TRANSIT COOPERATIVE RESEARCH PROGRAM

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TCRP Report 57

Track Design Handbook for Light Rail Transit

Transportation Research Board
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Track Design Handbook for Light Rail Transit

PARSONS BRINCKERHOFF QUADE & DOUGLAS, INC.
Herndon, VA

Subject Area

Rail

Research Sponsored by the Federal Transit Administration in Cooperation with the Transit Development Corporation

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The nation's growth and the need to meet mobility, environmental, and energy objectives place demands on public transit systems. Current systems, some of which are old and in need of upgrading, must expand service area, increase service frequency, and improve efficiency to serve these demands. Research is necessary to solve operating problems, to adapt appropriate new technologies from other industries, and to introduce innovations into the transit industry. The Transit Cooperative Research Program (TCRP) serves as one of the principal means by which the transit industry can develop innovative near-term solutions to meet demands placed on it.

The need for TCRP was originally identified in *TRB Special Report 213—Research for Public Transit: New Directions,* published in 1987 and based on a study sponsored by the Urban Mass Transportation Administration—now the Federal Transit Administration (FTA). A report by the American Public Transportation Association (APTA), *Transportation 2000*, also recognized the need for local, problem-solving research. TCRP, modeled after the longstanding and successful National Cooperative Highway Research Program, undertakes research and other technical activities in response to the needs of transit service providers. The scope of TCRP includes a variety of transit research fields including plan-ning, service configuration, equipment, facilities, operations, human resources, maintenance, policy, and administrative practices.

TCRP was established under FTA sponsorship in July 1992. Proposed by the U.S. Department of Transportation, TCRP was authorized as part of the Intermodal Surface Transportation Efficiency Act of 1991 (ISTEA). On May 13, 1992, a memorandum agreement outlining TCRP operating procedures was executed by the three cooperating organizations: FTA, the National Academies, acting through the Transportation Research Board (TRB); and the Transit Development Corporation, Inc. (TDC), a nonprofit educational and research organization established by APTA. TDC is responsible for forming the independent governing board, designated as the TCRP Oversight and Project Selection (TOPS) Committee.

Research problem statements for TCRP are solicited periodically but may be submitted to TRB by anyone at any time. It is the responsibility of the TOPS Committee to formulate the research program by identifying the highest priority projects. As part of the evaluation, the TOPS Committee defines funding levels and expected products.

Once selected, each project is assigned to an expert panel, appointed by the Transportation Research Board. The panels prepare project statements (requests for proposals), select contractors, and provide technical guidance and counsel throughout the life of the project. The process for developing research problem statements and selecting research agencies has been used by TRB in managing cooperative research programs since 1962. As in other TRB activities, TCRP project panels serve voluntarily without compensation.

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The TCRP provides a forum where transit agencies can cooperatively address common operational problems. The TCRP results support and complement other ongoing transit research and training programs.

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To save time and money in disseminating the research findings, the report is essentially the original text as submitted by the research agency This report has not been edited by TRB

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FOREWORD

By Staff
Transportation Research Board

This Handbook will be of interest to light rail track system design engineers, operations and maintenance professionals, vehicle design engineers and manufacturers, and others interested in the design of light rail track systems. The Handbook provides guidelines and descriptions for the design of various types of light rail transit track. The track structure types covered include ballasted, direct fixation ("ballastless"), and embedded track. The components of the various track types are discussed in detail. The guidelines consider the characteristics and interfaces of vehicle wheels and rail, track and wheel gauges, rail sections, alignments, speeds, and track moduli. The Handbook includes chapters on vehicles, alignment, track structures, track components, special trackwork, aerial structure/bridges, corrosion control, noise and vibration, signals, and traction power. These chapters provide insight into considerations that affect track design and require interface coordination.

Transit agencies frequently build new light rail transit (LRT) systems, procure light rail vehicles (LRVs), and undertake track improvements to existing systems to increase operating speeds, enhance service, and expand ridership. Many agencies have experienced accelerated vehicle wear and track degradation, attributed to the increased speeds and incompatibility of contemporary LRVs with the track structure. These problems lead to reduced service quality and increased maintenance expenditures. Considerable research has been conducted in recent years to understand the mechanisms involved in track-vehicle interaction and its effect on track design. However, no widely accepted guidelines have been developed to aid in the design of light rail transit track. Consequently, transit agencies have frequently relied on practices developed primarily for heavy rail transit and freight operations that are not well suited for light rail transit systems.

Under TCRP Project D-6, research was undertaken by Parsons Brinckerhoff Quade & Douglas to (1) better understand the interactions among track structure, LRVs, and operating characteristics and (2) develop a Handbook for the design of light rail transit track to assist the various transit disciplines in selecting the appropriate track and vehicle characteristics for specific situations.

To achieve the project objectives, the researchers first identified the track-structure parameters, vehicle characteristics, environmental factors, and operating conditions that influence track-vehicle interaction and, hence, should be considered in the design of ballasted, direct fixation, and embedded track systems. The researchers then collected and reviewed information pertaining to the design and construction of light rail transit track. A literature search of articles, manuals, texts, and manufacturers' pamphlets pertinent to light rail transit was conducted. In addition, a review of 17 North American light rail systems, as well as systems in Belgium, France, and Germany, was undertaken to investigate the different methods of design and construction. In most cases, site visits were conducted that included extensive interviews with operating and maintenance engineers. Design and construction techniques were then assessed in terms of performance, safety, and constructability. On the basis of this assessment, a Handbook providing guidance for the design of light rail track systems was prepared.

CONTENTS

AUTHOR ACKNOWLEDGMENTS

The research and development of the Track Design Handbook was performed under TCRP Project D-6 by Parsons Brinckerhoff Quade and Douglas, Inc.; Wilson, Ihrig and Associates, Inc.; and Laurence E. Daniels. Parsons Brinckerhoff was the prime contractor for this project. Parsons Brinckerhoff subcontracted noise and vibration studies to Wilson, Ihrig and Associates and track research to Laurence E. Daniels.

