test methods to verify performance.

4.2.2.4 For all conditions of dead load (paragraph 4.1.2), live load (paragraph 4.1.3), and buff or draft loads (paragraph 4.1.8) applied singly or in combination, the load factor for each shall be 1.8 and the allowable design stress shall be the yield or 80% of ultimate, whichever is lower, or the critical buckling stress.

4.2.2.5 For all conditions resulting from dead load (paragraph 4.1.2), live load (paragraph 4.1.3) in combination with compressive end load (paragraph 4.1.9), tractor for loading trailers (paragraph 4.1.3.5.1.2), lift truck wheel loads (paragraph 4.1.4), vertical loads on coupler (paragraph 4.1.5), jacking load (paragraph 4.1.6), or roof loads (paragraph 4.1.7), the load factors applied to each load shall be 1.0 and the allowable design stress shall be the yield or 80% of ultimate, whichever is lower, or the critical buckling stress.

4.2.2.6 For all conditions resulting from dead load (paragraph 4.1.2), live load (paragraph 4.1.3) in combination with impact load (paragraph 4.1.10) and vertical forces (paragraph 4.1.11), the load factor applied to the impact load shall be 1.0, and such loading may develop the ultimate load carrying capacity of the member being investigated.

4.2.2.7 Load Factors and Allowable Design Stresses for Lateral Forces

4.2.2.7.1 When the design of a member is governed by lateral forces of paragraphs 4.1.14.1, 4.1.14.2, and 4.1.14.3 singly or in combination with a dead load (paragraph 4.1.2), live load (paragraph 4.1.3), and buff and draft loads (paragraphs 4.1.8), the load factor for the lateral forces shall be 1.5 and for the other loads it shall be 1.8. The allowable design stress shall be the yield or 80% of ultimate, whichever is lower, or the critical buckling stress.

4.2.2.7.2 When the lateral forces of paragraph 4.1.14.1 or paragraph 4.1.14.2 are combined with dead load (paragraph 4.1.2), live load (paragraph 4.1.3) in combination with compressive end load (paragraph 4.1.9), the load factor for the lateral forces and for the other loads shall be 1.0. The allowable design stress shall be the yield or 80% of ultimate, whichever is lower, or the critical buckling stress.

4.2.2.7.3 When the lateral forces of paragraph 4.1.14.1 or paragraph 4.1.14.2 are combined with dead load (paragraph 4.1.2), live load (paragraph 4.1.3), and impact load (paragraph 4.1.10), the load factor for the lateral forces and for the other loads shall be 1.0, and such loading may develop the ultimate load-carrying capacity of the member being investigated.Where stability is not critical, the following allowable stresses shall be used:

σ_y	=	yield stress
σ_t	=	σ_y or .80σ_u whichever is lower
σ_c	=	σ_y or .80σ_u whichever is lower
σ_b	=	σ_y or .80σ_u whichever is lower
σ_{br}	=	1.40 σ_y
τ	=	0.58 σ_y

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4.2.2.8 For rectangular plates, the following expression for buckling may be used:

γ = Poissons ratio = .303 for steel

$$\sigma_{cr} = K\left(\frac{E}{1-\gamma^2}\right)\left(\frac{t}{b}\right)^2$$

Values of K are tabulated below for the more common loading and edge conditions. For buckling of plates of other shapes and loading conditions, refer to standard works on stability and buckling.

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Rectangular plate under equal uniform compression on two opposite edges

Edge Conditions																		
Edge a	Edge b	a/b	0.2	0.3	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0	2.2	2.4
S.S.	S.S	K	22.2	10.9	6.92	4.23	3.45	3.29	3.40	3.68	3.45	3.32	3.29	3.32	3.4
Clamped	Clamped	a/b	1	2	3	∞							2.7	3	∞
		K	7.7	6.7	6.4	5.73							3.32	3.29	3.29
Clamped	S.S.	a/b	0.4	0.3	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.1	∞	
		K	7.76	6.32	5.90	5.76	6.00	6.32	5.8	5.76	6.00	5.80	5.76	5.73	
One edge S.S. One edge free	S.S.	a/b	0.5	1.0	1.2	1.4	1.6	1.8	2.0	2.5	3.0	4.0	5.0		
		K	3.62	1.18	0.934	0.794	0.687	0.622	0.574	0.502	0.464	0.425	0.416		
One edge Clamped One edge free	S.S.	a/b	1	1.1	1.2	1.3	1.4	1.5	1.6	1.7	1.8	1.9	2.0	2.2	2.4
		K	1.40	1.28	1.21	1.16	1.12	1.10	1.09	1.09	1.10	1.12	1.14	1.19	1.21
S.S.	Clamped	a/b	0.6	0.8	1.0	1.2	1.4	1.6	1.7	1.8	2.0	2.5	3.0		
		K	11.0	7.18	5.54	4.80	4.48	4.39	4.39	4.26	3.99	3.72	3.63		

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Rectangular plate under linearly varying stress on edges (b) (bending or bending combined with tension or compression)

Edge a	Edge b	α	a/b	0.4	0.5	0.6	0.667	0.75	0.8	0.9	1.0	1.5
S.S.	S.S.	0.5	K	23.9	21.1	19.8	19.7	19.8	20.1	21.1	21.1	19.8
		0.75	K	15.4		10.6		9.5	9.2		9.1	9.5
		1.00	K	12.4		8.0		6.9	6.7		6.4	6.9
		1.25	K	10.95		6.8		5.8	5.7		5.4	5.8
		1.50	K	8.9		5.3		5.0	4.9		4.8	5.0
		∞	K	6.92		4.25			3.45		3.29	3.57

$$\alpha = \frac{\sigma_c}{\sigma_c - \sigma_\upsilon}$$

Rectangular plate under uniform shear on edges

Edge a	Edge b	a/b	1.0	1.2	1.4	1.5	1.6	1.8	2.0	2.5	3.0	∞				
S.S	S.S	K	7.75	6.58	6.00	5.84	5.76	5.59	5.43	5.18	5.02	4.40				
Clamped	Clamped	a/b	1	2												
		K	12.7	9.5	7.38											

Rectangular plate under uniform shear on all edges and bending stresses on edges b

$\frac{\tau}{\tau_{cr}} = \frac{\text{Actual Shear Stress}}{\text{Critical Shear Stress}}$

If shear acting along
For a/b = 1

Edge a	Edge b	$\frac{\tau}{\tau_{cr}}$	0	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
S.S	S.S.	K	21.1	20.4	19.6	18.5	17.7	16.0	14.0	11.9	8.20	0

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4.2.2.9 For cases of buckling under combined loading conditions not defined in paragraph 4.2.2.8, it may be assumed that the following interaction criteria are valid:

Compression and bending $R_b^{1.75} + R_c = 1$

Compression and shear $R_s^{1.5} + R_c = 1$

Bending and shear $R_b^{1.75} + R_s^{1.75} = 1$

where $R_b = \left(\frac{\sigma}{\sigma_{cr}}\right)_b$ $R_c = \left(\frac{\sigma}{\sigma_{cr}}\right)_c$ and $R_s = \left(\frac{\tau}{\tau_{cr}}\right)_s$ are computed for each condition separately.

The margin of safety for the combined loading may be computed as follows:

4.2.2.9.1 Enter the coordinates R_b and R_c, R_b and R_s, or R_c and R_s on Fig. 4.1.

4.2.2.9.2 Construct a line from the origin through the coordinates defined in the preceding paragraph and extending to the curve defining the appropriate interaction expression.

4.2.2.9.3 Note the coordinates of the intersection defined above, i.e., R_b' and/or R_c' and/or R_s'.

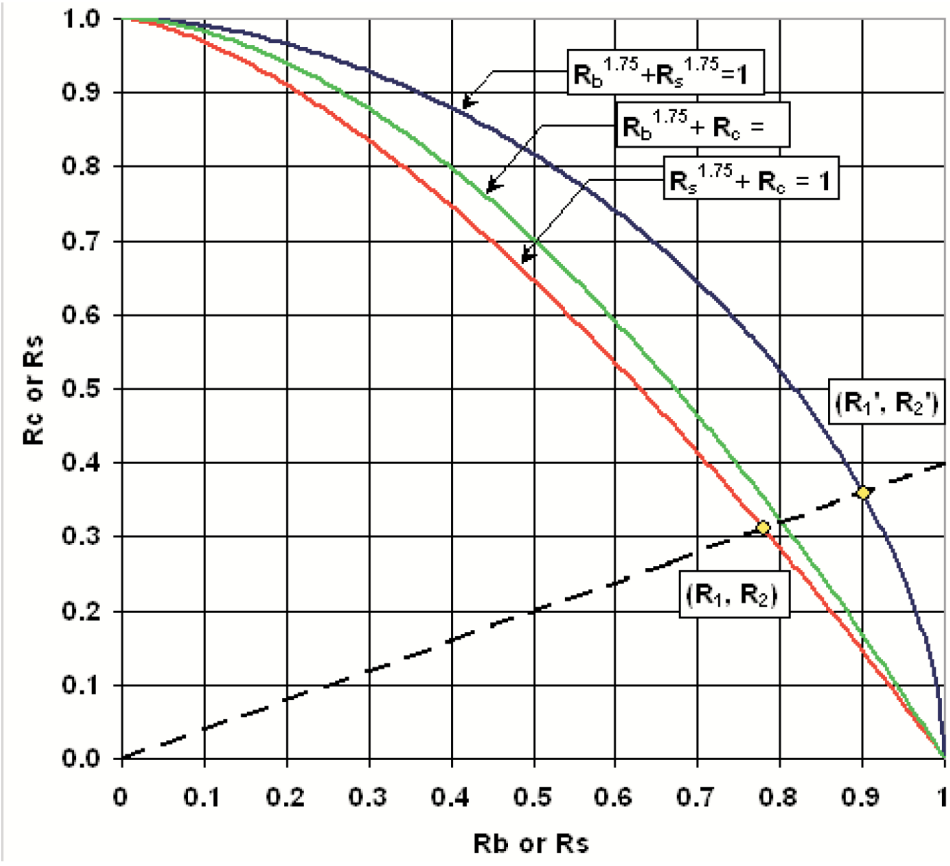


Fig. 4.1 Interaction curves—critical buckling

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4.2.2.9.4 The margin of safety is then defined by:

$$\text{M.S.} = \frac{R_b'}{R_b} - 1 \qquad \text{M.S.} = \frac{R_c'}{R_c} - 1 \qquad \text{or} \qquad \text{M.S.} = \frac{R_s'}{R_s} - 1$$

4.2.2.9.5 It should be noted that if the coordinates referred to in paragraph 4.2.2.9.4 lie above and/or to the right of the appropriate interaction curve, the design is unsafe by inspection.

4.2.2.10 For concentrically loaded columns, the allowable buckling stresses may be computed from the following equations:

long columns: $$\sigma_{cr} = \frac{C\pi^2 E}{\left(\frac{L}{r}\right)^2}$$

short columns: $$\sigma_{cr} = \sigma_1 - \frac{\sigma_1^2}{4C\pi^2 E}\left(\frac{L}{r}\right)^2$$

where the value of C is dependent on the end fixity and may be considered to be

.25 for one end restrained, other end free;
1.0 for both ends simply supported;
2.0 for one end restrained, other end simply supported;
4.0 for both ends restrained;
1.0 both ends restrained, one end free to translate.

σ_1 shall be the material yield or the critical buckling strength of the most critical element of the column cross section.

The short column formula will apply up to L/r values of $\sqrt{\frac{2C\pi^2 E}{\sigma_1}}$

The long column formula will apply for all other values.

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4.2.2.11 Eccentrically Loaded Columns

For columns loaded eccentrically with respect to the major axis, the stresses may be computed from the following:

$$\sigma = \frac{P}{A}\left\{1 + \frac{e_x c_x}{r_x^2}\sec\left[\sqrt{\frac{P}{4EA}}\left(\frac{L'}{r_x}\right)\right] + \frac{e_y c_y}{r_y^2}\sec\left[\sqrt{\frac{P}{4EA}}\left(\frac{L'}{r_y}\right)\right]\right\}$$

where

P = Applied load multiplied by the load factor (See paragraphs 4.2.2.3 to 4.2.2.7.3, inclusive)

e_x, e_y = Eccentricity with respect to the corresponding axis

c_x, c_y = Distance from centroidal axis to most stressed fiber in compression

r_x, r_y = Radius of gyration with respect to corresponding axis

L' = Effective length of column as shown below:

[a/] L'	=	0.75 L	for riveted ends
[a/] L'	=	0.85 L	for pinned ends
L'	=	2 L	for one end free, the other end fixed
L'	=	0.7 L	for one end rounded, the other fixed
L'	=	L	for both ends, rounded
L'	=	0.5 L	for both ends fixed

where L = total length of column

[a/] ASCE Special Committee on Steel Column Research

When eccentricity occurs with respect to one of the major axis only, the expression ec/r that does not apply becomes zero.

In the above equation, σ may not exceed the yield point of the material or the critical compressive buckling stress of any element in the cross section. (See paragraph 4.2.2.8)

4.2.2.12 Axially Loaded Beams

Within the elastic limit, the maximum stress in the extreme fiber of a beam simultaneously subjected to an axial load in addition to transverse loads is expressed by the following:

$$\text{Max } \sigma = \frac{P}{A} \pm \frac{M}{S}$$

where P = the axial load

A = the cross-sectional area of the beam

S = the section modulus of the beam

M = maximum bending moment produced by both axial and transverse loads

For those cases in which the beam deflection resulting from transverse loads is of a low magnitude or when precambering of the car minimizes the eccentricity of the axial load produced by the deflection of the beam, beam column analysis is not required. However, for long and slender members subject to axial compression and transverse loading, not meeting the above criteria, beam column analysis must be performed. Beam column formulas to obtain the maximum bending moments from the combined loading may be used from any engineering reference book.

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4.2.2.13 Plate Girders—Shear Resistant

When a girder web or car side skin acting as a girder web can perform its function without buckling when the girder is subjected to the design loads specified in paragraph 4.2.2.2, the beam shall be considered shear resistant and may be designed in accordance with the following procedure.

4.2.2.13.1 The maximum flange stresses may be computed utilizing the usual flexural formula

$$\sigma = Mc \div I$$ where I may include all web material.

4.2.2.13.2 The maximum shear flow in the web may be computed from the formula

$$q = \frac{VQ}{I}$$ where V = girder maximum shear
Q = first moment of the entire cross-sectional area above the neutral axis
I = as defined in paragraph 4.2.2.13.1

The maximum shear stress is then $\tau = q \div t$.

4.2.2.13.3 The web buckling stress may be computed by the procedure of paragraphs 4.2.2.8 and 4.2.2.9 for the most critical condition for web buckling, i.e., shear or bending or a combination of shear and bending for the conditions specified in paragraph 4.1.2.

4.2.2.13.4 The web stiffeners shall be designed for the loads specified in paragraph 4.2.2.7.1, but in no case shall the stiffener moment of inertia be smaller than

$$\frac{2.29d}{t}\left(\frac{Vh}{33E}\right)^{4/3}$$ where d = stiffener spacing
h = web height

4.2.2.13.5 Flange-to-web connection must be capable of resisting shear flows of a magnitude equal to $\frac{VQ}{I}$ where Q is the first moment of the flange only about the neutral axis of the entire girder. This shear flow may be compared directly with weld allowables in pounds per inch or may be converted to load per rivet by multiplying by the rivet spacing.

4.2.2.13.6 Web splices shall be designed to carry the shear and moment across the joint. The vertical component of the joint load may be computed by the method described in paragraph 4.2.2.13.5 using a value of Q consistent with the joint extremities. The horizontal component may be computed from the maximum bending stress in the web at the joint extremity. The resultant load on the joint is obtained by vector addition of the above components.

4.2.2.13.7 When multiple piece flanges or reinforcements are used, the attachment of these components may be investigated by the method of paragraph 4.2.2.13.5, except that Q shall be the first moment of the cross-sectional area beyond the joint about the neutral axis.

4.2.2.14 Plate Girders—Partial Tension Field

When a girder web or car side skin acting as a girder web buckles at loads smaller than the design loads specified in paragraph 4.2.2.2, the beam shall be considered a semi-tension field and may be designed in accordance with the following procedure.

4.2.2.14.1 Compute the shear load carried by web at the web buckling stress by

$$V_{cr} = \tau_{cr} ht$$ where τ_{cr} = the shear buckling stress; may be computed from paragraph 4.2.2.8 using a value of k that is midway between simply supported and clamped edge conditions

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4.2.2.14.2 The shear load resistance developed by the web at the web critical buckling stress is then

$$V_{cr} = \tau_{cr} ht$$

4.2.2.14.3 Compute the web tension field angle from

$$\sin^2\alpha = \sqrt{a^2 + a} - a \qquad \text{where} \qquad a = \frac{1 + \dfrac{ht}{A_u + A_1}}{\dfrac{dt}{A_s} - \dfrac{ht}{A_u + A_1}}$$

A_u, A_1, A_s = areas of the upper and lower chords and the stiffener respectively neglecting any effective web areas

h = centroid to centroid distance between chords

d = an analogous distance between web stiffeners

t = web thickness

4.2.2.14.4 Compute the value of

$$wd = 1.25 d \sin\alpha \sqrt[4]{\frac{t}{(I_u + I_l)h}}$$ where I_u and I_l = moments of inertia of the upper and lower chords respectively about an axis perpendicular to the plane of the web

Obtain R from the table on the right using straight line interpolation for intermediate values.

wd	R
0	1.00
1	.99
2	.93
3	.72
4	.52

wd	R
5	.40
6	.32
7	.27
8	.23

4.2.2.14.5 For riveted construction obtain

$$K_r = \frac{\text{rivet spacing} - \text{rivet diameter}}{\text{rivet spacing}}$$

For welded construction, K is equal to 1.0.