Gordon W. Martyn, Senior Professional Associate, Parsons Brinckerhoff Transit and Rail Systems, was the principal investigator. The Handbook authors were as follows:

Chapter 1: General Introduction: Gordon W. Martyn, Eugene C. Allen, Lawrence G. Lovejoy
Chapter 2: Light Rail Transit Vehicles: Harold B. Henderson, Theodore C. Blaschke, Gordon W. Martyn
Chapter 3: Light Rail Transit Track Geometry: Lee Roy Padget
Chapter 4: Track Structure Design: Gordon W. Martyn
Chapter 5: Track Components and Materials: Gordon W. Martyn
Chapter 6: Special Trackwork: Lawrence G. Lovejoy
Chapter 7: Aerial Structures/Bridges: David A. Charters
Chapter 8: Corrosion Control: Kenneth J. Moody, Lawrence G. Lovejoy, Gordon W. Martyn
Chapter 9: Noise and Vibration: James T Nelson
Chapter 10: Transit Signal Work: Harvey Glickenstein, Gary E. Milanowski
Chapter 11: Transit Traction Power: Kenneth Addison, Lawrence G. Lovejoy

Technical editing was performed by Eugene C. Allen of Parsons Brinckerhoff. Charles G. Mendell edited the text. Research of transit agencies was undertaken by the project team members.

Chapter 1—General Introduction

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CHAPTER 1—GENERAL INTRODUCTION

1.1 INTRODUCTION

The purpose of this Handbook is to provide to those responsible for the design, procurement, construction, maintenance, and operation of light rail transit systems an up-to-date guide for the design of light rail track, based on an understanding of the relationship of light rail track and other transit system components. The contents of the Handbook were compiled as a result of an investigation of light rail transit systems, a review of literature pertaining to transit and railroad standards and methods, and personal hands-on experience of the authors. Current research also has been a source of valuable data.

This Handbook furnishes the reader with current practical guidelines and procedures for the design of the various types of light rail track including ballasted, direct fixation, and embedded track systems. It discusses the interrelationships among the various disciplines associated with light rail transit engineering—structures, traction power stray current control, noise and vibration control, signaling, and electric traction power. The Handbook includes a chapter on light rail vehicles, describing the impact of vehicle design and operation on the track system. It also discusses the interaction between tracks and aerial structures, which is crucial when continuously welded rail and direct fixation track are used.

There are many different practical designs for light rail track, and the goal of this Handbook is to offer a range of options to the engineer. A key focus of the Handbook is to differentiate between light rail transit track and those similar, but subtly different, track systems used for freight, commuter, and heavy rail transit operations. These differences present challenges both to light rail track designers and to the designers and manufacturers of light rail vehicles.

Much research has been conducted in an effort to understand the mechanisms involved in track-vehicle interaction and its impact on track design. However, no widely accepted guidelines exist to specifically aid in the design and maintenance of light rail transit track. Consequently the light rail transit industry frequently relies on practices developed primarily for heavy rail transit and railroad freight operations that are not necessarily well suited for light rail systems.

This Handbook does not seek to establish universal standards within an industry operating in a wide range of environments. Instead it seeks to offer choices and to present the issues that must be resolved during the design process.

The user of the Handbook assumes all risks and responsibilities for selection, design, and construction to the guidelines recommended herein. No warranties are provided to the user, either expressed or implied. The data and discussions presented herein are for informational purposes only.

The reader is assumed to be an engineer or individual familiar with trackwork terminology and experienced in the application of guideline information to design. For that reason, a glossary of terms that would be familiar to a trackwork engineer has not been included herein. Definitions of common trackwork terms are included in the *Manual for Railway Engineering*, published by the American Railway Engineering & Maintenance-of-Way Association. Terms that are unique to light rail transit are defined within the text of the Handbook as they are introduced.

1.2 PURPOSE

This Handbook furnishes the reader with current practical information about light rail trackwork and guidelines for the design of the various types of light rail track including ballasted, direct fixation, and embedded track. It describes the impacts of other disciplines on trackwork, which offers the designer insights into the coordination of design efforts among all disciplines. The purpose of this Handbook is to offer a range of design guidelines, not to set a standard for the industry.

1.3 WHAT IS LIGHT RAIL, AND WHY IS IT SO HEAVY?

Tracks for light rail transit are generally constructed with the same types of materials used to construct "heavy rail," "commuter rail," and railroad freight systems. Also, light rail vehicles may be as massive as transit cars on heavy rail systems. Consequently, the term "light rail" is somewhat of an oxymoron and often misunderstood. Therefore, for the purposes of this book, it is appropriate to define light rail transit.

- Light rail is a system of electrically propelled passenger vehicles with steel wheels that are propelled along a track constructed with steel rails.
- Propulsion power is drawn from an overhead distribution wire by means of a pantograph and returned to the electrical substations through the rails.
- The tracks and vehicles must be capable of sharing the streets with rubber-tired vehicular traffic and pedestrians. The track system may also be constructed within exclusive rights-of-way.
- Vehicles are capable of negotiating curves as sharp as 25 meters (82 feet) and sometimes even sharper, in order to traverse city streets.
- Vehicles are not constructed to structural criteria (primarily crashworthiness or "buff strength") needed to share the track with much heavier railroad commuter and freight equipment.

While purists may quibble with some of the finer points of this definition, it will suffice for the purposes of this Handbook.

The two most important defining elements of light rail trackwork are the construction of track in streets, and the interface between the wheel of the light rail vehicles and the rails. Track in streets requires special consideration, especially with regard to the control of stray electrical current that could cause corrosion and the need to create a formed flangeway that is large enough for the wheels but does not pose a hazard to other users of the street. Light rail wheels, in the past, were smaller and had shallower flanges; contemporary light rail vehicle wheels are smaller and narrower than standard railroad wheels. These variations require special care in track design, especially in the design of special trackwork such as switches and frogs. The compatibility of the vehicle and track designs is a central issue in the development of a light rail system if both components are to perform to acceptable standards. These issues are discussed at length in this Handbook.

While light rail may need to share right-of-way (ROW) with pedestrians and vehicles, the designer should create an exclusive ROW for light rail tracks wherever possible. This will make maintenance and operations less expensive, and will eliminate platform height issues associated with Americans with Disabilities Act (ADA) compliance.