4.2.2.14.6 Compute the shear strength of the web acting as a tension field when stressed to the yield point in tension from

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4.2.2.14.7 The total web shear resistance to yield is

$$V_y = \frac{I}{Q_f h}(V_{cr} + V_{ty})$$ where I = girder moment of inertia not including web material

Q_f = first moment of the flange only. The web margin of safety may be computed from

$$\text{M.S.} = \frac{V_y}{V} - 1$$

4.2.2.14.8 Web to flange attachment loads in pounds per inch of flange may be computed from

$$P_r = \left[\left(\frac{V_{cr}}{h}\frac{I_f}{I} + \frac{V_{ty}}{h}\right)^2 + \left(\frac{V_{ty}}{h}\tan\alpha\right)^2\right]^{1/2}$$

where I_f is the moment of inertia computed for flanges only. Rivet loads on a per rivet basis may be computed by multiplying above value of P_r by the rivet spacing.

4.2.2.14.9 When computing flange stresses it may be assumed that the entire web is effective up to its critical buckling stress in the section under investigation. Post buckling strength shall be computed utilizing flange areas only. Flange primary stresses may be computed from

$$\sigma = \pm\frac{M_{cr}Y}{I} \pm \frac{(M - M_{cr})Y}{I_f}$$ where M_{cr} = bending moment for the load that causes buckling of web

M = total bending moment on section

Y = c

4.2.2.14.10 The diagonal tension field produces secondary bending stresses in the flanges by inducing loads normal to the flange. This secondary bending moment may be computed from

$$M = \frac{C}{12}\frac{V_t}{h}d^2\tan\alpha$$ where V_T = that component of the shear due to tension field action

The value of C may be taken from the table on the right using straight line interpolation between values.

wh	C
0	1.00
1	1.00
2	.96
3	.87
4	.73

wh	C
5	.60
6	.49
7	.41
8	.35

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4.2.2.14.11 Compute stiffener load from

$$F = (-V_t)\frac{d}{h}\tan\alpha$$

Obtain a value of C_s from the table on the right using straight line interpolation between values.

Check stiffener as pin ended column with effective length equal to $C_s \times L$.

$\frac{d}{h \tan\alpha}$	C_s	$\frac{d}{h \tan\alpha}$	C_s
0	.38	1.2	.77
.2	.39	1.4	.82
.4	.42	1.6	.85
.6	.50	1.8	.88
.8	.63	2.0	.91
1.0	.72		

4.3 General Details

4.3.1 Gross and Net Sections

4.3.1.1 Definitions

The gross section of a member at any point shall be determined by adding the products of the thickness and the gross width of each element as measured normal to the axis of the member. The net section shall be determined by substituting for the gross width the net width computed in accordance with paragraphs 4.3.1.2, 4.3.1.3, and 4.3.1.4.

4.3.1.2 Net Width

4.3.1.2.1 When a part has one or more holes on a line perpendicular to the direction of stress, the net width shall be determined by subtracting the sum of the diameters of the holes from the gross width.

4.3.1.2.2 In the case of a chain of holes across a part in any diagonal or zigzag line, when not more than one hole is on a line perpendicular to direction of stress, the net width of the part shall be obtained by deducting from the gross width the sum of the diameters of all the holes in the chain, and adding for each gauge space in the chain, the quantity

$$\frac{S^2}{4g}$$

where S = longitudinal spacing (pitch)of any two successive holes (in.)
g = transverse spacing (gauge) of the same two holes (in.)

4.3.1.3 Size of Holes

In computing net area, the diameter of a hole shall be taken as $\frac{1}{16}$ in. greater than the nominal diameter of a rivet or bolt.

4.3.1.4 Angles

For angles, the gross width shall be the sum of the widths of the legs less the thickness. For determining net width (paragraph 4.3.1.2), the gauge for holes in opposite legs shall be the sum of the gauges from the back of the angle less the thickness.

4.3.1.5 Splice Members

For these members, there shall be sufficient rivets or welds on each side of the splice considered to develop the strength of the member.

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4.3.1.6 Application

- Tension members shall be designed on the basis of net section.
- Compression members shall be designed on the basis of gross section.

4.3.2 Rivets and Bolts

4.3.2.1 Diameter

In proportioning and spacing rivets, the nominal diameter of the undriven rivet shall be used.

4.3.2.2 Effective Bearing Area

The effective bearing area of rivets and bolts shall be the nominal diameter multiplied by the length in bearing; except that for countersunk rivets and bolts half of the depth of the countersink shall be deducted.

4.3.2.3 Double- and Single-Shear Bearing

Only that portion of a rivet or bolt shall be considered in double-shear bearing that lies between two portions that share the reaction therefrom. The remainder of the rivet or bolt shall be considered in single-shear bearing.

4.3.2.4 Maximum Grip of Rivets and Bolts

The maximum grip of rivets and bolts shall not exceed five diameters of the rivet.

4.3.3 Spacing of Rivets

4.3.3.1 Minimum Pitch

The minimum distance between centers of rivet holes shall be not less than three times the nominal diameter of the rivet; preferably 3 in. for $\frac{7}{8}$-in. rivets, 2 $\frac{1}{2}$ in. for $\frac{3}{4}$-in. rivets, and 2 in. for $\frac{5}{8}$-in. rivets.

4.3.3.2 Maximum Pitch

The maximum pitch in the line of stress of compression members composed of plates and shapes shall not exceed

$$1.814t\sqrt{\frac{E}{\sigma_y}}$$

where t is the thickness of the thinnest outside plate or shape. Rivet spacing for tank anchors shall be governed by the *MSRP*, Section C-I, Specification M-1002, Chapter 5.

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4.3.3.3 Minimum Edge Distance

The minimum distance from the center of a rivet or bolt hole to any edge shall be that given in Table 4.1.

Table 4.1 Minimum edge distance

Rivet or Bolt Diameter (in.)	Minimum Edge Distance for Punched, Reamed, or Drilled Holes (in.)	
	At Sheared Edges	At Rolled Edges of Plates, Shapes, or Bars or Gas-Cut Edges[a/]
3/8	3/4	5/8
1/2	7/8	3/4
5/8	1 1/8	7/8
3/4	1 1/4	1
7/8	1 1/2	1 1/8
1	1 3/4	1 1/4
1 1/8	2	1 1/2
1 1/4	2 1/4	1 5/8

[a/] All edge distances in this column may be reduced 1/8 inch when the hole is at a point where stress does not exceed 25% of the maximum allowed stress in the element.

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4.3.3.4 Rivet Heads

Fig. 4.2 lists the acceptable limits for rivet heads.

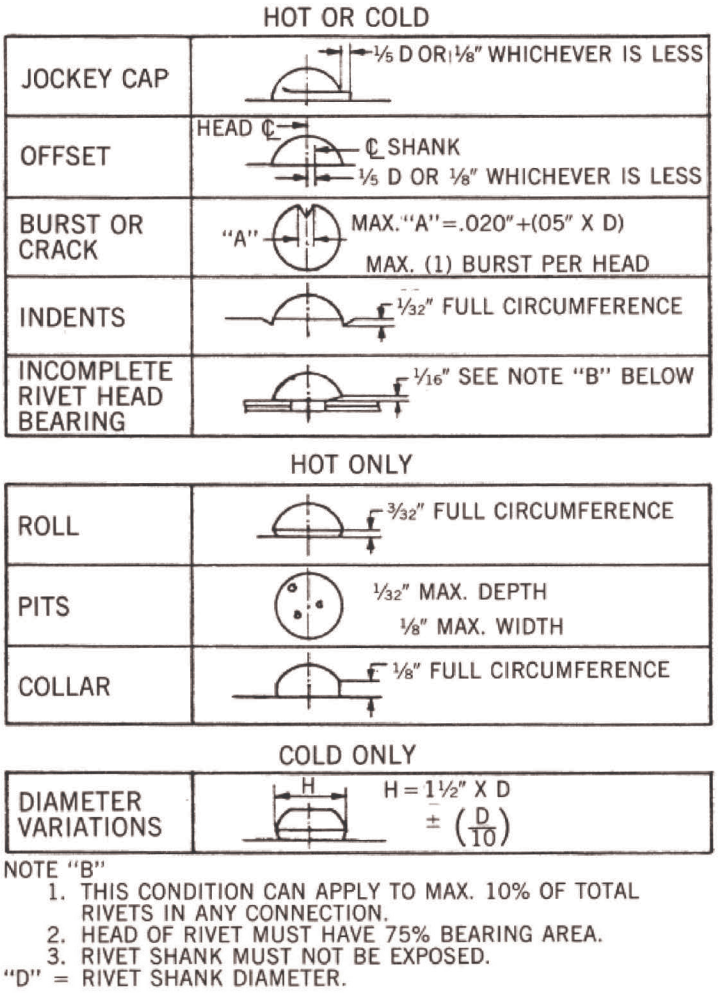


Fig. 4.2 Acceptable limits for rivet heads

4.3.3.5 Maximum Edge Distance

The maximum distance from the center of any rivet or bolt to the near edge shall be 12 times the thickness of the plate, but shall not exceed 6 in.

4.3.3.6 Allowable Bolt Stresses

The allowable design stresses for ASTM A325 and A490 bolts, limited to two-member connections, shall be those given in Table 4.2. The appropriate load factor as specified in paragraph 4.2.2 shall be applied to the loads and forces considered in the joint analysis. Tension connections shall consider prying effects. Shear connections subjected to load reversals or severe load fluctuations shall be designed as friction-type joints.

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Table 4.2 Allowable bolt stresses (ksi)

	ASTM A325 <1 1/8	ASTM A325 ≥1 1/8	ASTM A490
Ultimate tensile stress (see Notes 1 and 5)	120	105	150
Allowable tensile stress (see Notes 1 and 5)	92	81	127
Ultimate shear stress (see Notes 2 and 5)	72	63	90
Allowable shear stress (bearing—see Notes 2 and 5)	54	47	67
Allowable shear stress—friction (see Notes 1, 3, 4, and 5)	$95K_s$	$83K_s$	$113K_s$
Allowable bearing stress in the connected material (see Note 6)	$0.85\frac{L}{d}\sigma_u$ or $2.5\sigma_u$ whichever is less		

NOTES:
1. STRESSES ARE COMPUTED ON THE AREA OF THE BOLT THROUGH THE THREADS. ALLOWABLE TENSILE STRESS IS THE LESSER OF YIELD STRESSES OR .85 OF THE ULTIMATE STRESS.
2. STRESSES ARE COMPUTED ON THE AREA OF THE BOLT IN THE SHEAR PLANE.
3. K_s IS THE SLIP COEFFICIENT. USE .26 FOR CLEAR MILL SCALE SURFACE CONDITIONS.
4. THE FASTENER TENSION SHALL EQUAL OR EXCEED 70% OF THE SPECIFIED MINIMUM TENSILE STRENGTH OF THE BOLT. TORQUE CONTROL OR THE USE OF DIRECT TENSION INDICATORS ARE ACCEPTABLE METHODS FOR DEVELOPING THE REQUIRED TENSION. TURN-OF-THE-NUT METHOD IS ACCEPTABLE WHEN APPLIED TO A SNUG-TIGHT JOINT, WHEN SNUG-TIGHT IS DEFINED AS THE POINT AT WHICH ALL ASSEMBLY MEMBERS ARE IN CONTACT DIRECTLY UNDER THE FASTENER.
5. NOMINAL HOLE SIZE SHALL NOT EXCEED + 1/16 OF THE NOMINAL BOLT DIAMETER UNLESS THE STRESSES ARE REDUCED AS FOLLOWS: 1) FOR SHORT-SLOTTED HOLES AS DEFINED IN AISC SPECIFICATION FOR STRUCTURAL JOINTS, ASTM A-325 OR A490 BOLTS, USE 85%; 2) FOR LONG-SLOTTED HOLES, USE 70%.
6. *L* = DISTANCE FROM CENTER OF HOLE TO THE NEAREST EDGE IN THE LINE OF STRESS.
d = DIAMETER OF BOLT.

4.3.4 Welds

4.3.4.1 Effective Areas of Weld Metal

The effective area of all welds shall be as defined in AWS D15.1.

4.3.4.1.1 Allowable Weld Stresses

The allowable design stress for fillet welds shall be the minimum yield strength of the electrode or wire multiplied by 0.58. Table 4.3 shows the allowable design stresses for specific electrodes and matching base metals. The appropriate load factors as specified in paragraph 4.2.2 shall be applied to the loads and forces considered in the joint analysis.

See MSRP, Section C-III, Specification M-1002, Appendix E, for allowable weld stresses for tank cars.

4.3.4.1.2 For connections subject to fatigue loading, see Chapter 7.

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4.3.4.1.3 Allowable Design Stresses in Welds

Table 4.3 Allowable design stresses in welds

Kind of Stress	Allowable Design Stress	Metal and Filler Requirements
Tension and compression parallel to the axis of any complete-penetration groove weld	Same as for base metal	Per AWS D15.1, Table 8.1
Tension normal to the axis of complete-penetration groove weld	Same as allowable tensile stress for base metal	
Compression normal to the axis of complete- or partial-penetration groove weld	Same as allowable compressive stress for base metal	
Shear on effective throat of complete-penetration groove weld and partial-penetration groove weld	Same as allowable shear stress for base metal	
Shear stress on effective[a/] throat of fillet weld regardless of direction of application of load; tension normal[b/] to the axis on the effective throat of a partial-penetration groove weld; and shear stress on the effective area of a plug or slot weld. The given stresses shall also apply to such welds made with the specified electrode on steel having a yield stress greater than that of the "matching" base metal. The allowable stress, regardless of electrode classification used, shall not exceed that given in the table for the weaker "matching" base metal being joined.	29.0 ksi	Per AWS D15.1, Table 8.1 Class I
	33.06 ksi	AWS D15.1, Table 8.1 Class II
	38.86 ksi	AWS D15.1, Table 8.1 Class III
	50.46 ksi	AWS D15.1, Table 8.1 Class IV
	56.84 ksi	AWS D15.1, Table 8.1 Class V

[a/] For definition of effective throat of fillet welds and partial-penetration groove welds, see paragraph 4.3.4.1.

[b/] Fillet welds and partial-penetration groove welds joining the component elements of built-up members, such as flange-to-web connections, may be designed without regard to the tension or compression stress in these elements parallel to the axis of these welds.

4.3.4.1.4 Qualification of Weld Joints and Joint-Welding Procedure

Welding joints and joint-welding procedures for complete and partial-penetration welds, fillet welds, plug welds, and slot welds meeting AWS D15.1 (latest revision), Sections B and C, are considered prequalified.

Other joints and joint-welding procedures may be used provided they have been qualified in accordance with AWS D15.1.

4.3.4.1.5 Shielded Metal, Submerged Arc, Gas Metal, Flux-Cored Arc Welding Filler Metal, and Technique

Welding electrodes, filler metal requirements, and technique used shall meet the applicable requirements of AWS D15.1.

4.3.5 Connections

4.3.5.1 Strength of Connection

All connections shall be of sufficient strength to transmit the receive forces induced by the design loads.

4.3.5.2 Eccentric Connections

Eccentric connections shall be avoided if practicable, but if unavoidable, provision shall be made for the effect of the eccentricity.

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4.3.5.3 Fillers

The use of fillers in connections shall be avoided wherever practicable.

4.3.5.4 Rivets and Bolts in Combination with Welds

Rivets or bolts used in bearing-type connections shall not be considered as sharing the stress in combination with welds; the welds shall be considered to carry the full load.

4.3.6 Electrical Bonding—Car Grounding

The design of the car shall provide for the bonding of all parts so as to ensure an electrical ground through the car wheels. Where electrical isolation is possible, a bonding wire or suitable equivalent shall be provided for in the design of the car or component.

4.4 Underframe Members

4.4.1 General

The following details of design cover the principal underframe load-carrying members of conventional design cars.

All underframe members shall be designed to meet the requirements of paragraphs 4.1, 4.2, and 4.3. The specific conditions of loading stipulated hereafter are those that usually govern the design of the members described; however, other conditions of loading may be more critical for specific car configurations. The condition of loading stipulated hereafter shall not be interpreted as relieving the designer of the responsibility of meeting all the requirements of paragraphs 4.1, 4.2, and 4.3.

The methods of analysis shown are considered acceptable, but other rational methods of analysis are not precluded.

4.4.2 Decking or Flooring

Decking or flooring may be AAR standard wood decking boards, metal, laminated wood panels, composite wood-metal, or other proprietary types and shall be capable of sustaining the loads specified herein without exceeding the allowable unit stresses.

When using composite wood-metal, laminated wood panels, or other proprietary types of flooring, the manufacturer of such flooring shall submit either test data or stress analysis to confirm the capability of the flooring to meet the strength requirements. (Please reference the *MSRP*, Section C-I, Specification M-964 for nailable steel floors, and Specification M-908 for flat steel floors.)

When an untried material is used in the flooring, data shall be submitted to demonstrate its suitability for the intended usage.

4.4.2.1 Box Cars and General Service Flatcars

Flooring for box cars and general service flatcars shall be designed for lift truck wheel loads (paragraph 4.1.4) as applicable with maximum stresses conforming to the requirements of paragraph 4.2.2.5. The wheel loads shall be so located as to obtain critical loading.

4.4.2.2 Gondola Cars

Flooring for gondola cars shall be designed for a uniformly distributed load obtained from 87% live load distributed over the full width of car and over a length of 18 ft at center of car (paragraph 4.1.3.3.b) with maximum stresses conforming to the requirements of paragraph 4.2.2.4.

4.4.2.3 Hopper Cars

Flooring for hopper cars shall be designed for a car loaded to capacity with its load distributed according to the interior configuration of the car with maximum stresses conforming to the requirements of paragraph 4.2.2.4.

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4.4.2.4 Trailer Transport Flatcar

For cars designed for roll-on/roll-off service, the car deck shall be designed to sustain the loads imposed by tires of trailers as specified in paragraph 4.1.3.5, with stresses conforming to the requirements of paragraph 4.2.2.5; and loads imposed by the tires of tractors used in loadings in paragraph 4.1.3.5.1.2, with stresses conforming to the requirements of paragraph 4.2.2.5.

4.4.2.5 Center Beam Car

Transverse risers shall be designed for the load distribution shown in paragraph 4.1.3.4 with maximum stress conforming to the requirements of paragraph 4.2.2.4. Cars without risers shall be designed to support the load distributions on transversely oriented wooden 2 × 4s running the half width of car from center sill to side sill. These 2 × 4s shall be spaced uniformly between the load and the floor at distances not exceeding 8 ft.

4.4.3 Floor Stringers

4.4.3.1 Box Cars

The floor stringers shall be designed for the following loads:

4.4.3.1.1 Dead load combined with lift truck wheel loads (paragraph 4.1.4 as applicable), with maximum stresses conforming to the requirements of paragraph 4.2.2.5. The wheel loads shall be so located as to obtain critical loading.

4.4.3.1.2 Dead load and a uniformly distributed live load in accordance with paragraph 4.1.3.1(e), combined with the critical end load with maximum stresses conforming to the requirements of paragraph 4.2.2.5 or paragraph 4.2.2.6, as applicable. The portion of the live load on each stringer may be computed by the panel area method. The critical value of the end load shall be determined from paragraph 4.1.9.5 or paragraph 4.1.10.3.

4.4.3.2 Flatcars—General Service

The floor stringers shall be designed for the following loads:

4.4.3.2.1 Dead load combined with lift truck wheel loads (paragraph 4.1.4 as applicable), with maximum stresses conforming to the requirements of paragraph 4.2.2.5.

4.4.3.2.2 Dead load and a uniformly distributed live load in accordance with paragraph 4.1.3.2(c), combined with the critical end load with maximum stresses conforming to the requirements of paragraph 4.2.2.5 or paragraph 4.2.2.6, as applicable. The portion of the live load on each stringer may be computed by the panel area method. The critical value of the end load shall be determined from paragraph 4.1.9.5 or paragraph 4.1.10.3.

4.4.3.3 Gondola Cars

The floor stringers shall be designed for the following loads:

4.4.3.3.1 Dead load and a uniformly distributed live load in accordance with paragraph 4.1.3.3(b), with maximum stresses conforming to the requirements of paragraph 4.2.2.4. The portion of the live load on each stringer may be computed by the panel area method.

4.4.3.3.2 Dead load and a uniformly distributed live load in accordance with paragraph 4.1.3.3(b), combined with the critical end load with maximum stresses conforming to the requirements of paragraph 4.2.2.5 or paragraph 4.2.2.6, as applicable. The portion of the live load on each stringer may be computed by the panel area method. The maximum value of the end load shall be computed by paragraph 4.1.9.5 or paragraph 4.1.10.3

4.4.4 Crossties

4.4.4.1 Box and Flatcars

The crossties shall be designed for the following loads and load combinations:

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4.4.4.1.1 Dead load combined with the applicable lift truck wheel loads (paragraph 4.1.4), with maximum stresses conforming to the requirements of paragraph 4.2.2.5. The wheel loads shall be so located as to obtain critical loading.

4.4.4.1.2 Dead load and critical live load combined with vertical acceleration forces induced by the horizontal impact load. The maximum stresses shall conform to the requirements of paragraph 4.2.2.6.

4.4.4.2 Gondola Cars

The crossties shall be designed for the following loads and load combinations:

4.4.4.2.1 For cars with stringers between the center sill and side sills, the crosstie loads shall be considered as concentrated loads from the stringer reactions from the dead load; and live load determined from the requirements of paragraph 4.4.3.3.1. The maximum stresses shall conform to the requirements of paragraph 4.2.2.4.

4.4.4.2.2 For cars without stringers between the center sill and side sills, the crosstie load shall be considered as a uniformly distributed load and may be computed by the panel area method from the dead load and the live load distribution of paragraph 4.1.3.3(b). The maximum stresses shall conform to the requirements of paragraph 4.2.2.4.

4.4.4.2.3 The loads specified above shall be combined with vertical acceleration forces induced by the horizontal impact load. The maximum stresses shall conform to the requirements of paragraph 4.2.2.6.

4.4.5 Crossbearers

4.4.5.1 Box Cars

The crossbearers shall be designed for the following loads and load combinations:

4.4.5.1.1 Reactions from the center sill and floor stringers from the dead load and the critical uniformly distributed live load. The maximum stresses shall conform to the requirements of paragraph 4.2.2.4.

4.4.5.1.2 Reactions from the center sill and floor stringers from the dead load combined with the applicable lift truck wheel loads from paragraph 4.1.4, with the centerline of the axle on the centerline of the crossbearer, and with the center of the truck so located on the crossbearer as to obtain critical loading. The maximum stresses shall conform to the requirements of paragraph 4.2.2.5.

4.4.5.1.3 The loads specified in paragraph 4.4.5.1.1 combined with vertical acceleration forces induced by the horizontal impact load. The maximum stresses shall conform to the requirements of paragraph 4.2.2.6.

4.4.5.2 Flatcars

The crossbearers shall be designed for the following loads and load combinations:

4.4.5.2.1 Crossbearer shall be loaded with a concentrated load at center equal to 75% of load limit multiplied by the percentage of load carried by both side sills. The maximum stresses shall conform to the requirements of paragraph 4.2.2.4. Crossbearers for flatcars used in trailer/container or auto rack service are excluded from the requirements of this section.

4.4.5.2.2 Crossbearer shall be loaded with a concentrated load on each side sill equal to 75% of the load limit multiplied by one half the percentage of the load carried by the center sill. The maximum stresses shall conform to the requirements of paragraph 4.2.2.4. Crossbearers for flatcars used in trailer/container or auto rack service are excluded from the requirements of this section.

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4.4.5.2.3 The loads specified above combined with vertical acceleration forces induced by the horizontal impact load. The maximum stresses shall conform to the requirements of paragraph 4.2.2.6.

4.4.5.2.4 Crossbearer for flatcars in trailer/container or auto rack service must be designed for the loadings of paragraph 4.1, applicable to this service, with stresses conforming to paragraph 4.2.

4.4.5.3 Gondola Car

4.4.5.3.1 The crossbearer shall be designed for the dead load, and the reactions from a load applied on bearing piece across full width of car which is equal to 75% of the load limit, paragraph 4.1.3.3 diagram (d). The maximum stresses shall conform to the requirements of paragraph 4.2.2.4.

4.4.5.3.2 The crossbearer shall also be designed for the loading produced by the critical lateral force (paragraph 4.1.14.1 or paragraph 4.1.14.2). Stresses induced by these loadings are to be combined with stresses induced by the vertical loads and shall conform to the requirements of paragraph 4.2.2.7.1.

4.4.5.3.3 The crossbearer also shall be designed for the loads specified in paragraph 4.4.5.3.1 combined with vertical acceleration forces induced by the horizontal impact load. The maximum stresses shall conform to the requirements of paragraph 4.2.2.6.

4.4.5.4 Hopper Cars

4.4.5.4.1 The crossbearer (crossridge) shall be designed for the dead load and live load center sill reactions and that increment of these loads carried directly by the center sill and for that portion of the load carried directly by the crossbearer. Distribution of the load shall be in accordance with the configuration of the car. The maximum stresses shall conform to the requirements of paragraph 4.2.2.4.

4.4.5.4.2 The crossbearer also shall be designed for the lateral load (paragraph 4.1.14.1). Stresses induced by these forces are to be combined with the stresses induced by the vertical loads. The maximum combined stresses shall conform to the requirements of paragraph 4.2.2.7.1.

4.4.5.4.3 The crossbearer shall also be designed for the loads specified in paragraph 4.4.5.4.1 combined with vertical acceleration forces induced by the horizontal impact load. The maximum stresses shall conform to the requirements of paragraph 4.2.2.6.

4.4.6 Center Sill on Cars with Conventional Underframe

4.4.6.1 The center sill may be considered as a continuous beam supported by underframe members. The distribution of the loads on the center must be considered for all affected car structure. The combined bending moments in the center sill induced by the vertical dead and live loads, and the eccentric application of the end loads, shall be considered in the design analysis.

4.4.6.2 The center sill shall be designed for the following loads:

4.4.6.2.1 Dead load (paragraph 4.1.2), live load (paragraph 4.1.3) combined with buff or draft loads (paragraph 4.1.8) and shall conform to the requirements of paragraph 4.2.2.4.

4.4.6.2.2 Dead load (paragraph 4.1.2), live load (paragraph 4.1.3) combined with compressive end load (paragraph 4.1.9) and shall conform to the requirements of paragraph 4.2.2.5.

4.4.6.2.3 Dead load (paragraph 4.1.2), live load (paragraph 4.1.3 combined with impact load (paragraph 4.1.10) and vertical forces (paragraph 4.1.11) and shall conform to the requirements of paragraph 4.2.2.6. The combined bending moments in the center sill induced by the vertical dead and live loads, and the eccentric application of the end loads, shall be considered in the design analysis.

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4.4.7 Cushioned Underframe Cars

4.4.7.1 The fixed center sill shall be designed for dead load (paragraph 4.1.2), live load (paragraph 4.1.3) combined with buff or draft loads (paragraph 4.1.8) and shall conform to the requirements of paragraph 4.2.2.4.

4.4.7.2 The sliding center sill shall be designed for dead load (paragraph 4.1.2), live load (paragraph 4.1.3) combined with compressive end load (paragraph 4.1.9) and shall conform to the requirements of paragraph 4.2.2.5.

4.4.7.3 The fixed center sill shall be designed for dead load (paragraph 4.1.2), live load (paragraph 4.1.3) combined with impact load (paragraph 4.1.10.2) and vertical forces (paragraph 4.1.11 and shall conform to the requirements of paragraph 4.2.2.6. The combined bending moments in the center sill induced by the vertical dead and live loads, and the eccentric application of the end loads, shall be considered in the design analysis.

4.4.8 Body Bolster

The body bolsters shall be designed for the following loading conditions:

4.4.8.1 Dead load and live load with the car loaded to full capacity and, where applicable, 390,000-lb draft load or 1,000,000-lb compressive end load. When the dead and live loads are applied singly or in combination with the draft or compressive load, the maximum stresses shall conform to the requirements of paragraph 4.2.2.4. When the dead and live loads are applied in combination with the compressive end load, the maximum stresses shall conform to the requirements of paragraph 4.2.2.5.

4.4.8.2 Car loaded to full capacity with car body on jacks under ends of bolsters. The maximum stresses shall conform to the requirements of paragraph 4.2.2.5.

4.4.8.3 Car loaded to full capacity and subjected to a single-ended impact load (paragraph 4.1.10). The maximum stresses shall conform to the requirements of paragraph 4.2.2.6.

4.4.8.4 Car loaded to full capacity under bouncing and rocking conditions. The maximum stresses shall not exceed the stress limitations for yield. The center plate and side bearing loads are as follows:

		100 ton		
	70 ton	263K	286K	125 ton
Center plate load	230,000 lb	275,000 lb	299,000	325,000 lb
Side bearing load	144,000 lb	172,500 lb	187,500	207,000 lb

The center plate loads shall be applied at the following locations:

4.4.8.4.1 Uniformly distributed over the bearing surface of the center plate

4.4.8.4.2 Edge-of-bowl loading

	Diameter of Bowl (in.)	Distance Off Center (in.)
70 ton	13 3/4	6
100 and 125 ton	15 3/4	7

4.4.8.4.3 The side bearing load shall be applied on one side bearing.

4.4.9 Body Center Plates

Body center plate attachment to be subjected to a longitudinal force of 200,000 lb with a load factor of 1.0 and the allowable stresses not to exceed the yield.

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4.5 Superstructure Members

4.5.1 General

The following details of design cover the principal load-carrying members of conventional design cars.

All superstructure members shall be designed to meet the requirements of paragraphs 4.1, 4.2, and 4.3. The specific conditions of loading stipulated herein are those that usually govern the design of the members described; however, other conditions of loading may be more critical for specific car configurations. The conditions of loading stipulated herein shall not be interpreted as relieving the designer of the responsibility of meeting all the requirements of paragraphs 4.1, 4.2, and 4.3.

The methods of analysis shown are considered acceptable, but other rational methods of analysis are not precluded.

4.5.2 Roof

The roof shall be capable of withstanding a uniformly distributed load of 15 lb/ft^2 of projected area plus a concentrated load of 300 lb and shall conform to the requirements of paragraph 4.2.2.5.

4.5.3 Ends

The ends shall be capable of withstanding the longitudinal forces of paragraph 4.1.12.

4.5.4 Sides, Box Cars—Girder Type

The sides shall be designed as girders with overhanging ends supported at the bolsters for the following loads and load combinations:

4.5.4.1 Dead load and critical live load with maximum stresses in accordance with paragraph 4.2.2.4.

4.5.4.2 Dead load, critical live load, and the applicable lift truck wheel loads (paragraph 4.1.4) with maximum stresses in accordance with paragraph 4.2.2.5.

4.5.4.3 Dead load and critical live load combined with the compressive end load with maximum stresses in accordance with paragraph 4.2.2.5.

4.5.4.4 Loading Diagrams

Fig. 4.3 shows a loading diagram for a 50-ft car. The live load shown is the critical live load for a 50-ft car having 5 ft 5 ½ in. overhang from center of bolsters to striker face and side door openings up to 16 ft. For longer cars, or cars with longer overhang, or wider door openings, other live load distributions should be investigated to determine the critical live load.

Fig. 4.4 shows the location of the lift truck to be combined with the live load (Fig. 4.3) for computing stresses (paragraph 4.5.4.2).

4.5.4.5 Effects of Shear

In the following analysis, all connections between door posts and top and bottom chords are considered as fixed.

Fig. 4.5 shows the effects of vertical shear on the upper and lower chords. The conditions are for a car loaded as shown in Fig. 4.3 and Fig. 4.4. When the live load is placed in the opposite end of the car, the effect of the shear would be the reverse to that shown.