1.4 HANDBOOK ORGANIZATION

Chapter 2 elaborates on vehicle design and critical issues pertaining to track and vehicle interface. These topics include wheel/rail profiles, truck steering within restricted curves and primary and secondary suspension systems, and the effect of these parameters on track and operations

Chapter 3 details issues related to light rail track geometry with particular attention to restrictions imposed by alignment characteristics, such as tight radius curvature, severe vertical curves, and steep profile grade lines.

Chapter 4 elaborates on the three basic types of track structures: ballasted, direct fixation, and embedded track. The chapter takes the designer through a series of selections pertaining to the track design. The chapter discusses track and wheel gauges, flangeways, rail types, guarded track (restraining rail), track modulus, stray current, noise and vibration, and signal and traction power requirements.

The various track components and details are discussed in **Chapter 5**.

Chapter 6 provides guidelines for the design and selection of various types and sizes of special trackwork. Included are details pertaining to switches, frogs, guard rails, crossings (diamonds), and associated items.

Most light rail transit systems require bridges or similar structures. Aerial structures are not uncommon. **Chapter 7** provides a framework for determining the magnitude of forces generated due to differential thermal expansion between the rail (especially stationary continuous welded rail) and the structure. The analysis elaborates on structural restrictions, fastener elastomer displacement, fastening toe loads, friction and longitudinal restraint, and probable conditions at a rail break on the structure The analysis includes the conditional forces generated by locating special trackwork on an aerial structure and methods of contending with them

Corrosion control is a major issue arising from the use of the running rail as a negative return in the traction power system. **Chapter 8** highlights the issues pertaining to stray current and discusses the need to isolate the rail and retard the potential for electrical leakage Methodologies for establishing magnitude, identifying sources, and developing corrective measures are part of this chapter.

Chapter 9 introduces the designer to another environmental issue pertaining to light rail transit—noise and vibration. It explains wheel/rail noise and vibration and the fundamentals of acoustics. It also discusses mitigation procedures and treatments for tangent, curved, and special trackwork.

Chapter 10 highlights signal issues for light rail transit and discusses some of the interfacing issues and components that must be considered by a track designer.

Chapter 11 presents elements pertinent to traction power, including supply system and substations; the catenary distribution system; and the power return through the running rails. The chapter also discusses corrosion control measures to mitigate the effects of DC current to adjacent services.

An overall table of contents lists the eleven chapter topics. Each chapter contains its own table of contents, reference list, and list of figures and tables. Pages are numbered by chapter (for example: 4-24 is page 24 in Chapter Four).

Chapter 2 Light Rail Transit Vehicles

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CHAPTER 2—LIGHT RAIL TRANSIT VEHICLES

2.1 INTRODUCTION

Designers of the current generation of light rail vehicles (LRV) have primarily concentrated their efforts on achieving a comfortable ride for passengers and complying with Americans with Disabilities Act (ADA) requirements With respect to trucks (bogies), these efforts have resulted in primary and secondary suspension system designs that are significantly different than those employed on previous generations of electric streetcars, including the once radical design first used on Presidents' Conference Committee (PCC) trolley cars in the mid 1930s. As vehicle technology continues to evolve, so do propulsion and suspension system designs. Emerging concepts, such as independent steerable wheels, hub-mounted motors, etc., quickly lead to the conclusion that there are few hard and fast rules about the vehicle/track interface for light rail systems.

In spite of this lack of design consistency, there are several key vehicle-to-rail interface parameters that the track designer must consider during design of light rail systems. These include:

- Vehicle Weight (both empty and with full passenger load)
- Clearance
  - Required track-to-platform location tolerances to meet ADA requirements
  - Required clearance between cars on adjacent tracks considering car dynamics
  - Required route clearances (wayside, tunnel, bridge) considering car dynamics
- Wheel Dimensions
  - Wheel diameter, which can be very small in the case of low-floor vehicles and is virtually always smaller than that used on freight railroad equipment
  - Wheel profile, which must be compatible with the rail, particularly in the case of special trackwork
  - Wheel gauge to ensure compatibility with the track gauge including tolerances
  - Wheel back-to-back gauge that is compatible with flangeway dimensions and special trackwork check gauges
- Longitudinal Track Forces
  - Maximum acceleration (traction forces)
  - Deceleration from disc and tread brakes
  - Maximum possible deceleration from electromagnetic emergency track brakes
- Lateral Track Forces
  - Maximum lateral forces resulting from all speed and curvature combinations
- Dynamic Rail Forces
  - Impact of car and truck natural frequencies
  - Impact of wheel flats or damaged wheels

It is essential that the track designer and the vehicle designer discuss their designs to ensure full compatibility under all operating conditions.

Light rail vehicles are found in a variety of designs and dimensions. Cars may be unidirectional or bi-directional. In almost all cases, they are capable of being operated in coupled trains.

In most cases, LRVs are larger and heavier than their streetcar predecessors. Particularly on older existing systems, these larger replacement cars can challenge the track designer to come up with suitable methods to accommodate them.

Light rail vehicles vary in the following design characteristics:

- Unidirectional versus bi-directional
- Non-articulated versus articulated
- High floor; partially low floor (70%); low floor (100%)
- Overall size (width, length, and height)
- Truck and axle positions
- Suspension characteristics
- Performance (acceleration, speed, and braking)
- Wheel diameter
- Wheel gauge

These characteristics must be considered in the design of both the vehicle and the track structure.

The results of an investigation of the characteristics of 17 North American LRVs are summarized in **Table 2.1**. It is interesting to note that vehicle criteria published by vehicle manufacturer(s) rarely contain information on vehicle wheel gauge. Track and vehicle designers will have difficulty in the design process without first establishing this initial interface value and then determining the acceptable gap between the track and wheel gauges.

2.2 VEHICLE CHARACTERISTICS

2.2.1 Vehicle Design

2.2.1.1 Unidirectional/Bi-directional

Nearly all of the traditional streetcar systems that survived through the 1960s used unidirectional vehicles. That is, the cars were built with a control station in the forward end, doors on the right side, and a single trolley pole at the rear At the end of the line, cars negotiated a turning loop and ran to the opposite terminal. Because these vehicles could negotiate curves with centerline radii as small as 10.7 meters (35 feet), the amount of real estate needed for a turning loop was relatively small, usually only a single urban building lot. Transit companies typically found that the expense of buying properties and building loops was small compared to the savings associated with not having to maintain duplicate sets of control equipment in "double ended" trolley cars.