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4.5.4.6 Loading Diagrams

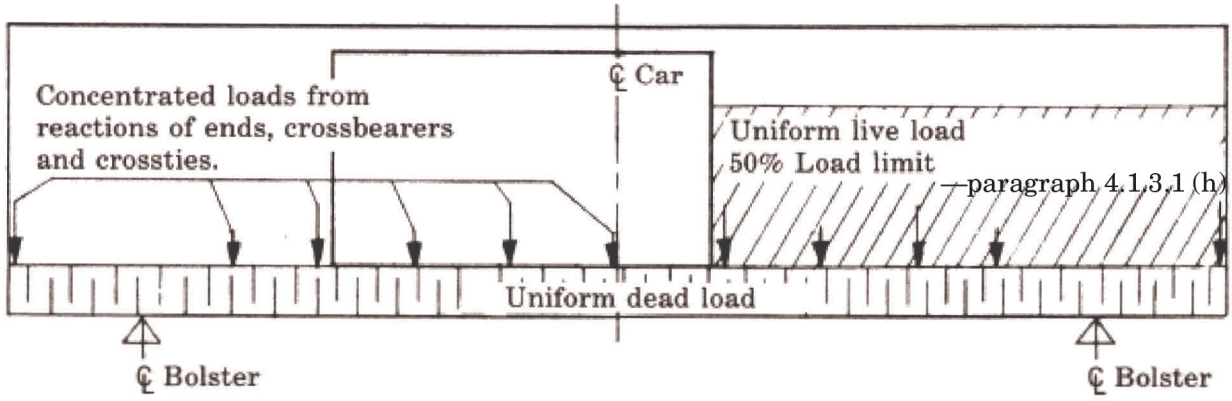
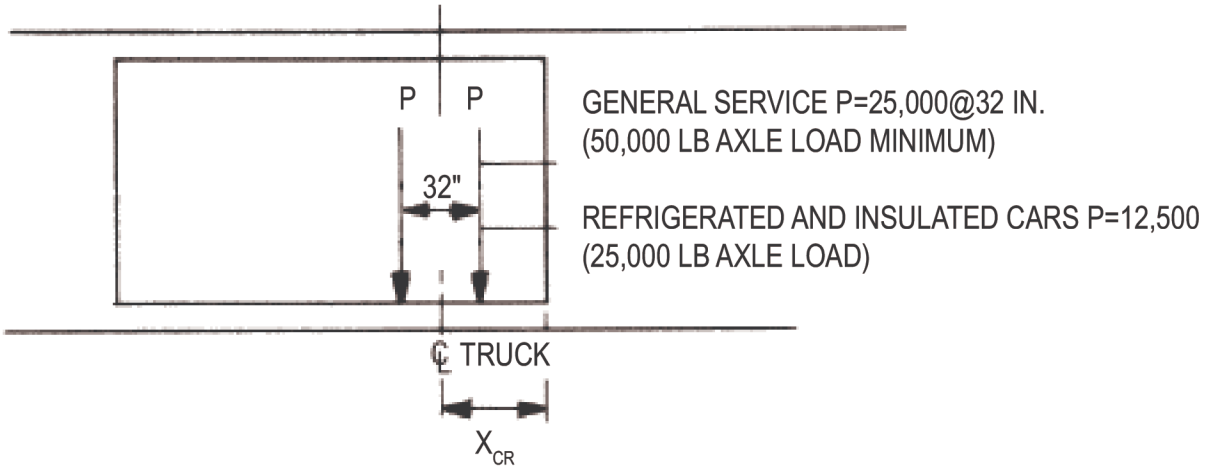


Fig. 4.3 Loading diagram



$$X_{CR} = \frac{D^2}{2T}$$

D = Width of door opening

T = Length between bolster truck centers as shown

Fig. 4.4 Loading—lift truck

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4.5.4.7 Effects of Shear

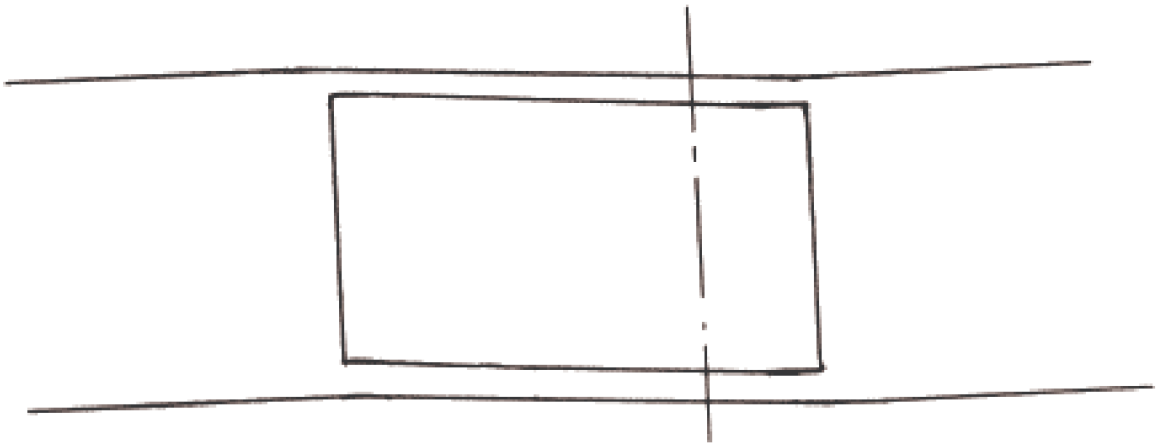


Fig. 4.5 Effect of shear



Fig. 4.6 Shear moment—upper and lower chords

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4.5.4.8 Computation of Moments and Stresses in Side Sill and Side Plate in Door Opening

4.5.4.8.1 Nomenclature

M_a	=	Bending moment at left door post from reaction at bolster (entire side as a girder)
M_b	=	Bending moment at right door post from reaction at bolster (entire side as a girder)
M_{fa}	=	Bending moment in side sill at left door post due to local loading in door opening
M_{fb}	=	Bending moment in side sill at right door post due to local loading in door opening
M_s	=	Shear moment—total
M_{st}	=	Shear moment resisted by top chord
M_s	=	Shear moment resisted by bottom chord
b		
H	=	Horizontal axial force in upper and lower chords
h	=	Distance between centers of gravity of upper and lower chords
A	=	Area of section
I_t	=	Moment of inertia of top chord
I_b	=	Moment of inertia of bottom chord
S_t	=	Section modulus of upper extreme fiber of section
S_b	=	Section modulus of lower extreme fiber of section
σ_t	=	Unit stress in upper extreme fiber of section—bending
σ_b	=	Unit stress in lower extreme fiber of section—bending
σ_c	=	Direct compressive unit stress in top and bottom chord
+		Denotes tensile stress
–		Denotes compressive stress

4.5.4.8.2 Local Bending in Side Sill in Door Opening

The lower chord or side sill is subject to local bending induced by the loads imposed thereon between the door posts. These bending moments are computed by the formulae for a beam with fixed ends.

4.5.4.8.3 Shear Moment, Side Sill, and Side Plate in Door Opening

Due to the absence of a web within the limits of the door opening, bending moments are induced in the side plate and side sill from the vertical shear in the girder across the door opening. These moments are maximum at the door posts and have equal values of opposite signs as illustrated in Fig. 4.6 and may be determined by Equation 1.

$$M_s = \frac{(M_b + M_{fb}) - (M_a + M_{fa})}{2} \qquad \text{Equation 1}$$

$$M_{st} = M_s \frac{I_t}{(I_t + I_b)} \qquad \text{Equation 2}$$

$$M_{sb} = M_s \frac{I_b}{(I_t + I_b)} \qquad \text{Equation 3}$$

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4.5.4.8.4 Maximum Unit Stresses, Side Plate and Side Sill, from Shear Moment in Door Opening

Side Plate—Left End

$$\sigma_t = +\frac{M_{st}}{S_t} \quad \text{Equation 4}$$

$$\sigma_b = -\frac{M_{st}}{S_b} \quad \text{Equation 5}$$

Side Plate—Right End

$$\sigma_t = -\frac{M_{st}}{S_t} \quad \text{Equation 6}$$

$$\sigma_b = +\frac{M_{st}}{S_b} \quad \text{Equation 7}$$

Side Sill—Left End

$$\sigma_t = +\frac{M_{sb}}{S_t} \quad \text{Equation 8}$$

$$\sigma_b = -\frac{M_{sb}}{S_b} \quad \text{Equation 9}$$

Side Sill—Right End

$$\sigma_t = -\frac{M_{sb}}{S_t} \quad \text{Equation 10}$$

$$\sigma_b = +\frac{M_{sb}}{S_b} \quad \text{Equation 11}$$

4.5.4.8.5 Direct Stress, Side Plate and Side Sill, from Axial Force in Door Opening

The top and bottom chords (side plate and side sill) are also subject to direct compressive and tensile stresses, respectively, due to a horizontal axial force H, determined by Equation 12.

$$H = \frac{(M_b + M_{fb}) + (M_a + M_{fa})}{2h} \quad \text{Equation 12}$$

$$\sigma_c = \pm\frac{H}{A} \quad \text{Equation 13}$$

4.5.4.8.6 Direct Compressive Stress, Side Sill, Due to Increment of End Load

The side sill is also subjected to a direct axial compressive stress induced by the increment of the compressive end load.

4.5.4.8.7 Total Maximum Stresses, Side Sill and Side Plate

The total maximum unit stresses in the side sill and side plate are determined by algebraic summation of the unit stresses found in accordance with paragraphs 4.5.4.8.2, 4.5.4.8.4, 4.5.4.8.5, and 4.5.4.8.6, as applicable.

4.5.4.8.8 Door Posts and Intermediate Side Posts

The door posts and intermediate side posts shall be designed for the dead load and critical live load combined with lateral forces and shall conform to the requirements of paragraph 4.2.2.7.

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4.5.5 Sides, Gondola Cars

The sides shall be designed as girders supported at bolsters for the following loads and load combinations:

- All conditions of dead load (paragraph 4.1.2), live load (paragraph 4.1.3), buff or draft load (paragraph 4.1.8), lateral pressure load (paragraph 4.1.14.1) or leaning load (paragraph 4.1.14.2) and centrifugal load (paragraph 4.1.14.3). The maximum stresses shall conform to the requirements of paragraph 4.2.2.7.1.
- All conditions of dead load (paragraph 4.1.2), live load (paragraph 4.1.3), compressive end load (paragraph 4.1.9) and lateral pressure load (paragraph 4.1.14.1) or leaning load (paragraph 4.1.14.2). The maximum stresses shall conform to the requirements of paragraph 4.2.2.7.2.
- All conditions of dead load (paragraph 4.1.2), live load (paragraph 4.1.3), impact load (paragraph 4.1.10) and lateral pressure load (paragraph 4.1.14.1) or leaning load (paragraph 4.1.14.2). The maximum stresses shall conform to the requirements of paragraph 4.2.2.7.3.

4.5.5.1 Side Stakes

Side stakes and supporting structure shall be designed for the entire critical lateral load. The maximum stresses shall conform to the requirements of paragraph 4.2.2.7.

4.5.5.2 Corner Posts, Drop End Gondola Cars

The corner posts for drop end gondola cars shall be considered a cantilever beam fixed at the end sill, and the strength of the end sill must be adequate to support this condition. The section modulus of each corner post about its axis parallel to the side of the car shall be equal to 50% of the required section moduli of all the bolster and crossbearer stakes on one side of the car.

4.5.5.3 Top Chords, High Side Gondola

Analysis is to account for $^{3}/_{4}$-in. deviation from straightness of the top chord over the total length.

4.5.6 Sides, Open Top Hopper Cars

The sides shall be designed as girders supported at bolsters for the following loads and load combinations:

- All conditions of dead load (paragraph 4.1.2), live load (paragraph 4.1.3), buff or draft load (paragraph 4.1.8), lateral pressure load (paragraph 4.1.14.1) and centrifugal load (paragraph 4.1.14.3). The maximum stresses shall conform to the requirements of paragraph 4.2.2.7.1.
- All conditions of dead load (paragraph 4.1.2), live load (paragraph 4.1.3), compressive end load (paragraph 4.1.9) and lateral pressure load (paragraph 4.1.14.1). The maximum stresses shall conform to the requirements of paragraph 4.2.2.7.2.
- All conditions of dead load (paragraph 4.1.2), live load (paragraph 4.1.3), impact load (paragraph 4.1.10) and lateral pressure load (paragraph 4.1.14.1) or leaning load. The maximum stresses shall conform to the requirements of paragraph 4.2.2.7.3.

4.5.6.1 Top Chord

Analysis is to account for $^{3}/_{4}$-in. deviation from straightness of the top chord over the total length.

4.5.6.2 Side Stakes

Requirements of paragraphs 4.5.5.1 shall apply.

4.5.7 Sides, Covered Hopper Cars

The requirements of paragraph 4.5.5 apply where applicable.

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4.5.7.1 Side Plates

Side plates for covered hopper cars shall be designed for the following load combinations:

4.5.7.1.1 Dead load and critical live load. The maximum stresses shall conform to the requirements of paragraph 4.2.2.4.

4.5.7.1.2 Dead load, critical live load, lateral load, and draft load. The load factor for the lateral forces shall be 1.5, and for the other loads it shall be 1.8. The allowable design stress shall be the yield or 80% of ultimate, whichever is lower, or the critical buckling stress.

4.5.7.1.3 Dead load, critical live load, and impact load. The load factor for the impact load shall be 1.0, and the load factor for the dead and live loads shall be the dynamic amplification factor per paragraph 4.1.11.2. Such loading may develop the ultimate load carrying capacity of the member being investigated.

4.6 Safety Appliance Specifications and Application

Safety appliances must meet FRA and AAR requirements, specifically the *MSRP*, Section C-I, Standards S-224, S-2035, S-2042, and S-2044.

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CHAPTER 5
FABRICATION AND CONSTRUCTION

5.1 Workmanship

5.1.1 General

5.1.1.1 Cars shall be fabricated and constructed in accordance with approved drawings and/or specifications.

5.1.1.2 All workmanship shall be equal to the best practice in modern car builders' shops.

5.1.1.3 Jigs, templates, gauges, and fixtures shall be used to ensure interchangeability of parts.

5.1.1.4 All wheel and axle work shall be done in accordance with the AAR *Manual of Standards and Recommended Practices (MSRP)*, Section G-II, "Wheel and Axle Manual"; and the *Field and Office Manuals of the AAR Interchange Rules*.

5.1.1.5 Roller bearings shall be mounted in accordance with *MSRP* Section H Part II, "Roller Bearing Manual."

5.1.2 Cutting

5.1.2.1 Flame cutting shall preferably be done by machine. Flame-cut edges that will be subjected to substantial stress shall be free from gouges or nicks. Any gouges or nicks remaining after cutting must be ground smooth to eliminate stress raisers. Car designer shall specify on the drawing those edges that carry high stress and shall specify that they shall not contain nicks and gouges however caused.

5.1.2.2 Flame-cut edges that are to have weld metal deposited on them shall not have gouges or nicks deeper than $\frac{1}{8}$ in. thickness of the plate. Any gouges or nicks deeper than $\frac{1}{8}$ in. thickness of the plate, that cannot be fused during welding, shall be ground prior to welding.

5.1.2.3 All grinding shall be done in the direction of the length of the member and not transversely.

5.1.3 Finishing of Edges

Sheared or flame-cut edges of plates and shapes will not be repaired unless specifically called for on the drawings; or included in a specified edge preparation for welding; or where gouges or nicks remain after flame cutting per paragraph 5.1.2.

5.1.4 Holes for Rivets, Bolts, and Brake Pins

5.1.4.1 Holes for rivets or bolts shall be $\frac{1}{16}$ in. larger than the nominal diameter of the rivet or bolt for rivets or bolts $\frac{1}{2}$ in. diameter or larger. For rivets or bolts less than $\frac{1}{2}$ in. diameter, the holes shall be $\frac{1}{32}$ in. larger than the nominal diameter of the rivet or bolt.

5.1.4.2 Holes in material whose thickness is not greater than the diameter of the rivet or bolt plus $\frac{1}{8}$ in. may be punched. Holes in material of greater thickness should be drilled.

5.1.4.3 Where subpunching is specified, the holes shall be punched or drilled and reamed to the required diameter. Holes in center sill for bolster center filler castings and striker castings must be subpunched.

5.1.4.4 Holes for brake pins must be drilled. Hot-punched holes are acceptable only in the clevis of the hand brake chain and clevises on both ends of slack adjusters. Drilling is optional.

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5.1.5 Holes for Proprietary Fasteners

Diameter of holes for proprietary fasteners shall be in accordance with the fastener manufacturer's recommendations.

5.1.6 Slotted Holes

Slotted holes for welding or for other purposes shall be punched (preferably) or flame cut. The edges of flame-cut holes shall conform to the requirements of paragraph 5.1.2.

5.1.7 Riveting

5.1.7.1 All rivets must have button heads unless specified otherwise on drawings.

5.1.7.2 All parts shall be securely held together by bolts or other means while riveting. In no case shall the rivets be depended upon to draw the parts together.

5.1.7.3 Prior to riveting, holes should match and have full bearing for the rivets. Holes that must be enlarged to admit the rivets shall be reamed. The use of drift pins to enlarge the holes shall not be permitted.

5.1.7.4 Rivets shall be driven by power riveters, of either compression or manually operated type. When driven, they shall be tight and have full concentric heads in full contact with the surface. Rivet heads should be of uniform size throughout the work for the same size rivet. Rivets shall ordinarily be cold-driven.

5.1.7.5 Loose, burned, or otherwise defective rivets shall be replaced. In removing rivets, care should be taken not to damage the adjacent metal.

5.1.8 Welding

5.1.8.1 Introduction

5.1.8.1.1 These specifications are intended as a guide covering the general welding practices related to freight car construction, which must conform to AWS D15.1. The welding requirements specifically listed in these specifications shall be considered mandatory unless specific designs and manufacturing and service conditions dictate the application of welding procedures and techniques not so described.

5.1.8.1.2 The welding processes, techniques, and procedures listed herein are not intended to preclude the use of other processes, techniques, and procedures provided they 1) conform to established welding standards or proprietary car builder's specifications; and 2) produce welding quality consistent with design requirements and good manufacturing techniques.

5.1.8.1.3 These specifications apply only to arc welding by any of the following processes:

- Manual welding with electrodes.
- Full and semiautomatic welding by the submerged arc process
- Full and semiautomatic welding by the gas metal-arc welding process using carbon dioxide, argon, helium, oxygen, or mixtures of same
- Full and semiautomatic welding by the magnetic flux and inert-gas process
- Full and semiautomatic welding by the cored wire process, with or without inert gas

5.1.8.2 Operator Qualification

All welders and welding operators engaged in construction of cars and components shall be qualified to weld with appropriate wire and base metals in all the positions in which they are to work, in accordance with AWS D15.1 (latest revision).

5.1.8.3 Welding Symbols

Welding symbols shall be in accordance with AWS standards.

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5.1.8.4 Size of Weld

Weld size shall be as defined by the AWS. Fillet welds shall be measured with standard gauges (General Electric type or equivalent) to determine the theoretical throat section and leg size.

5.1.8.5 Material Preparation

All welding must be in accordance with AWS D15.1.

5.1.9 Cold Bending—Minimum Inside Radius

There shall be no cracks in the material after the bending operation is complete.