Current designs of high-capacity light rail vehicles have much larger minimum radius limitations and the amount of real estate that is required to construct a turning loop is much greater. Accordingly, most contemporary LRVs have control cabs in both ends and can reverse direction anywhere that a suitable crossover track or pocket track can be provided. This arrangement is usually more economical in terms of space required and has become the norm for modern light rail transit (LRT) systems. Such arrangements can be sited within the confines of a double-track right-of-way, and do not require the property acquisition (and subsequent maintenance) needed for turning loops.

2.2.1.2 Non-Articulated/Articulated

Non-articulated (rigid) cars are single car bodies carried on two four-wheel trucks. Articulated cars, on the other hand, will have two or more body sections that are connected by flexible joints.

There is a common misconception that articulated cars can negotiate sharper curves than a rigid body car. This is not true. They are limited in length primarily due to the fact that the lateral clearances required in curves increase dramatically as the distance between the trucks increases. If lateral clearances are not an issue, rigid body cars are a practical alternative that can be appreciably cheaper to procure and maintain than articulated cars of similar capacity. In North America, modern non-articulated cars are used only in Philadelphia, Buffalo, and Toronto.

Table 2.1 Contemporary Light Rail Vehicle Characteristics Matrix

City	Vehicle Manufacturer and Model	Empty Vehicle Weight (kg)	Articulated/ Non-Articulated	Truck/Bogie Centers (mm)	Wheel Base (mm)	Wheel Diameter New/Used (mm)	Track Gauge Wheel Gauge Delta Δ
Baltimore	ABB Traction	48 526	Artic.	9,144	2,286	771	1,435 1,421 5 Δ13.5
Boston (3 Vehicles)	Boeing Vertol Kinki Sharyo #7 Breda #8	30,390 38,460 39,000	Artic Artic Double Artic.	7,010 7,137 7,351	1,855 1,905 1,900	660 new 660 711/660	1,455 1,427.2 Δ27.8
Buffalo	Tokyo Car	32,233	Single Unit. Rigid	11,024	1,880	660/610	1,432 1,414 5 Δ17.5
Dallas	Kinki Sharyo	49,900	Artic	9,449	2,083	711	1,435 1,409 0 Δ26
Denver	Siemens Duewag SD 100	40,000	Artic.	7,720	1,800	720/660	1,435 1,413.9 Δ21.1
Los Angeles	Kinki Sharyo Blue Line Siemens Duewag Green Line	44,500	Artic Artic.	8,534 9,449	2,007 2,100	711/660	1,435 1,412 9 Δ21.1
Philadelphia City Division	Kawasaki	26,000 SE	Single Unit, Rigid	7,620 SE	1,900	660 new	1,581 1,578 Δ3
Philadelphia Suburban Division	Kawasaki	27,000 DE	Single Unit, Rigid	8,400 DE	1,900	660 new	1,588 1,578 Δ10
Pittsburgh	Siemens Duewag U2-A	40,000	Artic	8,950	2,100	720/670	1,587.5 1,577.5 Δ10
Portland (2 vehicles)	Bombardier Siemens Duewag SC 600	41,244 47,600	Artic. Artic.	9,040 10,515	1,900 1,800	711/660	1,435 1,421 Δ14
Sacramento	Siemens Duewag U2	47,160	Artic	7,723	1,800	720/660	1,435 1,414 Δ21
San Diego (2 vehicles)	Duewag Type U2 Siemens Duewag SD 100	32,600	Artic Artic.	7,720	1,800	720/660 720/660	1,435 1,414 Δ 21
San Jose	UTDC	44,724	Artic	8,611	1,905	711	1,435 1,416 Δ19
St. Louis	Siemens Duewag	40,993	Artic.	9,677	2,100	711/660	1,435 1,418 Δ17
San Francisco	Boeing Vertol Breda	30,390 36,200	Artic. Artic.	7,010 7,315	1,855 1,900	660 711	1,435 1,425.5 Δ9.5
Toronto (2 vehicles)	UTDC Hawker Siddley	22,685 36,745	Single Unit, Rigid Artic.	7,620 7,620	1,829 1,829	660/610 660/600	1,495 0 1,492 5 Δ2.5
Calgary	Duewag Type U2	32,600	Artic.	7,720	1,800	720/660	1,435 1,429 Δ6
Edmonton	Duewag Type U2	31,600	Artic	7,720	1,800	720/660	1,435 1,418 Δ17

Articulated LRVs developed in order to improve the ratio of passengers carried per vehicle operator. By attaching two or more body sections together, the car capacity can be increased while maintaining the capability to negotiate sharp curves without excessive lateral clearance excursions. Where two body sections meet, a turntable and bellows arrangement connects the sections, allowing free passage for passengers. Each LRV manufacturer has devised its own specific design for such articulation joints. In some cases, particularly in Europe, multiple body sections have been joined in double, triple, and even quadruple arrangements to form multi-articulated cars.

More recently, European manufacturers have created a variety of modular designs, particularly for low-floor cars. Typically, these designs include separate modules for cab, door, and body sections. They are joined in both rigid and articulated arrangements, allowing a vehicle to be tailored to meet a range of curve radius requirements. Low-floor LRV designs may incorporate stub axles, independent wheels, small trucks, small diameter wheels, hub-mounted motors, body-mounted motors, vertical drives, and a variety of other unique technological solutions that permit vehicles to incorporate very low floors

2.3 VEHICLE CLEARANCE

Clearance standards for various types of railroad vehicles are well documented by the use of graphics or "plates". One standard is the common Plate "C." Any car whose dimensions fit within the limits established on Plate C can travel virtually anywhere on the North American railroad system. Transit systems do not share this standard. Therefore, vehicle manufacturers must develop clearance plates based on the characteristics of the existing system for which the car is intended. While manufacturers can, in theory, build cars to any dimension, it is usually more economical to choose vehicles that are already engineered or in production. Therefore, the facility designer of a new system should establish a clearance envelope that accommodates vehicles from several manufacturers to maximize opportunities for competitive bidding.