5.1.10 Handholds and Ladder Treads

5.1.10.1 All handholds and ladder treads shall be hot-formed. The rivet or bolt holes may be hot-punched or drilled.

5.1.10.2 Care should be exercised to have the material at the proper heat when forming, and the finished article must be free of any defects.

5.1.11 Tolerances on General Dimensions of a Completed Car

Following are recommended tolerances of the general dimensions of the completed car:

5.1.11.1	[a/]Inside length (box cars and insulated box-type cars)	– 0 in. + 3/4 in.
5.1.11.2	[a/]Inside width (box cars and insulated box-type cars)	– 0 in.[b/] + 1/2 in.[b/]
5.1.11.2.1	Inside width of posts	– 0 in.[c/] + 1/2 in.[b/]
5.1.11.2.2	Plane between posts on the same side	– 1/2 in. + 1/2 in.
5.1.11.3	Inside height (box cars and insulated box-type cars)	– 1/4 in. + 1/2 in.
5.1.11.4	Length over end sills (flatcars—all types)	– 1/2 in. + 1/2 in.
5.1.11.5	Width over side sills and/or between inside of stake pockets (flatcars—all types)	– 1/4 in. + 1/4 in.
5.1.11.6	Width between guard rails (trailer flatcars)	– 1/4 in. + 1/4 in.
5.1.11.7	Length between bulkheads (bulkhead flatcars)	– 1/2 in. + 1/2 in.
5.1.11.8	Inside length (open top cars and covered hopper cars)	– 1/2 in. + 1/2 in.
5.1.11.9	Inside width (open top cars and covered hopper cars)—except at top chord of open top car	– 1/2 in. + 1/2 in.
5.1.11.10	Width between top chords of open top type cars	– 1/2 in. + 1 in.
5.1.11.10.1	Lateral straightness of open top-type cars either in or out from a straight line between top end chords provided the requirements of paragraph 5.1.11.10 are complied with	maximum 3/4 in.
5.1.11.11	Length over strikers (all types of cars)	–1/2 in. + 1/2 in.

[a/] Measurements shall be made at sufficient locations to assure that the inside dimensions are within the tolerances specified and shall be taken in a plane at the inside surface of the sides.

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b/ – ½ in., + ½ in. for cars equipped with side wall fillers on directly opposite walls; and –¼ in., + ½ in. for cars equipped with side wall fillers in diagonal opposite corner.

c/ – ½ in., + ½ in. for cars equipped with side wall fillers on directly opposite walls; and –¼ in., + ½ in. for cars equipped with side wall fillers in diagonal opposite corner.

5.1.11.12 Draft Gear Pocket (All Types of Cars)

Tolerances are shown in *MSRP* Section C-I, Standard S-254 for draft gear pocket and Standard S-255 for cast steel strikers.

5.1.11.13 Door Opening (Box Cars and Insulated Box Type Cars)

5.1.11.13.1 Width of Door Opening

5.1.11.13.1.1	Single sliding door (distance between inside edge of front stop and outside edge of post strip)	+ ⅛ in.
5.1.11.13.1.2	Double sliding door (over outside edges of post strip)	– 0 in. + 3/16 in.
5.1.11.13.1.3	Single or double plug doors (between inside faces of door posts)	– 0 in. + ¼ in.
5.1.11.13.1.4	Combination plug and sliding doors (between inside face of combination doors or posts and outside edge of post strip)	– 0 in. + ¼ in.

5.1.11.13.2 Height of Door Opening

5.1.11.13.2.1	Height from top of door track to underside of top retainer	– 0 in. + ⅛ in.
5.1.11.13.2.2	Height from top of threshold plate to underside of door header plug door	– 0 in. + ¼ in.

5.1.11.13.3 Door Opening Out of Square

The distances between two diagonal corners of door opening shall not vary more than ⅜ in.

5.1.11.13.4 Straightness of Door Post

The door post, front stop, or post strip shall not vary more than ⅛ in. from a straight line connecting its ends.

5.1.11.13.5 Straightness of Car Sides—Arrangement of Door in Open Position

The exterior surface of the car side shall not bulge out more than ⅜ in. from the nominal vertical plane between side plate and side sill for the entire length of the door track.

5.1.11.14 Gravity-Fed Outlet Gates

For bolt-on application, flanges shall be measured for flatness at the bolt centerline. For outlets with a mounting flange length of up to and including 60 in., the outlet flange shall have a flatness of ±1/16 in. Mounting flanges over 60 in. shall have a flatness of ±3/32 in.

5.1.11.15 On new car construction and retrofit applications (with gravity-fed outlet gates), it is recommended that a 3-in. minimum clearance be maintained between the car body and a 6-ft-long, 1 1/2-in.-diameter bar in any position.

5.1.12 Air Brakes

Air brakes must be designed and tested as outlined in *MSRP,* Section E-I.

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5.1.13 High-Strength Bolts

High-strength bolts, except those used to attach combined reservoir and AB pipe bracket to supports, must be torqued sufficiently to produce the required minimum bolt tension as recommended by the bolt manufacturer.

5.1.14 Camber

5.1.14.1 General

The purpose of camber is to prevent undesirable conditions created by deflections under load.

5.1.14.1.1 Where the maximum deflection under permissible loading increases the eccentricity from line of buff or draft to neutral axis of section and produces stresses beyond permissible limits, positive camber should be provided to maintain stresses within tolerable limits.

5.1.14.1.2 Where the maximum deflection under permissible loading would encroach on clearance, sufficient positive camber shall be provided to maintain established vertical clearances as required by paragraph 2.1.4.3.

5.1.14.1.3 When camber is not specified by the customer, the amount of camber shall be left to the judgment of the car builder.

5.2 Painting

5.2.1 General

Painting and marking shall meet FRA and AAR requirements and be in accordance with specifications as approved by the customer. AAR marking requirements can be found in the *MSRP*, Section C-I, Standard S-910.

5.2.2 Preparing Surfaces for Painting

5.2.2.1 All steel surfaces to be painted must be thoroughly cleaned to remove loose mill scale, rust, grease, and foreign matter before painting is begun. This should preferably be done by blast-abrasive cleaning. If the car is blast-cleaned, trucks must be removed or protective covering applied prior to blasting, and the following parts should be thoroughly protected from the blasting abrasive:

- Air brake equipment
- Hand brake
- Slack adjuster
- Couplers
- Draft gears
- Cushioning devices
- Coupler yokes
- Plug door mechanism

5.2.2.2 Before replacing trucks or removing protective covering, blast abrasive must be thoroughly blown off car.

5.2.2.3 Cars should be primed or painted as soon as practicable after blast and cleaning to prevent rusting.

5.2.2.4 Where cars are not blast-abrasive cleaned, steel surfaces must be thoroughly cleaned with solvent or other approved chemical methods. Soil and rust that cannot be removed by solvent or other approved chemical methods must be removed by use of steel brushes or emery disc buffers.

5.2.3 Protecting Wheels

Wheels should be suitably protected during painting to avoid paint being deposited on rims and treads.

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5.3 Inspection

5.3.1 The car builder shall provide an inspection crew for inspection of material and workmanship during construction at the plant.

5.3.2 Material and workmanship shall be subject to inspection by the customer. The customer shall have the privilege of placing as many inspectors as customer deems necessary in the builder's plant. Inspectors are to have free access to all phases of the work.

5.4 Clearance Test—Truck, Car Body, and Brake Rigging

5.4.1 With the trucks angled equivalent to the minimum radius required for the uncoupled car (paragraph 2.1.4.2.2) and a simulated condition of truck springs solid, there shall be no interference between trucks and car body components.

5.4.2 With the trucks angled equivalent to the minimum radius required for the uncoupled car (paragraph 2.1.4.2.2) and a simulated condition of springs at full load, there shall be no fouling contact restricting brake application and release. The test shall be carried out by applying the brakes both with new brake shoes and with all brake shoes removed.

5.5 Weighing and Final Checking

5.5.1 Upon completion, each car shall be weighed and marked in accordance with the *Field Manual of the AAR Interchange Rules,* Rule 70.

5.5.2 Coupler heights and side bearing setup height shall be given a final check while the car is on scales or other level track.

5.6 Summary

5.6.1 Cars must be built in the best, most substantial, and workmanlike manner, according to the true intent and meaning of the approved specifications.

5.6.2 Upon completion, the interior of the car must be clean and free of all foreign matter and in a suitable condition acceptable for loading.

The exterior of the car and the inside of truck castings must be free of all loose foreign material.

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CHAPTER 6
GENERAL DESIGN AND TEST REQUIREMENTS—TANK CARS

6.1 General Design—Car Structure

6.1.1 General

Chapters 5, 6, and 7 of AAR Specification M-1001 must be used as a guide in the design, fabrication, and construction of all components of tank cars. Nonstructural attachments, upon approval by the Tank Car Committee, can be exempted from Chapter 7. The manner in which tanks are attached to the car structure must be approved.

6.1.2 Center Sills or Draft Sills

6.1.2.1 Structural steel must comply with the current issues of ASTM specifications listed in paragraph 3.1.

6.1.2.2 Center sills or draft sills must be equipped with cast steel or forged steel center plates. Steel castings must comply with the current issue of the AAR *Manual of Standards and Recommended Practices (MSRP)*, Section S-I, Specification M-201.

6.1.2.3 The minimum center sill area between points of impact must be 24.24 in.^{2} (15.639 mm^2) provided the design requirements specified in paragraph 6.2 are met.

6.1.2.4 Draft sills on all tank cars without continuous center sills must be attached to reinforcing plates that are in turn attached to the tank and must be designed as follows:

6.1.2.4.1 The welds securing the sill to the reinforcing plate shall have a total throat area not exceeding 85% of the total throat area of the reinforcing plate to shell welds. These areas may be modified to use equivalent area values if the welding procedure differs for each weld area under consideration and may also be modified for parent metal strength considerations. For calculations performed for this paragraph, only the weld securing the reinforcing plate to the shell within 16 T of weld securing the sill to the pad shall be considered effective (T = reinforcing pad thickness). The relative ultimate tensile strength shall be used for the calculation of equivalent areas. The minimum strength of the attachment of the draft sill to the reinforcing pad must comply with the requirements of paragraph 6.2.

6.1.2.4.2 The reinforcing plates must extend transversely at least 1 in. (25.4 mm) on either side of the draft sill attachment.

6.1.2.4.3 The termination of the draft sill reinforcing plates must have corner radii not less than 1 in. (25.4 mm) except that a termination on a head must have corner radii not less than 3 in. (76.2 mm).

6.1.2.5 Center sills must be reinforced at all cut-outs outside the anchor area to provide the moment of inertia, section modulus, and radius of gyration equivalent to the original section and must be reinforced against local buckling. Within the center anchor area, the sill must either be reinforced to replace the cross sectional area of the cut-outs or be restored to comply with paragraph 6.1.2.3. The reinforcement material and its attachment must develop full required strength of the center sill. Intermittent welding of the reinforcement is not permitted. Attention also should be given to the anchor ends and reinforcements to minimize stress concentrations in the center sill.

Anchor connection plates may be considered as usable reinforcements of the sill when adequately attached by continuous weld of the proper size in the area of the cut-out.

For additional anchor requirements, see *MSRP* Section C-III, Appendix E.

6.1.2.6 All cut-outs in the center sill that are flame- or arc-cut must be radiused and finished smooth without notches or must have a welded rim reinforcement applied. See paragraph 5.1.2.

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6.1.2.7 For tank cars built after November 1, 2015, in the region between the bolster centerline and the head seam, transverse welds on sill reinforcing pads that penetrate the tank shell must not extend the full width of the reinforcing pad and must not intersect longitudinal welds attaching the draft sill to the reinforcing pad.

6.1.3 Body Bolsters

6.1.3.1 Body bolsters must be steel, either cast or fabricated. Body bolsters must be designed for the loading conditions and maximum stresses contained in paragraphs 6.2.2, 6.2.3, and 4.4.8.4.

6.1.3.2 The minimum contact angle subtended by the bolster structure must be not less than 120°, exclusive of the saddle plate. Consideration must be given to the compressive load on the bolster slabbing area where applicable.

6.1.3.3 The configuration of the bolster must be such that the stress in the tank shell at the horn of the bolster saddle and at the termination of the saddle plate under static load must not exceed one sixth of the minimum tensile strength of the tank material as given in *MSRP* Section C-III, Appendix M. These stress values must be determined by tests or by other acceptable analytical methods.

6.1.3.4 Bolsters on all tank cars without continuous center sills must be attached to reinforcing plates that are in turn attached to the tank and designed as follows:

6.1.3.4.1 The welds securing the bolster to the reinforcing plate shall have a total throat area not exceeding 85% of the total throat area of the reinforcing plate-to-shell welds. These areas may be modified to use equivalent area values if the welding procedure differs for each weld under consideration and may also be modified for parent metal strength considerations. The relative tensile strengths must be used for the calculation of equivalent areas. The design of the bolster must comply with the requirements of paragraphs 6.1.3 and 6.3.3.1.

6.1.3.4.2 The upper termination of the bolster reinforcing plates shall have corner radii not less than 3 in. (76.2 mm).

6.1.4 Jacking Pads

Provision must be made for jacking under bolsters, or at an approved alternate location, when required by car design.

6.1.5 Draft Gears, Draft Attachments, and Cushion Devices

Draft gears, draft attachments, and cushion devices must comply with *MSRP* Section B except where superseded by this Specification M-1001.

6.1.6 Couplers

6.1.6.1 Couplers must comply with *MSRP* Section S-III except where superseded by this Specification M-1001 or the *Field Manual of the AAR Interchange Rules*.

6.1.6.2 Couplers on DOT-specification cars must comply with 49 CFR 179.14 , “Coupler vertical restraint system.” All tank cars must be equipped with top-and-bottom shelf couplers.

6.1.7 Brakes

Brakes, including badge plates, must comply with *MSRP* Section E-I except where superseded by this Specification M-1001.

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6.1.8 Trucks

6.1.8.1 Trucks must comply with *MSRP* Section D.

6.1.8.2 Block-type side bearings are prohibited on tank cars ordered built new after December 31, 2001. Side bearings on tank cars ordered built new after December 31, 2001, must conform to the nominal 5 1/16-in. side bearing height per *MSRP,* Section D, Specification M-948.

6.1.9 Center of Gravity

6.1.9.1 The center of gravity of the loaded car must be reported on the Certificate of Construction and must not exceed 98 in. (2489 mm).

6.1.9.2 In computing the center of gravity, the factors for the moment arms must be as follows:

6.1.9.2.1 Trucks

Trucks based on a moment arm equal to the distance from top of rails to centerline of axle.

6.1.9.2.2 Underframe

Underframe based on a moment arm equal to the distance from top of rails to centerline of draft, less the amount of spring deflection as shown in Table 6.1. For trucks having spring deflection not shown, use deflection corresponding to that caused by gross rail load.

Table 6.1 Spring deflection

Spring	Spring Travel (in.)	Spring Deflection (in.)
D-5	3 11/16	1 5/8
D-7	4 1/4	1 7/8

6.1.9.2.3 Tank

6.1.9.2.3.1 The weight of the tank shall include the connections, insulation, and other attachments.

6.1.9.2.3.2 The moment arm calculations shall take into consideration the spring deflection (see Table 6.1).

6.1.9.2.3.3 For cylindrical tanks, the moment arm shall be measured from the center of the tank.

6.1.9.2.3.4 For non-cylindrical tanks, the moment arm shall be determined by a method that takes such geometry into effect.

6.1.9.2.4 Lading

6.1.9.2.4.1 The weight of the lading must be the difference between the rail load limit and the light weight.

6.1.9.2.4.2 The moment arm calculations shall take into consideration the spring deflection (see Table 6.1).

6.1.9.2.4.3 For cylindrical tanks with an expansion dome, the moment arm shall be the same as that used for the tank.

6.1.9.2.4.4 For cylindrical tanks without an expansion dome, the moment arm shall be 3/4 in. less than that used for the tank.

6.1.9.2.4.5 For non-cylindrical tanks, the moment arm shall be determined by a method that takes such geometry into account.

6.1.10 Limiting Outline

The limiting outline of cars must comply with paragraph 2.1.2.

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6.1.11 Horizontal and Vertical Curves

6.1.11.1 Tank cars must be designed to operate over horizontal and vertical curves in accordance with the requirements of paragraph 2.1.4.

6.1.11.2 Extreme projections of appurtenances at the centerline of car must have a minimum rail clearance in accordance with *MSRP* Section C-III, Appendix E.

6.1.12 Standard Dimensions

Standard dimensions must comply with paragraph 2.1.5.

6.2 Design Loads and Stresses

6.2.1 Vertical Loads on Pulling Face of Coupler

The car structure must be designed to withstand a 50,000 lb (222.4 kN) vertical load applied in both directions on the coupler head at the pulling face without yielding the loaded members or exceeding their critical buckling strengths.

6.2.2 Jacking Load

6.2.2.1 Car structure shall be designed to sustain 40% of gross rail load applied to each jacking pad, with stresses not to exceed yield strength. The jacks shall be placed under the portion of the car extending outside of rails, preferably under the side sills at the body bolster or under the ends of the body bolster. Suitable jacking pads shall be provided for the jacks. Lifting provisions must meet the requirements of the *MSRP*, Section C-I, Standard S-234.

6.2.2.2 The car structure of a stub sill car must be designed to withstand a load representing 125% of the upward force required to lift the fully loaded car body and nearest truck, including wheel sets. The load shall be applied to the coupler shank immediately adjacent to either the striker face or to the face of the cushion unit body. Maximum allowable stress shall be as defined in paragraph 4.2.2.5.

6.2.3 Draft and Compressive End Loads

6.2.3.1 For cars with or without continuous center sills, the car structure, including tank shell and draft attachments, must be designed so that under the most severe combination of dead and live loads and 390,000-lb (1557 kN) draft load, the induced primary plus secondary stresses do not exceed the yield or critical buckling stresses of the loaded members divided by 1.8; and under the most severe combination of dead and live load, including 15 psi (103 kPa) external pressure if applicable and 1,000,000 lb (4448 kN) compressive end load, the induced primary plus secondary stresses do not exceed the yield or critical buckling stresses of the loaded members. The compressive end load must be applied at the rear draft lugs at each end of the car on the nominal centerline of the couplers. For cars without continuous center sills, attention must be directed to the critical areas at the termination of the stub sill and the stub sill reinforcing plate. In these critical areas, the flexibilities of the mating structures must be considered.

6.2.3.2 In addition, for carbon or alloy steel tanks without continuous center sills, the primary stress in the bottom center of the tank under the combined loadings of loaded tank weight, internal pressure at the safety relief valve setting, and 390,000-lb tensile end load applied on the nominal centerline of couplers shall not exceed the minimum tensile strength of the tank material as given in *MSRP* Section C-III, Appendix M, divided by 3.7.

6.2.4 Impact Load

Design requirements for impact loads and stresses must comply with paragraph 4.1.10.

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6.3 Test Requirements

6.3.1 General Test Requirements for Car Structure

New and untried designs must be tested in accordance with this section to show compliance with requirements of paragraph 6.2.

6.3.2 Static Vertical Test

A 50,000-lb vertical load must be applied in both directions to the coupler head as near the pulling face as practicable and held for 60 seconds. Strains, when strain gauges are used, and loads must be recorded.

6.3.3 Jacking Test

6.3.3.1 The test car must be loaded to at least its rail load limit, and one end of the car must be jacked at the ends of the body bolster or at other approved jacking locations until the wheels are lifted off the rail.

6.3.3.2 A vertical upward load must be applied to the coupler shank between the coupler horn and striker at one end of the car sufficient in magnitude to lift the fully loaded car with the truck nearest the applied load until the wheels are lifted off the rail and held for 60 seconds.

6.3.4 Static Compression Test

A horizontal compressive end load of 1,000,000 lb must be applied on the centerline of draft at the rear draft lugs and held for 60 seconds. Draft gears may be removed from the test car for the static tests. Strains and compressive end loads are to be recorded to show compliance with paragraph 6.2. Strain gauges are to be located at critical locations selected by the car builder.

6.3.5 Dynamic Squeeze Test

6.3.5.1 This test is optional by the car builder but mandatory when requested by the Equipment Engineering Committee.

6.3.5.2 An impact test may be performed using a minimum of three cars backing up the test car and four impacting cars, each loaded to a minimum rail load of 177,000 lb (80,286 kg). The brakes must be set on all standing cars after all slack between cars has been eliminated. There must be no precompression of the draft gears. The standing cars must be on level tangent tracks. The striking cars, coupled together, must be used to provide repeated impacts. After each impact, the standing cars must be adjusted if necessary to restore the original condition. A series of impacts must be made at increments of 2 mph (3.22 km/h), starting at 6 mph (9.66 km/h) until a coupler force of 1,250,000 lb (5560 kN) has been reached. A visual inspection of the test car must be made during and after each of the tests. Any permanent damage requiring shopping of the car for repairs will be sufficient cause for disapproval of the design. If requested by the Equipment Engineering Committee, strain gauges are to be applied at critical locations and strains recorded during test. Critical locations are to be determined by stress coating or other approved method. Coupler forces and impact speeds must be recorded.

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6.3.6 Impact Test

The test car must be loaded to at least the rail load limit for the number and size of axles used under the car and must be equipped with the draft gear or cushioning device for which the car was designed. All cars other than the test car must be equipped with draft gears meeting the requirements of MSRP Section B, Specification M-901E, and either the struck or striking car must be equipped with an approved means for measuring coupler force. The test car must be impacted by a car of 70-ton nominal capacity, loaded to allowable gross weight on rails prescribed in paragraph 2.1.5.17, in increments of 2 mph (3.22 km/h), until a coupler force of 1,250,000 lb (5560 kN) has been reached. The impacting car should be loaded with a high-density material to provide a low center of gravity braced to prevent shifting. A visual inspection of the test car must be made during and after each of the tests. Any permanent damage requiring shopping of the car for repairs will be sufficient cause for disapproval of the design. If requested by the Equipment Engineering Committee, strain gauges are to be applied at critical locations and strains recorded during test. Critical locations are to be determined by stress coating or other approved method. Coupler forces and impact speeds must be recorded.

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CHAPTER 7
FATIGUE DESIGN OF NEW FREIGHT CARS

7.1 Administrative Provisions

7.1.1 Purpose and Scope

7.1.1.1 General

This specification is the requirement for fatigue analysis of freight cars and is not intended to supersede or make obsolete any existing AAR design requirement. In general, existing specifications are concerned with static and high-impact loads. However, when a member or connection is subjected to fluctuating stresses, failure can occur under stresses considerably lower than those that would cause failure under steady conditions. This specification gives the designer or analyst a method for estimating the fatigue life of a freight car or component when subjected to fluctuating stresses. The accuracy of fatigue analysis is, however, not as high as that of static stress analysis. While fatigue analysis methods give an indication of the adequacy of the design from a fatigue viewpoint, the actual life estimates must be utilized with caution. The intent of the specification is to ensure that all car builders uniformly utilize best practices for fatigue analysis.

The purpose of this fatigue design specification is to provide assurance that a proposed car design, or a current design with structural modifications not contemplated in the original design (and fatigue analysis), will meet the high utilization or full-interchange mileage criteria specified below.

In addition to the basic strength requirements for modifications or "new appurtenances" discussed in paragraph 1.2, a determination of satisfactory fatigue life of the modification is required. Generally, the demonstration of basic strength of appurtenances or structural modifications under static and/or impact loading alone is not sufficient.

7.1.1.2 Related Specifications and Publications

Portions of the following specification and publications form a part of this specification where applicable:

- AAR *Manual of Standards and Recommended Practices (MSRP)*, Section C-III, Specification M-1002, "Specifications for Tank Cars"
- The International Institute of Welding. 1996. *Fatigue Design of Welded Joints and Components—Recommendations of IIW Joint Working Group XIII-XV*. A. Hobbacher, Cambridge: Abington Publishing
- "Code of Practice for Fatigue Design and Assessment of Steel Structures," BS7608:1993

7.1.2 Analysis Requirements

7.1.2.1 Mileage Criteria for Analysis

The following minimum mileage criteria are to be used to determine the acceptability of fatigue life estimates. Higher mileage criteria are allowed for equipment in North American interchange service):

Unit train and high utilization cars	3,000,000 miles
General interchange	1,000,000 miles

7.1.2.2 Car Types Requiring Fatigue Analysis

This specification applies to all proposed car designs. In order to perform the analysis, it is necessary to have relevant environmental data. If these data are not available for a given car type or type of service, then the fatigue analysis must be performed using a data set that most closely matches the car type and service. The analyst must be able to justify the analysis assumptions made and adjustments made to the data.

Application of fatigue analysis to all proposed car designs is mandatory.

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7.1.2.3 Components Requiring Fatigue Analysis

The following components and their attachments require fatigue analysis. This does not preclude fatigue analyses of other possible critical components in new designs.

- Bolsters
- Center sills and stub center sills
- Buff and draft attachments and supports
- Box car doorway area
- Side sills on center sill-less cars
- Covered hopper car roofs
- Covered hopper car interior partitions
- Flatcar trailer-hitch supports
- Top chords and top chord reinforcements

7.1.3 Analysis Methods

The fatigue analysis methods used in the guideline are not considered the only methods available. Many other methods have been proposed, but only those generally accepted and proven over a wide range of test cases will be accepted in place of these methods. Those wishing to use methods other than these must be able to verify the methods' general acceptance in the field of fatigue analysis.

7.1.4 Glossary of Terms

α_i	=	Fraction of the total number of cycles applied at given stress levels $S_{max\ i}$ and $S_{min\ i}$
β	=	Total number of spectrum cycles per mile
b	=	Fatigue limit for stress ratio $R = 0$; The y-axis intercept on modified Goodman diagram curve.
CR	=	Critical stress region
K_f	=	Fatigue notch factor
K_t	=	Theoretical stress concentration factor
k	=	Absolute value of the slope of the *S-N* curve from 10,000 cycles to 2×10^6 cycles
k_2	=	Absolute value of the slope of the S-N curve for all cycles greater than 2×10^6
m	=	Slope of the modified Goodman diagram curve
N'	=	Number of cycles applied at a given level of stress
N	=	Number of cycles to produce failure at a given level of stress
N_e	=	Number of cycles that produce failure at a stress level of S_e
N_T	=	fatigue life (total number of cycles to failure) under a given spectrum loading
N_i'	=	Number of cycles applied at each level of $S_{max\ i}$ and $S_{min\ i}$
N_i	=	Number of cycles to produce failure at each level of $S_{max\ i}$ and $S_{min\ i}$
R	=	Stress ratio, S_{min}/S_{max}
S_e	=	Stress that produces failure in N_e cycles (N_e assumed 2×10^6 cycles for steel)
S_i	=	Fatigue limit computed for failure at N_i cycles
S_{max}	=	Maximum or highest algebraic applied stress
S_{min}	=	Minimum or lowest algebraic applied stress
S_r	=	Range of stress. The algebraic difference between the maximum and minimum stress in one cycle, that is, $S_r = S_{max} - S_{min}$

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7.2 Fatigue Analysis Methods

7.2.1 General

The methodology presented in this section provides an acceptable approach to fatigue analysis. The approach is to use nominal stresses and an applicable member detail. One shortcoming of this approach is the limited number of details available. However, the selection of details should be broad enough to allow their use to be expanded for analysis of similar but not exactly matching details. Great care will be required in the selection of the "closest" detail. When doubt exists among several choices, the more conservative properties should be used.

7.2.2 Analytical Method Using Applicable Fatigue Properties of Members and Details

7.2.2.1 Assumptions

7.2.2.1.1 Miners linear cumulative damage rule using nominal stresses and experimental data applies.

7.2.2.1.2 Linear material properties are assumed.

7.2.2.1.3 Nominal stresses are accurately determined analytically or experimentally.

7.2.2.1.4 Detail being analyzed corresponds realistically to the detail chosen from paragraph 7.4.

7.2.2.1.5 *S-N* curve is a straight line on log-log paper from 10,000 cycles to 2,000,000 cycles for steels with a slope independent of the stress ratio, R. For stress ranges corresponding to life less than 10,000 cycles, this analysis method does not apply.

7.2.2.1.6 For steels, the slope of the S-N curve is constant up to a life of 2,000,000 cycles. At cycles beyond 2,000,000, the slope of the S-N curve is decreased by a factor of 0.50. For aluminum alloys, the slope of the S-N curve remains constant.

7.2.2.1.7 Order of load application for environmental spectra will not affect fatigue damage accumulation.

7.2.2.1.8 The procedure for the fatigue analysis of welded joints is based on the assumption that it is necessary to consider only the ranges of cyclic principal stresses in determining fatigue damage (i.e., mean stresses are neglected for fatigue assessment of welded connections).

7.2.2.1.9 The stress range is equal to the algebraic difference between the minimum and maximum stresses.

7.2.2.1.10 The values of the stress ratio, $R = S_{min}/S_{max}$, vary from –1.0 to +1.0, inclusive.

7.2.2.1.11 No previous fatigue damage has occurred in the detail being analyzed.

7.2.2.1.12 Failure in fatigue has occurred when a crack is detected by the unaided eye.

7.2.2.1.13 The slope of the Modified Goodman Diagram is 1 (m=1), except for base material (see Table 7.75, Fig. 7.4.1.1).

7.2.2.1.14 Except for intermodal and unit train cars, the empty car stress environment resulting from longitudinal coupler forces should be considered in the fatigue life calculations.

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7.2.2.2 Derivation of Equations

According to the linear cumulative damage hypothesis, every time a component is stressed, some amount of fatigue damage occurs. This damage is a fraction of the total amount of damage that the component can withstand prior to crack initiation. Consequently, it is assumed that failure from fatigue will result when the summation of the "increments of damage" equals unity. If N' represents the number of cycles applied at some stress level and N represents the number of cycles at that same stress level that would result in failure, then failure occurs when

$$\Sigma \frac{N_i'}{N_i} = 1 \qquad \text{Equation 1}$$

If N_T represents the total number of cycles to failure of a component under a spectrum of loads, it is convenient to denote the fraction of the total number of cycles applied at each stress level by α. Therefore, $N'_i = \alpha_i N_T$ and Equation (1) may be rewritten

$$N_T \Sigma \frac{\alpha_i}{N_i} = 1 \qquad \text{Equation 2}$$

A relationship for the fatigue life under spectrum loading may now be expressed as

$$N_T = \frac{1}{\Sigma \frac{\alpha_i}{N_i}} \qquad \text{Equation 3}$$

where N_T is the total number of spectrum cycles.

The S-N curves developed using the fatigue properties table of paragraph 7.4 can be used to determine the damaging stress levels. Using the stress spectra derived from the load spectra of paragraph 7.3, the fraction of total loaded cycles, α, falling into any range can be obtained for use in Equation (3).

The maximum stress for any cycle is the algebraic total of the static stress and the dynamic stress. The fatigue properties presented in paragraph 7.4 were developed for average quality workmanship and were obtained from available literature.

It is noted that the fatigue property tables cannot be used directly to indicate the number of cycles at stress levels that would produce a fatigue failure at life less than 10,000 cycles. Experimental evidence has been collected that indicates that the classical *S-N* curve is essentially a straight line on log-log paper between a minimum of about 10,000 cycles and a maximum of 2×10^6 cycles for steel. At a life greater than 2×10^6, however, the slope of the S-N curve is reduced to approximately one half of the slope from 10,000 to 2,000,000 cycles. Knowing the slopes of the S-N curve, it is possible to predict the life N_i (number of cycles to failure) of a component subjected to any stress level. Slopes of curves are presented as "k" values in paragraph 7.4 of this specification. An equation for life, N_i, may be derived from an *S-N* diagram as follows:

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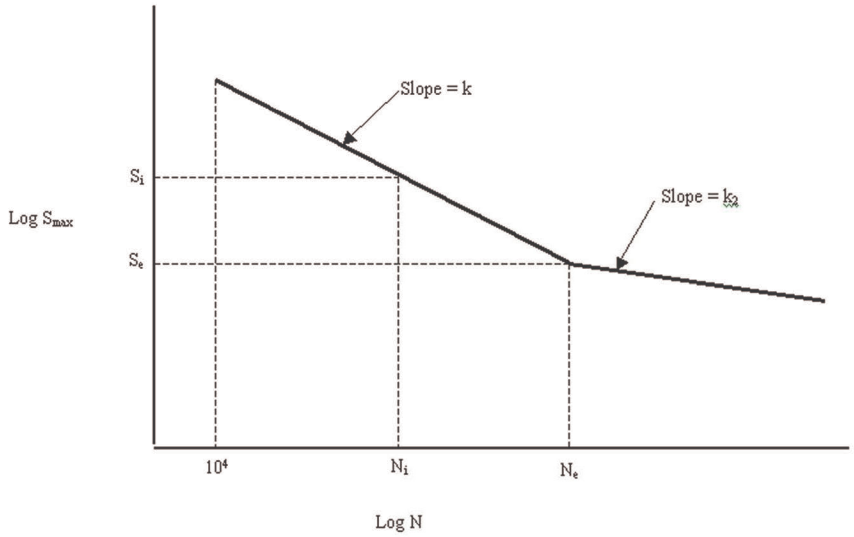


Fig. 7.1 Idealized S-N diagram

Let k = absolute slope of S-N line.

$$\frac{\log S_i - \log S_e}{\log N_e - \log N_i} = k \qquad \text{Equation 4}$$

$$k \log \frac{N_e}{N_i} = \log \frac{S_i}{S_e} \qquad \text{Equation 5}$$

$$\frac{N_e}{N_i} = \left(\frac{S_i}{S_e}\right)^{\frac{1}{k}} \qquad \text{Equation 6}$$

$$N_i = \frac{N_e}{\left(\frac{S_i}{S_e}\right)^{\frac{1}{k}}} \qquad \text{Equation 7}$$

In practice, S_r (or $S_{max\,i}$ if mean stress effects are included) replaces S_i giving the equation

$$N_i = \frac{N_e}{\left(\frac{S_r}{S_e}\right)^{\frac{1}{k}}} \qquad \text{Equation 8}$$

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Since *Se* varies with different values of *R* for unwelded plate, an equation for *Se* may be obtained from the modified Goodman diagram.