The clearance diagram must consider both the vehicle's static outline and its dynamic outline. The static outline is the shape of the car at rest. The dynamic outline includes the allowable movement in the suspension system, end overhang, and mid-ordinate overhang. The manufacturer develops the dynamic outline for each type of transit vehicle. To establish clearances along the right-of-way, a vehicle dynamic clearance envelope must also be developed. Using the vehicle dynamic outline along with the associated track components, track tolerances, wear limits of the components, and a clearance zone with a safety factor of 50 millimeters (1.968 inches), the dynamic vehicle clearance envelope can be established. For additional information on vehicle clearances, refer to Section 3.4 of this handbook.

2.3.1 Static Outline

The static outline of an LRV is its dimensions at rest, including elements such as side view mirrors. The resulting diagram will show the minimum overhang on tangents and curves. The dynamic outline of the car is more significant to the track designer.

2.3.2 Dynamic Outline

The dynamic outline of an LRV describes the maximum space that the vehicle will occupy as it moves over the track The dynamic outline or "envelope" includes overhang on curves, lean due to the action of the vehicle suspension and track superelevation, track wear, wheel/track spacing, and abnormal conditions that may result from failure of suspension elements (e g. deflation of an air spring).

2.3.2.1 Car Length: Over Coupler Face and Over Anticlimber

When considering the length of a light rail vehicle, it is important to distinguish between the actual length of the car body over the anticlimbers and its length over the coupler faces.

- Over Coupler Face—The coupler is the connection between LRVs that operate together. It extends beyond the front of the car structure. The length over the couplers becomes a consideration for determining the requisite length of facilities such as station platforms and storage tracks.
- Over Anticlimber—The anticlimber is the structural end of the car. As its name implies, it is designed to reduce the possibility of one car climbing over an adjacent car during a collision. The length of the vehicle over the anticlimber is used to determine clearances.

2.3.2.2 Distance between Truck Centers

The distance between adjacent truck pivot points determines the overhang of a car's midsection for given track curvature.

2.3.2.3 Distance between End Truck and Anticlimber

This dimension and the car body taper determine the overhang of the car front for a given track curvature.

2.3.2.4 Vehicle Components Related to Dynamic Positions

- Primary/Secondary Suspension Systems
- Maximum Lean/Sway
- Maximum Lean due to Total Failure of All Truck Components
- Wheel Flange Wear

2.3.2.5 Track Components Related to Dynamic Positions

- Track Surface—Maximum Cross-Level Limits and Lateral Tolerance of Rails
- Rail Headwear and Side Gauge Face Wear
- Track Superelevation
- Wheel Gauge to Track Gauge Lateral Clearance
- Truck/Wheel Set (Axle) Spacing

2.3.2.6 Ensuring Adequate Vehicle Clearance

Where facility clearance restrictions exist, the track designer should coordinate with the vehicle and structural designers to ensure that adequate car clearance is provided. Vehicle dynamics are governed by the car's suspension system(s) and, therefore, indirectly by numerous factors of track and vehicle interaction. For multiple-track situations, multiple clearance envelopes must be considered. Overlapping must be avoided. The resulting requirements will dictate minimum track centers and clearances for tangent and curved track, including tolerances and safety factors.

2.3.2.7 Pantograph Height Positions

Outside Height: Roof and Pan Lock-Down—Should include all roof-mounted equipment.

Roof — The roof of an LRV is typically curved, with the highest dimension at the car centerline. However, the LRV pantograph establishes the maximum car height.

Pantograph Operation — Light rail facility designers are typically interested in the absolute minimum clearance between top of rail and an overhead obstruction, such as a highway bridge. This dimension must accommodate not only the pantograph when operating at some working height above lock-down, but also the depth of the overhead contact wire system. The minimum pantograph working height above lock-down includes an allowance for pantograph "bounce" so that lock-down does not occur accidentally. Maximum pantograph height is the concern of vehicle and overhead catenary system (OCS) designers, unless the light rail guideway must also accommodate railroad freight traffic and attendant overhead clearances. If railroad equipment must be accommodated, the clearance envelope will be dictated by Association of American Railroads (AAR) plates, which do not include clearance for the overhead catenary system. Additional clearances may be required between the underside of the contact wire system and the roof of any railroad equipment in order to meet electrical safety codes.

2.4 VEHICLE-TRACK GEOMETRY

The most demanding light rail transit alignments are those running through established urban areas. Horizontal curves must be designed to suit existing conditions, which can result in curves below a 25-meter (82-foot) radius. Vertical curves are required to conform to the existing roadway pavement profiles, which may result in exceptionally sharp crest and sag conditions.

LRVs are specifically designed to accommodate severe geometry by utilizing flexible trucks, couplings, and mid-vehicle articulation. Articulation joints, truck maximum pivot positions, coupler-to-truck alignments, vehicle lengths, wheel set (axle) spacing, truck spacing, and suspension elements all contribute to vehicle flexibility.

The track designer must take the vehicle characteristics defined below into account in developing route designs. The values associated with these characteristics are furnished by the manufacturer. For vehicles supplied for existing systems, the vehicle manufacturer must meet the minimum geometrical requirements of the system.

2.4.1 Horizontal Curvature—Minimum Turning Radius of Vehicle

The minimum turning radius is the smallest horizontal radius that the LRV can negotiate. The value may be different for a single versus coupled LRVs or for a fully loaded LRV versus an empty one.

2.4.2 Vertical Curvature—Minimum Sag and Crest Curves

The minimum vertical curvature is the smallest vertical curve radius that the LRV can negotiate. The maximum sag and crest values are typically different, with the sag value being more restrictive. Vehicle builders describe vertical curvature in terms of either radius of curve or as the maximum angle in degrees through which the articulation joint can bend. The trackway designer must relate those values to the parabolic vertical curves typically used in alignment design.

2.4.3 Combination Conditions of Horizontal and Vertical Curvature

The car builder may or may not have a graph that displays this limitation. If a route design results in significant levels of both parameters occurring simultaneously, the design should be reviewed with potential LRV suppliers to establish mutually agreeable limits.

2.4.4 Vertical Alignment—Maximum Grades

The maximum allowable route grade is limited by the possibility that the LRV could stall or the traction motors overheat. This is the steepest grade the LRV can negotiate. A short grade that the LRV enters at speed should not be a problem up to about 6%. Above that the operational requirements should be reviewed. Grades of up to 10% are possible. At grades between 6% and 10%, wheel-to-rail slippage may occur in poor conditions, such as when ice or wet leaves are on the rail. This may result in wheel flats during braking or rail burns during acceleration.