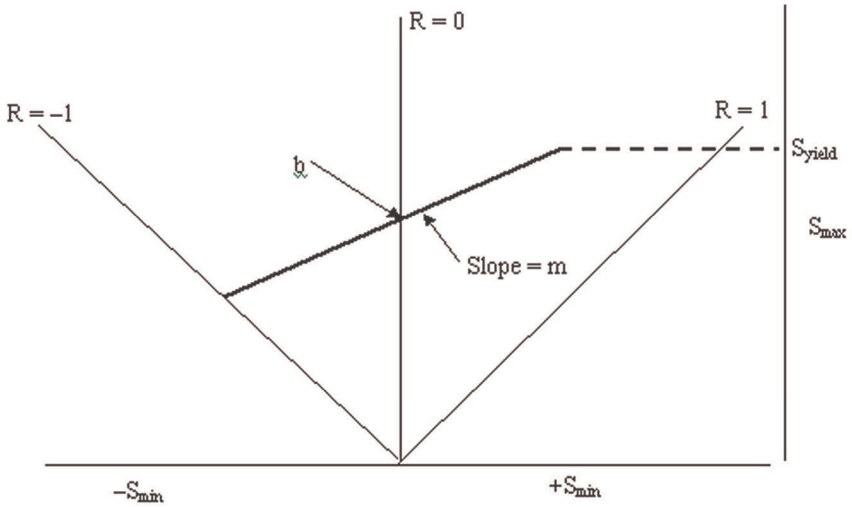


Fig. 7.2 Idealized modified Goodman diagram

Equation of MGD curve:

$$S_{max} = mS_{min} + b \qquad \text{Equation 9}$$

$$S_{min} = \frac{S_{max} - b}{m} \qquad \text{Equation 10}$$

By definition, $R = \frac{S_{min}}{S_{max}} = \frac{S_{max} - b}{\frac{m}{S_{max}}}$, but $S_{max} = S_e$ at fatigue limit. Therefore,

$$R = \frac{S_e - b}{S_e \times m} \qquad \text{Equation 11}$$

or

$$S_e = \frac{b}{1 - mR} \qquad \text{Equation 12}$$

where

b = stress range at change of slope at *R* = 0
m = slope of the Goodman curve

Using Equations (3), (8), and (12), it is possible to predict the number of cycles to failure of any component for which fatigue properties (paragraph 7.4) are available.

Finally, to determine the life of a component,

$$\text{Life (in miles)} = \frac{N_T}{\beta} \qquad \text{Equation 13}$$

where N_T = total cycles to failure
β = total spectrum cycles per mile

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7.2.2.3 Applied Stresses

Locations of likely crack initiation sites are shown for each detail in the figures of paragraph 7.4. In the case of welded connections, the stresses required are nominal stresses without consideration of stress concentrations of the detail. The nominal stresses serve as indices of load intensity (see paragraph 7.7). This approach was chosen since the actual stresses at the concentrations associated with weldments are extremely difficult if not impossible to accurately determine either analytically or experimentally.

7.2.2.4 Environmental Load Spectra

Several environmental load spectra are given in paragraph 7.3 for specific car types. The spectra are to be used to derive an appropriate stress spectrum for use in a fatigue analysis. Spectra are provided for various force components acting on the car body structure. Considerable emphasis must be placed on an accurate conversion of the load spectra into stress spectra. The stress analysis, whether analytical or experimental, is the foundation upon which the fatigue analysis is based. Since the fatigue analysis predictions are not linear relative to the stress level inputs, errors in the stress analysis can make any method of fatigue life prediction grossly inaccurate.

Two or more force component spectra may act sequentially or simultaneously to produce fatigue damage at the critical location. Completely accurate treatment of such combined spectra situations is not possible in many circumstances because of the usual uncertainty in force component time phase relationship and local stress directional effects, etc.

The current recommendation for combining fatigue lives computed from different load histories is shown in the following equation:

$$\text{Life} = \frac{1}{\frac{1}{\text{Life1}} + \frac{1}{\text{Life2}} + \ldots} \qquad \text{Equation 14}$$

where Life 1, Life 2, etc., = lives under the different force component spectra

If analysis requires a spectrum of a car type that is not available, a “best” estimate may be made by choosing a spectrum of a car type that would exhibit similar dynamic characteristics. As examples, comparisons could be made of the car body bending stiffness, height of center of gravity, torsional stiffness, types of draft gear or cushioning, etc.

Users are cautioned that selecting a “worst case” or most conservative load spectrum may result in completely unrealistic life estimates.

Dimensional, structural, and suspension characteristics of specific cars for which environmental load spectra are available are presented in paragraph 7.3.

In most cases, the effect of the empty-to-loaded car miles ratio should be considered in order to accurately describe the life of a freight car. Except for intermodal and unit train cars, the empty car stress environment resulting from longitudinal coupler forces should be considered in the fatigue life calculations. Additional factors that could modify the basic spectrum are items of varying load distribution. As an example, the distribution of trailer length carried may affect the actual stress cycles of an area being analyzed on a piggyback flatcar.

Paragraphs 4.1.3.1, 4.1.3.2, 4.1.3.2.3, 4.1.3.2.4, and 4.1.3.3 show distributions of loading to be used in the static stress analysis of the car body. They represent worst cases and may be too conservative for fatigue analysis. Fatigue analysis requires typical loading distributions.

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The three-dimensional spectrum could be simply described as an elaboration of the conventional histogram. It is put in the form of a modified Goodman diagram using similar coordinates; that is, maximum load versus minimum load. Thus, for each maximum-minimum load combination, there is entered the percentage occurrences (the third dimension) that the cycles are encountered in the environment. These percentage occurrence distributions are determined from actual environmental test records reduced to the three-dimensional format. Reduction of the test data (basically cycle counting) could be done using one of many methods; however, the Rainflow counting method is currently considered the best available and should be used in obtaining load spectrum information.

Note: A description of the Rainflow counting method with application to freight car fatigue life analysis is provided in "User's Guide to Rainflow Counting Program," AAR report R-274, July 1977.

In addition to the selection of spectra based on dynamic characteristics, the empty-loaded ratios given in paragraph 7.3 should be used when applicable (i.e., vertical load spectra). These ratios should be used as listed unless more specific information is available for the car being analyzed.

7.2.2.5 Material Properties for Fatigue Analysis

The modified Goodman diagram or MGD given in paragraph 7.2.2.2 is a convenient means of indicating the allowable stresses for dynamic stress cycles of various mean stress levels for unwelded plate. The horizontal scale shows the minimum stress for the cycle, and the vertical scale shows the maximum stress (positive or tensile stress). A vertical line is drawn through zero on the horizontal scale representing all points for zero-to-tension cycles; i.e., the minimum stress is zero with any selected maximum tensile stress. This line is denoted as the $R = 0$ line, where b is the stress ratio $R = S_{min}/S_{max}$. Cycles to the right of the $R = 0$ line will be tension-to-tension cycles, while to the left will be compression-to-tension cycles. The line to the right inclined at 45° is the $R = +1$ line representing a static stress condition. A similar line to the left is the $R = -1$ line representing complete stress reversal.

The modified Goodman diagram constants for unwelded plate are presented in tabular form in paragraph 7.4. The tabular data are conservative estimates that are not rigidly based on statistical analysis of the test data. The tabulated values have been factored from the mean values. This was necessary due to significant differences in sample sizes for various details and corresponding ranges in scatter of data. A factor of 0.70 was generally used to obtain representative MGD y-axis-intercept values (b) at 2×10^6 cycles for steels based on the mean values at that life. These reduced values correspond to a value approximately two standard deviations below the mean strength at 2×10^6 cycles.

If the data are insufficient to determine the fatigue properties of a member or detail in aluminum and the fatigue strength is available for the member or detail in steel, then it can be assumed that the aluminum fatigue strength (MGD y-axis-intercept) is about equal to one-third (1/3) of the steel fatigue strength and the MGD has the same S-N slope, k, and the same MGD slope, M, with verification or adjustment of these strengths by comparison with other design codes. The one-third factor assumes that fatigue life consists of fatigue crack propagation only, which is a function of the modulus of elasticity.

The use of stress-life (S-N curve) details is desirable, since this data represents tests of an actual construction; however, the test samples do not necessarily represent the actual connections as built in a freight car. While the details account for concentrations associated with average quality welding, influences such as residual stresses, welding variation, temperature, and corrosion are not considered. When choosing a detail to approximate a similar detail for analysis, care must be used to consider the geometry of the members and the resulting concentrations. A convenient approach is to compare the stress flow path through the detail being analyzed and the available details.

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7.2.2.6 Analysis Outline

7.2.2.6.1 Determine the nominal stress-load relationship for all applicable load inputs.

7.2.2.6.2 Select an appropriate load spectrum.

7.2.2.6.3 Select appropriate stress-life and/or modified Goodman diagram constants for the detail to be analyzed.

7.2.2.6.4 Tabulate $\Sigma\frac{\alpha_i}{N_i}$, including all factors such as empty-loaded ratio and multiple-load spectra.

7.2.2.6.5 Determine β for the chosen spectrum and calculate the life = N_T/β in miles.

7.2.3 Analytical Method Using Maximum Stress

7.2.3.1 Assumptions

The same assumptions (paragraph 7.2.2.1) made in the nominal stress method apply.

7.2.3.2 Analysis Outline

7.2.3.2.1 Determine nominal stress-load relationship for all applicable load inputs.

7.2.3.2.2 Obtain maximum stresses by multiplying nominal stresses by appropriate fatigue notch factor, K_f. If the value of K_f cannot be obtained, then use K_t, the theoretical stress concentration factor. The use of K_t will always result in a conservative analysis for ductile steels.

Note: K_f and K_t can be obtained from textbooks such as *Stress Concentration Design Factors*, R.E. Peterson, Wiley and Sons, 1974, or *Stress and Strength of Manufactured Parts*, Lipson, Noll, and Clock, McGraw-Hill Company, 1950.

7.2.3.2.3 Use fatigue properties for base material (paragraph 7.4)

7.2.3.2.4 Calculate fatigue life from formulas previously developed for nominal stress method.

7.2.4 Manual Calculation Example

7.2.4.1 Manual Calculation—Analytical Method Using Applicable Fatigue Properties of Members and Details

The following fatigue calculation is based on single-spectrum loading. The member analyzed is assumed to be part of a body bolster for an unspecified car (for illustration purposes only). Because the detail being analyzed is a beam reinforced by a partial cover plate, the design stresses will be the nominal stresses at the end of the cover plate and not the true maximum stresses that would be used if we were doing a basic material analysis. The nominal maximum and minimum stresses developed are fictitious and are used only for purposes of illustration.

7.2.4.1.1 Data Required for Calculations

7.2.4.1.1.1 Environmental Stresses (S_{max} and S_{min})

The dynamic stresses are determined from the road environment-percentage occurrence spectrum for vertical Gs at bolster, for an unspecified 100-ton car, fully loaded (Table 7.3). The static (1G) stress due to dead and live loads is assumed to be 10,000 psi tension, and the dynamic stress is assumed for this example to vary in direct ratio to the static stress. For example, for a ±0.2G acceleration, the dynamic stress is ±0.2(10000) = ±2000 psi.

7.2.4.1.1.2 Total Cycles per Mile (β)

The average cycles per mile is 354.661 (see Table 7.3).

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7.2.4.1.1.3 Material

The material is assumed to be ASTM A441 having Sy = 50000 psi.

7.2.4.1.1.4 Fatigue Properties of Detail

The detail chosen to represent this case is a beam reinforced with a partial length cover plate applied with longitudinal and transverse fillet welds. The intercept (S_e) is 7400 psi, and the S-N slopes (k and k_2) are 0.35 and 0.35/2. Since this is a welded joint, the mean stress is assumed to have no effect upon the fatigue damage, and the MGD slope is not needed.

7.2.4.1.1.5 Ratio of Empty to Loaded Miles

Referring to paragraph 7.3.1 for the ratios of empty to loaded freight car miles by type of car, the ratio for equipped box cars is 0.93.

7.2.4.1.2 Calculations

7.2.4.1.2.1

Tables 7.1 and 7.2 present the calculated results. In column 1 of Table 7.1, the dynamic stresses are calculated using the G values from the spectrum. From first entry in the spectrum of Table 7.3, the accelerations are 0.1G up (maximum) and 0.0G down (minimum) for 1.415% occurrences. It is assumed that acceleration up creates tensile stresses and acceleration down causes compressive stresses. Therefore, S_{max} = 0.1(10,000) = 1,000 psi, and S_{min} = 0 psi.

For a later entry in the spectrum, the accelerations are 0.3G up and 0.4G down for 0.067% occurrences. The corresponding stresses are 0.3(10,000) = 3,000 psi and –0.4(10,000) = –4,000 psi. Stress ranges, $S_r = S_{max} - S_{min}$, are listed in column 2 of Table 7.1.

For those spectrum components in which S_r equals or exceeds S_e, the computed value $(S_r / S_e)^{1/k}$ is listed in column 3. Similarly, for those spectrum components in which S_e exceeds S_r, the computed value $(S_r / S_e)^{1/k2}$ is listed in column 4. N_i, listed in column 5, is then computed to be equal to N_e (= 2,000,000) divided by the appropriate value from column 3 or column 4.

Fatigue damage fractions are computed in Table 7.2. The cycle fractions for each spectrum component, α_i, from Table 7.3 are included in column 1. These are obtained from the percentages listed in the REPOS divided by 100. The cycles to failure for each spectrum component, N_i, from column 5 of Table 7.1 are included in column 2. The total damage per cycle for each spectrum component is then calculated to equal α_i / N_i and listed in column 3. These damages are then summed, $S(\alpha_i / N_i)$, to give the total damage per cycle, 2.467×10^{-9}. The fatigue life in cycles, N_T, is the reciprocal of that sum ($N_T = 1 / 2.467 \times 10^{-9}$ = 405,423,000 cycles), and the fatigue life in miles is

$$\text{Life} = \frac{N_T}{\beta} = \frac{405{,}423{,}000 \text{ cycles}}{354.661 \text{ cycles/mile}} = 1{,}143{,}000 \text{ miles}$$

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Table 7.1 Calculation examples for unspecified car

1		2	3	4	5
Dynamic Stress		S_r Stress Range	$\left(\frac{S_r}{S_e}\right)^{\frac{1}{0.35}}$	$\left(\frac{S_r}{S_e}\right)^{\frac{2}{0.35}}$	N_i Number of Cycles to Failure
S_{max}	S_{min}				
1000	0	1000	NA	0.0000108	185,382,355,000
2000	0	2000	NA	0.000566	3,530,994,000
3000	0	3000	NA	0.00575	348,066,000
1000	-1000	2000	NA	0.000566	3,530,994,000
2000	-1000	3000	NA	0.00575	348,066,000
3000	-1000	4000	NA	0.0297	67,255,000
4000	-1000	5000	NA	0.1064	18,791,000
1000	-2000	3000	NA	0.00575	348,066,000
2000	-2000	4000	NA	0.0297	67,255,000
3000	-2000	5000	NA	0.1064	18,791,000
4000	-2000	6000	NA	0.3017	6,630,000
1000	-3000	4000	NA	0.0297	67,255,000
2000	-3000	5000	NA	0.1064	18,791,000
3000	-3000	6000	NA	0.3017	6,630,000
4000	-3000	7000	NA	0.7279	2,747,000
5000	-3000	8000	1.2495	NA	1,601,000
1000	-4000	5000	NA	0.1064	18,791,000
2000	-4000	6000	NA	0.3017	6,630,000
3000	-4000	7000	NA	0.7279	2,747,000
4000	-4000	8000	1.2495	NA	1,601,000
5000	-4000	9000	1.7494	NA	1,143,000
1000	-5000	6000	NA	0.3017	6,630,000
2000	-5000	7000	NA	0.7279	2,747,000
3000	-5000	8000	1.2495	NA	1,601,000
4000	-5000	9000	1.7494	NA	1,143,000
5000	-5000	10,000	2.3639	NA	846,000
2000	-6000	8000	1.2495	NA	1,601,000
3000	-6000	9000	1.7494	NA	1,143,000
4000	-6000	10,000	2.3639	NA	846,000
5000	-6000	11,000	3.1038	NA	644,000
6000	-7000	13,000	5.0024	NA	399,810
2000	-7000	9000	1.7494	NA	1,143,000
4000	-7000	11,000	3.1038	NA	644,000
5000	-7000	12,000	3.9797	NA	503,000
0	-1000	1000	NA	0.0000108	185,382,355,000
0	-2000	2000	NA	0.000566	3,530,994,000
-1000	-2000	1000	NA	0.0000108	185,382,355,000
0	-3000	3000	NA	0.00575	348,066,000
0	-4000	4000	NA	0.0297	67,255,000
-1000	-4000	3000	NA	0.00575	348,066,000
-2000	-4000	2000	NA	0.000566	3,530,994,000
-2000	-7000	5000	NA	0.1064	18,791,000

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Table 7.2 Calculation examples for unspecified car

1	2	3
α_i	Ni from Table 7.1	$\frac{\alpha_i}{N_i}$
0.014150	185,382,355,000	0.076×10^{-12}
0.000130	3,530,994,000	0.037×10^{-12}
0.000010	348,066,000	0.029×10^{-12}
0.381790	3,530,994,000	108.1×10^{-12}
0.007750	348,066,000	22.27×10^{-12}
0.000180	67,255,000	2.676×10^{-12}
0.000010	18,791,000	0.532×10^{-12}
0.054630	348,066,000	157.0×10^{-12}
0.016290	67,255,000	242.2×10^{-12}
0.001160	18,791,000	61.73×10^{-12}
0.000060	6,630,000	9.050×10^{-12}
0.002520	67,255,000	37.47×10^{-12}
0.004530	18,791,000	241.1×10^{-12}
0.001740	6,630,000	262.4×10^{-12}
0.000140	2,747,000	50.96×10^{-12}
0.000020	1,601,000	12.49×10^{-12}
0.000090	18,791,000	4.790×10^{-12}
0.000470	6,630,000	70.89×10^{-12}
0.000670	2,747,000	243.9×10^{-12}
0.000260	1,601,000	162.3×10^{-12}
0.000040	1,143,000	35.00×10^{-12}
0.000010	6,630,000	1.508×10^{-12}
0.000030	2,747,000	10.92×10^{-12}
0.000170	1,601,000	106.2×10^{-12}
0.000190	1,143,000	166.2×10^{-12}
0.000020	846,000	23.64×10^{-12}
0.000010	1,601,000	6.246×10^{-12}
0.000020	1,143,000	17.50×10^{-12}
0.000030	846,000	35.46×10^{-12}
0.000040	644,000	62.11×10^{-12}
0.000020	399,810	50.02×10^{-12}
0.000010	1,143,000	8.749×10^{-12}
0.000010	644,000	15.53×10^{-12}
0.000020	503,000	39.76×10^{-12}
0.498420	185,382,355,000	2.635×10^{-12}
0.013630	3,530,994,000	3.860×10^{-12}
0.000010	185,382,355,000	5.394×10^{-17}
0.000070	348,066,000	0.201×10^{-12}
0.000010	67,255,000	0.149×10^{-12}
0.000010	348,066,000	0.029×10^{-12}
0.000010	3,530,994,000	2.832×10^{-15}
0.000010	18,791,000	0.532×10^{-12}

$$\sum \frac{\alpha_i}{N_i} = 2.276 \times 10^{-9}$$

$$N_T = \frac{1}{2.467} \times 10^{-9} = 439{,}302{,}017 \text{ cycles}$$

$$\text{Life} = \frac{N_T}{\beta} = \frac{405{,}423{,}000}{354.661} = 1{,}238{,}653 \text{ miles (loaded only)}$$

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Table 7.3 Road environment percent occurrence spectrum
Vertical Gs at bolster for unspecified car

Vertical Gs		Percent
Maximum	Minimum	Occurrence
0.1	0.0	1.4150
0.2	0.0	0.0130
0.3	0.0	0.0010
0.1	-0.1	38.1790
0.2	-0.1	0.7750
0.3	-0.1	0.0180
0.4	-0.1	0.0010
0.1	-0.2	5.4630
0.2	-0.2	1.6290
0.3	-0.2	0.1160
0.4	-0.2	0.0060
0.1	-0.3	0.2520
0.2	-0.3	0.4530
0.3	-0.3	0.1740
0.4	-0.3	0.0140
0.5	-0.3	0.0020
0.1	-0.4	0.0090
0.2	-0.4	0.0470
0.3	-0.4	0.0670
0.4	-0.4	0.0260
0.5	-0.4	0.0040

Vertical Gs		Percent
Maximum	Minimum	Occurrence
0.1	-0.5	0.0010
0.2	-0.5	0.0030
0.3	-0.5	0.0170
0.4	-0.5	0.0190
0.5	-0.5	0.0020
0.2	-0.6	0.0010
0.3	-0.6	0.0020
0.4	-0.6	0.0030
0.5	-0.6	0.0040
0.6	-0.6	0.0020
0.2	-0.7	0.0010
0.4	-0.7	0.0010
0.5	-0.7	0.0020
0.0	-0.1	49.8420
0.0	-0.2	1.3630
-0.1	-0.2	0.0010
0.0	-0.3	0.0070
0.0	-0.4	0.0010
-0.1	-0.4	0.0010
-0.2	-0.4	0.0010
-0.2	-0.7	0.0010

Positive force is UP
Negative force is DOWN

Avg. cycles/mile: 354.661

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7.2.4.1.3 Effect of Empty to Loaded Ratio

As stated for this example,

$$\frac{\text{empty mileage}}{\text{loaded mileage}} = 0.93$$

or

$$\text{empty mileage} = 0.93 \times \text{loaded mileage}$$

or

$$\text{total mileage} = 1.93 \times \text{loaded mileage}$$

7.2.4.1.3.1 Recalculate life for loaded portion of usage as

$$\text{Life}_{LD} = \frac{N_T}{\beta_{LD}}$$

where β_{LD} = number of occurrences per mile

$$\beta_{LD} = \beta \times \frac{\text{loaded mileage}}{\text{total mileage}}$$

For this example,

$$\beta_{LD} = 354.661 \times \frac{1}{1.93} = 183.762 \text{ loaded cycles/mile}$$

$$\text{Life}_{LD} = \frac{N_T}{\beta_{LD}} = \frac{439{,}302{,}017}{183.762} = 2{,}390{,}601 \text{ miles}$$

7.2.4.1.3.2 Calculate empty car stresses:

In this example, empty car stresses are obtained by ratio of car weights

$$\text{empty car stress} = \text{loaded car stress} \times \frac{\text{car body weight}}{(\text{maximum weight on rails}) - (\text{truck weight})}$$

In this example,

$$S_{EMP} = S_{LD} \times \frac{90{,}000}{240{,}000} = .375\,S_{LD}$$

7.2.4.1.3.3 Repeat the analysis as done in Tables 7.1 and 7.2, but with the reduction factor (0.375 in this example) applied to S_{max} and S_{min}. The result from this is 84,968,963,000 cycles or 239,578,000 miles for 354.661 cycles/mile.

7.2.4.1.3.4 Recalculate life for unloaded portion of usage as

$$\text{Life}_{EMP} = \frac{N_T}{\beta_{EMP}}$$

where β_{EMP} = number of occurrences per mile

$$\beta_{EMP} = \beta \times \frac{\text{empty mileage}}{\text{total mileage}}$$

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For this example,

$$\beta_{EMP} = 354.661 \times \frac{0.93}{1.93} = 170.899 \text{ empty cycles/mile}$$

$$\text{Life}_{LD} = \frac{N_T}{\beta_{LD}} = \frac{88{,}186{,}009{,}217}{170.899} = 516{,}012{,}967 \text{ miles}$$

7.2.4.1.3.5 Calculate total life (loaded and empty):

$$\text{Total life} = \frac{1}{\frac{1}{\text{Life}_{LD}} + \frac{1}{\text{Life}_{EMP}}}$$

$$\text{Total life} = \frac{1}{\frac{1}{2{,}206{,}000} + \frac{1}{498{,}445{,}000}} = 2{,}379{,}577 \text{ miles}$$

7.2.4.2 Manual Calculation—Maximum Stress Method

The following fatigue calculations illustrate the maximum stress method of determining fatigue life. The member analyzed is assumed to be a center sill of a covered hopper subjected to a single spectrum loading. For the purposes of this example, it is assumed that the REPOS is identical for loaded and unloaded miles.

7.2.4.2.1 Data Required for Calculations

7.2.4.2.1.1 Environmental Stress (S_{max} and S_{min})

The static stress for this example is assumed to be zero. The dynamic stresses are determined from the road environment percentage occurrence spectrum for longitudinal coupler force—100 ton hopper car (as in Table 7.7). The stresses are assumed to apply for both empty and loaded cars. The nominal stress for a 100,000 lb draft coupler force is assumed to be 9000 psi and for a 100,000 lb buff force is –9000 psi. At the location of the above nominal stresses in the center sill, a sharp groove with a very small radius exists. It is assumed this notch creates a fatigue notch factor K_f of 3.0 and is similar to the detail shown in chart 77 on page 135 of *Stress and Strength of Manufactured Parts*, Lipson, Noll, and Clock, McGraw-Hill, 1950.

7.2.4.2.1.2 Total Cycles per Mile (β)

The average cycles per mile from the REPOS is 210.07.

7.2.4.2.1.3 Material

The material is assumed to be ASTM A242 having S_y = 50,000 psi.

7.2.4.2.1.4 Fatigue Properties

The fatigue properties applicable for this analysis are in paragraph 7.4, Table 7.75, Fig. 7.4.1.1 (Plate, as-rolled—axial load). The MGD intercept (b) is 31,000 psi, the slope of the MGD (m) is 0.86, and the S-N slopes (k and k_2) *are* 0.10 and 0.05.

7.2.4.2.2 Calculations

Tables 7.4, 7.5, and 7.6 show the computation to determine the number of cycles to failure, N_i. In Table 7.4, the coupler forces in column 1 are obtained from the load spectrum. The maximum dynamic stresses in column 2 include a fatigue notch factor of 3, so that

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$$\text{Stress} = \text{Force} \times 3 \times \frac{9000 \text{ psi}}{100{,}000 \text{ lb}}$$

Column 3 lists the fraction of total loaded cycles, α_i. These are obtained from the percentages listed in the REPOS and divided by 100. The calculated values of the stress ratio, $R = \frac{S_{MIN}}{S_{MAX}}$, are in column 4. The values of S_e in column 5 are computed as

$$S_e = \frac{b}{1 - mR} = \frac{31{,}000}{1 - 0.86R}$$

Ni is computed in Table 7.5. Columns 1 and 2 are the S_{max} and S_e values taken from columns 2 and 5 of Table 7.4. For those spectrum components in which S_{max} equals or exceeds S_e, the computed value $(S_{max} / S_e)^{1/k}$ is listed in column 3. Similarly, for those spectrum components in which S_e exceeds S_{max}, the computed value $(S_{max} / S_e)^{2/k}$ is listed in column 4. N_i, listed in column 5, is then computed to be equal to N_e (= 2,000,000) divided by the appropriate value from column 3 or column 4.

Fatigue damage fractions are computed in Table 7.5. The cycle fractions for each spectrum component, α_i, from column 3 of Table 7.4, are included in column 1. The cycles to failure for each spectrum component, N_i, from column 5 of Table 7.5, are included in column 2. The total damage per cycle for each spectrum component is then calculated to equal α_i / N_i and listed in column 3. These damages are then summed, $\Sigma (\alpha_i / N_i)$, to give the total damage per cycle, 7.195×10^{-11}. The fatigue life in cycles, N_T, is the reciprocal of that sum ($N_T = 1 / 7.195 \times 10^{-11} = 13{,}898{,}682{,}000$ cycles), and the fatigue life in miles is

$$\text{Life} = \frac{N_T}{\beta} = \frac{13{,}898{,}682{,}000 \text{ cycles}}{105.03 \text{ cycles/mile}} = 132{,}331{,}000 \text{ miles}$$

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Table 7.4 Calculation examples for unspecified car

1		2		3	4	5
Coupler Force (lb) from REPOS		Maximum Dynamic Stresses (psi)		α_i Fraction of Total Cycles	R $\frac{S_{min}}{S_{max}}$	S_e Reference Stress
Maximum	Minimum	S_{max}	S_{min}			
20,000	0	5,400	0	0.436560	0	31,000
40,000	0	10,800	0	0.268660	0	31,000
60,000	0	16,200	0	0.000910	0	31,000
80,000	0	21,600	0	0.000900	0	31,000
100,000	0	27,000	0	0.000010	0	31,000
40,000	20,000	10,800	5400	0.248080	0.500	54,400
60,000	20,000	16,200	5400	0.019790	0.333	43,400
80,000	20,000	21,600	5400	0.000220	0.250	39,500
100,000	20,000	27,000	5400	0.000070	0.200	37,400
120,000	20,000	32,400	5400	0.000020	0.167	36,200
60,000	40,000	16,200	10,800	0.017730	0.667	72,800
80,000	40,000	21,600	10,800	0.002350	0.500	54,400
100,000	40,000	27,000	10,800	0.000040	0.400	47,300
140,000	40,000	37,800	10,800	0.000020	0.285	41,100
80,000	60,000	21,600	16,200	0.001830	0.750	87,300
100,000	60,000	27,000	16,200	0.000150	0.600	64,000
180,000	60,000	48,600	16,200	0.000020	0.333	43,400
20,000	-20,000	5,400	-5400	0.001460	-1.000	16,700
40,000	-20,000	10,800	-5400	0.000020	-0.500	21,700
60,000	-20,000	16,200	-5400	0.000020	-0.333	24,100
20,000	-40,000	8,100*	-8100	0.000190	-1.000	16,700
40,000	-40,000	10,800*	-10,800	0.000010	-1.000	16,700
40,000	-60,000	13,500[a/]	-13,500	0.000010	-1.000	16,700
40,000	-80,000	16,200*	-16,200	0.000010	-1.000	16,700
40,000	-100,000	18,900*	-18,900	0.000010	-1.000	16,700
20,000	-120,000	18,900*	-18,900	0.000010	-1.000	16,700
0	-20,000	2,700*	-2700	0.001010	-1.000	16,700
0	-40,000	5,400*	-5400	0.000350	-1.000	16,700
-20,000	-40,000	2,700*	-2700	0.000130	-1.000	16,700
0	-60,000	8,100*	-8100	0.000040	-1.000	16,700
-20,000	-60,000	5,400*	-5400	0.000020	-1.000	16,700
-40,000	-60,000	2,700*	-2700	0.000020	-1.000	16,700
-20,000	-80,000	8,100*	-8100	0.000010	-1.000	16,700
-40,000	-80,000	5,400*	-5400	0.000020	-1.000	16,700
-20,000	-100,000	10,800*	-10,800	0.000010	-1.000	16,700
-80,000	-100,000	2,700*	-2700	0.000010	-1.000	16,700
-100,000	-140,000	5,400*	-5400	0.000010	-1.000	16,700
-120,000	-140,000	2,700*	-2700	0.000010	-1.000	16,700
-120,000	-180,000	8,100*	-8100	0.000010	-1.000	16,700
-160,000	-180,000	2,700*	-2700	0.000020	-1.000	16,700

[a/] Per paragraph 7.2.2.1.9, "The stress range is equal to the algebraic difference between the minimum and maximum stresses.."

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Table 7.5 Calculation examples for unspecified car

1	2	3	4	5
S_{max} from Table 7.1	S_e from Table 7.1	$\left(\frac{S_{max}}{S_e}\right)^{\frac{1}{0.1}}$	$\left(\frac{S_{max}}{S_e}\right)^{\frac{2}{0.1}}$	N_i Number of Cycles to Failure
5,400	31,000	NA	0.662×10^{-15}	3.02×10^{21}
10,800	31,000	NA	0.694×10^{-9}	2.88×10^{15}
16,200	31,000	NA	0.00000231	866,889,178,000
21,600	31,000	NA	0.000728	2,749,089,000
27,000	31,000	NA	0.0631	31,695,000
10,800	54,400	NA	9.05×10^{-15}	0.221×10^{21}
16,200	43,400	NA	2.75×10^{-9}	0.725×10^{15}
21,600	39,500	NA	5.72×10^{-6}	349,862,089,000
27,000	37,400	NA	0.00148	1,352,654,000
32,400	36,200	NA	0.1088	18,378,000
16,200	72,800	NA	91.1×10^{-15}	21.9×10^{18}
21,600	54,400	NA	9.49 x 10-9	0.211×10^{15}
27,000	47,300	NA	0.0000135	148,244,137,000
37,800	41,100	NA	0.1875	10,667,000
21,600	87,300	NA	0.739×10^{-12}	2.71×10^{18}
27,000	64,000	NA	31.9×10^{-9}	62.7×10^{12}
48,600	43,400	3.1007	NA	645,000
5,400	16,700	NA	0.156×10^{-9}	12.8×10^{15}
10,800	21,700	NA	0.870×10^{-6}	2.30×10^{12}
16,200	24,100	NA	0.000355	5,637,292,000
8,100	16,700	NA	0.519×10^{-6}	3.85×10^{12}
10,800	16,700	NA	0.000164	12,214,742,000
13,500	16,700	NA	0.0142	140,826,000
16,200	16,700	NA	0.5445	3,673,000
18,900	16,700	3.4471	NA	580,000
18,900	16,700	3.4471	NA	580,000
2700	16,700	NA	0.149×10^{-15}	13.4×10^{21}
5,400	16,700	NA	0.156×10^{-9}	12.8×10^{15}
2,700	16,700	NA	0.149×10^{-15}	13.4×10^{21}
8,100	16,700	NA	0.519×10^{-6}	3.85×10^{12}
5,400	16,700	NA	0.156×10^{-9}	12.8×10^{15}
2,700	16,700	NA	0.149×10^{-15}	13.4×10^{21}
8,100	16,700	NA	0.519×10^{-6}	3.85×10^{12}
5,400	16,700	NA	0.156×10^{-9}	12.8×10^{15}
10,800	16,700	NA	0.000164	12,214,742,000
2,700	16,700	NA	0.149×10^{-15}	13.4×10^{21}
5,400	16,700	NA	0.156×10^{-9}	12.8×10^{15}
2,700	16,700	NA	0.149×10^{-15}	13.4×10^{21}
8,100	16,700	NA	0.519×10^{-6}	3.85×10^{12}
2,700	16,700	NA	0.149×10^{-15}	13.4×10^{21}

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