2.4.5 Maximum Allowable Track Vertical Misalignment

Truck equalization refers to the change in wheel loading that occurs when one wheel moves above or below the plane of the other three wheels on a two-axle truck. If a wheel is unloaded significantly, it may climb the rail and derail. LRV truck equalization must be compatible with the expected track vertical surface misalignment to prevent conditions that can cause a derailment.

2.4.6 Ride Comfort and Track Geometry

2.4.6.1 Track Superelevation

Passenger safety and ride comfort limit vehicle speed on sections of curved track. Experience has shown that safety and comfort can be achieved if vehicle speed is limited such that 75 to 115 mm (3 to 4.5 inches) of superelevation is required in the outer rail to achieve equilibrium (a balanced condition) on transit track. Equilibrium exists when loads on the inner and outer rails are equal and the centrifugal force on the car body and the passengers is in balance with the superelevation of the track. Track designers often limit actual superelevation and permit an unbalanced condition where the forces on vehicles and passengers are not equal. Unbalanced superelevation results in an unbalanced amount of lateral acceleration that the passenger feels. The standard limit is 76 mm (3 inches) of unbalanced superelevation which is equal to about 0 1 g. Chapter 3 elaborates on the formulas used to establish the amount of superelevation for both actual and underbalanced conditions.

2.4.6.2 Lateral Acceleration on Track Curves

Ride comfort is an important and very complex issue. Acceleration is a good measure of ride comfort and is a criterion for ride comfort on track curves. The rate of change of acceleration (jerk) is another important criterion. Industry standards have established that a lateral acceleration of 0.1 g can be tolerated with comfort. Chapter 3 elaborates on formulas used to establish the spiral criteria considering lateral acceleration.

2.4.6.3 Transition Spirals on Track Curves

A proper transition curve between the tangent track and the circular portion of the track curve is a recognized requirement for a smooth, comfortable ride on track curves. The change from no curve to a given constant curvature must be made gradually so that lurching does not occur at the entrance and exit of the curve. The usual method is to introduce curvature and superelevation in the transition curve uniformly along the curve.

Since the centrifugal force is inversely proportional to the radius of the curve and the superelevation for a given speed, both radius and superelevation change at a linear rate. Thus, lateral acceleration increases at a constant rate until the full curvature of the circular portion of the curve is reached, where the acceleration remains constant until the exit spiral is reached.

As a guideline, the transit industry has established 0.03 g per second as the desired maximum rate for change of acceleration. As stated previously, constant lateral acceleration in the central part of a track curve is comfortable at 0.10 g. Therefore, if the allowable maximum acceleration in the circular curve is 0.10 g and the rate of attainment is 0.03 g per second the time the train traverses the spiral must be no less than:

$$\frac{0\ 10g}{0.03g/sec} = 3.33 \text{ seconds}$$

The formulas presented in Chapter 3 are based on the 0.03 g per second rate of change of acceleration, with the provision to increase to 0.04 g per second when realigning existing tracks to fit built-in conditions.

The main objective is to design spirals that are sufficiently long enough to provide satisfactory ride comfort. Considering the average vehicle roll tendency and allowing for variability in tracks and vehicles, the rate of change of unbalanced lateral acceleration acting on the passenger should not exceed 0.03 g per second. In difficult situations, an acceleration of 0.04 g per second may be acceptable.

Passenger comfort on track curves is based on the theory that the spiral must be long enough so that excessive lateral force is not required to accelerate the vehicle up to the constant angular rotation of the circular curve. The spiral curve must be long enough, relative to the length of the vehicle, so that there is not excessive twisting of the vehicle, since twisting forces tend to produce derailments.

2.5 VEHICLE STATIC AND DYNAMIC FORCES

2.5.1 Static Vertical

The following parameters establish the LRV vertical wheel load on the rail head. The vehicle manufacturer generally provides these values.

2.5.1.1 AW0/AW1 Loads

AW0 is the total car weight, in a ready for revenue service condition, with no passengers on board. AW1 is the car weight with a fully seated passenger load, at 155 pounds per passenger.

2.5.1.2 AW2/AW3/AW4 Loads

AW2 (Design Load) is seated load plus standing passengers at 4 per square meter of suitable standing space. AW3 (Crush Load) is seated load plus standing passengers at 6 per square meter of suitable standing space. AW4 (Structure Design) is seated load plus standing passengers at 8 per square meter of suitable standing space. Since the seating and suitable standing space is a function of the vehicle design, the loading should be defined by the car builder.

2.5.1.3 Wheel Loading Tolerance (Car Level)

If exact wheel loadings must be known, the variations in each wheel load due to design and manufacturing tolerances must be considered.

2.5.1.4 Wheel Loading @ Maximum Stationary Superelevation, Considering Car Tilt and Uniform AW3 Load

Worst-case wheel/rail force is expected when a fully loaded car stops on a maximum superelevated track structure Car tilt will also add to the lateral and vertical forces on the lower rail.

2.5.1.5 Unsprung Weight (Truck Frame, Wheels, Axle, Bearings, and Portions of the Motor/Gear Units)

Unsprung weight is a significant contributing factor to dynamic track loading as these items are not isolated from the track by the car primary suspensions.

2.5.1.6 Truck Weight

Truck weight and yaw inertia will affect rail forces on curved track. Total truck weight will also affect dynamic forces as only the car body is isolated by the truck secondary suspensions.

2.5.1.6.1 Motorized Trucks

Motorized trucks (typically at the ends of the car) may have either one monomotor or two motors that drive both axles, along with gear units that connect the motors to the axles. The motors may be either DC or AC design depending on the vehicles control system package. Newer designs may have unique wheel and drive support systems that do not resemble traditional truck designs.

2.5.1.6.2 Non-Motorized Trucks

All trucks under a specific LRV will not have the same mass or the same inertia. Non-motorized trucks will not have motors and gear units, but may have axle-mounted disc brakes. They are typically located under the articulation joints of LRVs. On some vehicles, the wheels may be independently mounted rather than configured as a conventional truck.

2.5.1.7 Load Leveling

To meet ADA car threshold-to-platform alignment standards, track and platform designers must also consider the accuracy of car leveling systems that compensate for variable passenger loading. Load leveling can be provided by the secondary air springs or hydraulic actuators. For ADA requirements see Section 2.8 herein.

2.5.2 Dynamic Horizontal/ Longitudinal

The following parameters establish the maximum forces along the direction of the rails.

2.5.2.1 Maximum Acceleration

The maximum car acceleration provided by the car propulsion system is the resulting force at the wheel tread to rail head interface. The amount of adhesion is the measure of the force generated between the rail and wheel before slipping. A typical 4.8 kilometer per hour per second (3 miles per hour per second) acceleration rate is equivalent to a 15% adhesion level, if all axles are motorized. For a typical LRV with four of six axles motorized, the adhesion rate is 22.5%.

2.5.2.2 Maximum Deceleration (Wheels)

The maximum car deceleration rate is established by the retarding force at the wheel tread. The deceleration force can be the result of a combination of disc brakes, wheel tread brakes, and traction motor electrical brakes, either dynamic or regenerative.

2.5.2.3 Maximum Deceleration (Track Brakes)

Deceleration force is generated by electromagnetic brakes applied to the rail head, in addition to that produced at the wheel. This force is developed at the track brake-to-rail head interface and can provide an additional 4.8 kilometers per hour per second (3 mphps) of deceleration.

2.5.2.4 Tolerances

All acceleration and deceleration values also have tolerances that are due to many factors. The major factors for acceleration tolerance are traction motor tolerances, actual wheel diameter size, and generation and interpretation of master controller commands This tolerance may range from ±5 to 7%.

All actual deceleration values are dependent on friction coefficients as well as the above issues. The expected tolerance for friction and track brakes should be obtained from the supplier

2.5.2.5 Maximum Train Size

Acceleration and deceleration forces are applied per car. Therefore, the total rail force per train will depend on the maximum train size. If more than one train can be on common rails at one time, this should also be considered.

2.5.2.6 Load Weight

If the LRV has a load weight function, the acceleration and deceleration forces will be increased at car loadings above AW0, to some maximum loading value. These values should be defined to establish maximum longitudinal track force.

2.5.2.7 Sanding

Car sanders apply sand to the head of the rail in front of the wheel to obtain a higher adhesion coefficient. Sanding in specific locations has a fouling effect on track ballast that should be considered.

2.5.3 Dynamic Vertical

Determination of total track force is a complex issue that depends on LRV design features. Typically the vehicle total weight is increased by a factor to include dynamic loading effects. The characteristics of the LRV suspension system should be defined by the manufacturer, who should also provide the dynamic load factor to the track designer.

2.5.3.1 Primary Suspension

Primary suspension provides support between the truck frame and the axle journal bearings. It is the first level of support for the bearings above the wheel set.

2.5.3.1.1 Spring Rate

Spring rate is the force per travel distance for the coil or chevron primary springs. This relationship may be non-linear for long travel distances. The equivalent vertical, longitudinal, and lateral spring rates will be different.

2.5.3.1.2 Damping Rate

Damping rate is the "shock absorber" action that provides a force proportional to the velocity of the spring movement. It is designed to minimize oscillation of the springs/mass system.

2.5.3.2 Secondary Suspension

Secondary suspension supports the car body on the truck and controls the range of car body movement with relation to the truck. The suspension and track alignment basically establish the LRV ride quality. The secondary springs can be either steel coils or air bags.

2.5.3.2.1 Damping Rate

Damping rate is optimized for ride quality. With an air bag system, orifices in the air supply to the air bags can adjust the damping rate.

2.5.3.2.2 Yaw Friction

Yaw is the amount of rotation of the truck with relation to the car body Some yaw is normal on curved track. The truck design and materials used will establish the friction force that restrains truck swivel. Yaw contributes to lateral track forces, which can produce conditions where the wheel climbs over the rail head. The design of related friction surfaces should be such that the friction factor remains constant as service life increases.

2.5.3.3 Maximum Speed

The operating speed limit for all track considers passenger comfort and safety. This criterion should be coordinated with the car design. Civil speed limits are set by determining the maximum rate of lateral acceleration that passengers can comfortably endure. This is usually in the range of 0.1 g, which establishes the level of unbalanced superelevation on curves. Speed limits on curves are then established based on the actual and unbalanced superelevation.

2.5.3.4 Car Natural Frequency

The natural frequency of cars should be coordinated with the natural frequency of civil structures such as bridges or elevated guideways. Trucks and car bodies each have different natural frequencies that should also be considered. Also, car loaded weight affects the car body's natural frequency. Therefore, natural frequency should be defined at car weight extremes, AW0 to AW3.

2.6 VEHICLE WHEEL GAUGE/TRACK GAUGE/ WHEEL PROFILE

2.6.1 Track Gauge

The American Railway Engineering Maintenance of Way Association (AREMA) standard track gauge for railways shown on Portfolio Plan 793-52 is established at 1,435 millimeters (56 5 inches) New light rail transit systems generally adopt railway gauge as standard. The use of AAR and AREMA standards facilitates procurement of track materials and track maintenance. For additional information on track gauge refer to Chapter 4

2.6.2 Vehicle Wheel Gauge

AAR standard wheel gauge for railroad cars per AREMA Portfolio Plan 793-52 is established at 1,414.5 millimeters (55.7 inches). The inside gauge of flanges (wheel back-to-back distance) considering the common 29.4-millimeter (1.2-inch) wide wheel flange is 1,355.7 millimeters (53 4 inches). Transit standard wheel gauge generally conforms to track gauge with a minimal clearance, resulting in wheel gauge width of 1,429 millimeters (56.25 inches). Vehicle wheel gauge is a very important interface issue that must be addressed jointly by vehicle and track designers.

2.6.3 Wheel Profiles—United States, Canada, Europe

Wheel profile is one of the most critical vehicle parameters to consider in track design, since the wheel is the primary interface between the vehicle and the track structure. The wheel profile must be compatible with the rail section(s); the special trackwork components, including switch points and frog flangeways or moveable point sections; the guard rail positions to protect special trackwork components; and the guarded track restraining rail positions on shorter or sharp radius track curves.

Once approved, any changes to the wheel profile (especially tread and flange width) must be evaluated by both vehicle and track designers. In more than one instance, the wheel profile has been altered at the last minute without informing the track designer, resulting in unsatisfactory performance of both the track and vehicle. Selected wheel profiles are shown below [1]:

USA.	Figure 2.6.1	Baltimore
		Los Angeles
		Boston (2)
		Pittsburgh
		Dallas
		Portland (2)
		Denver
		Sacramento
	Figure 2.6.2	San Diego
		San Francisco (2)
		San Jose
		Philadelphia
		St. Louis
		Toronto
		Calgary
		Edmonton
Europe	Figure 2.6.3	Koln
		Zurich
		Karlsruhe

A cursory review of the selected profiles (**Figures 2.6.1 to 2.6.3**) clearly indicates that transit vehicle designers virtually always utilize unique wheel profiles, unlike the railroad industry, which has adopted standard profiles

In 1928, the AAR established the recently outdated AAR standard wheel profile as shown on AREMA Drawing 793-52. In 1991, the AAR revised this standard wheel profile to the current AAR-1B narrow flange profile.[2] These two wheel profiles are shown in **Figure 2.6.4**.

Many transit agencies have adopted a "worn wheel" design, featuring wheel contours that approximate the template to which railway wheels wear in service. These designs are intended to:

- Reduce wheel and rail wear
- Reduce likelihood of derailment under adverse operating conditions
- Enhance stable performance over the nominal range of speeds
- Provide reasonable contact stress characteristics

Tests by the AAR at the Transportation Test Center in Pueblo, Colorado have shown that the AAR-1B wheel provides:

- A lower lateral over vertical (L/V) ratio in a 233-meter (764-foot) radius curve than the previous AAR 1:20 profile
- A lower rolling resistance than the previous AAR 1:20 profile.
- Lower critical hunting speeds than the new AAR 1.20 profile

New transit agencies must review the advantages of adopting either the AAR-1B wheel profile or a similar worn wheel design adapted to the local needs of the transit system, considering factors such as the overall tread width, wheel diameter, and flange width and depth.

2.6.4 Wheel/Rail Profiles

Wheel profile is a flexible design decision, drawn from the different profile sections used throughout the transit industry. The same

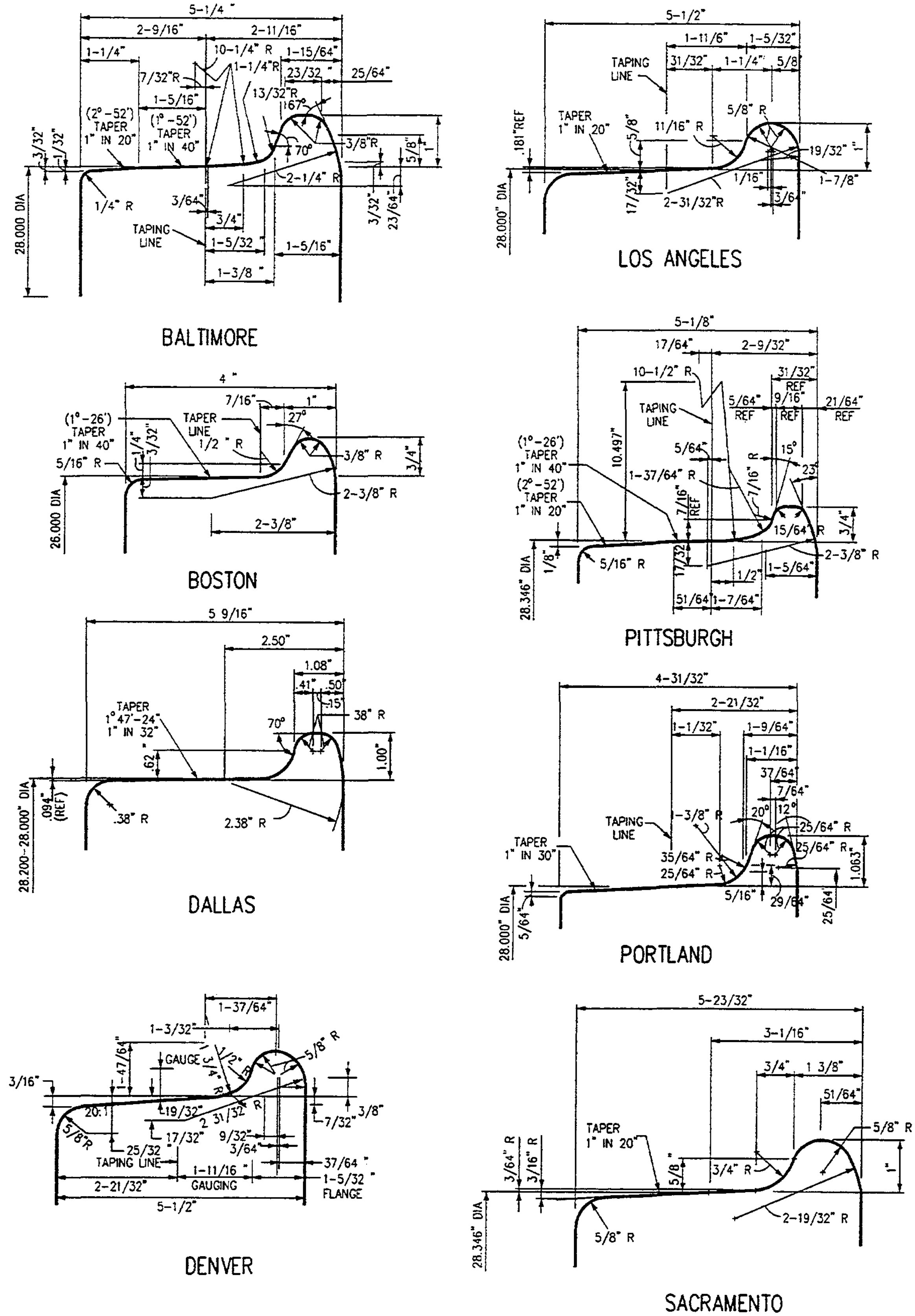

Figure 2.6.1 Wheel Profiles (U.S.)

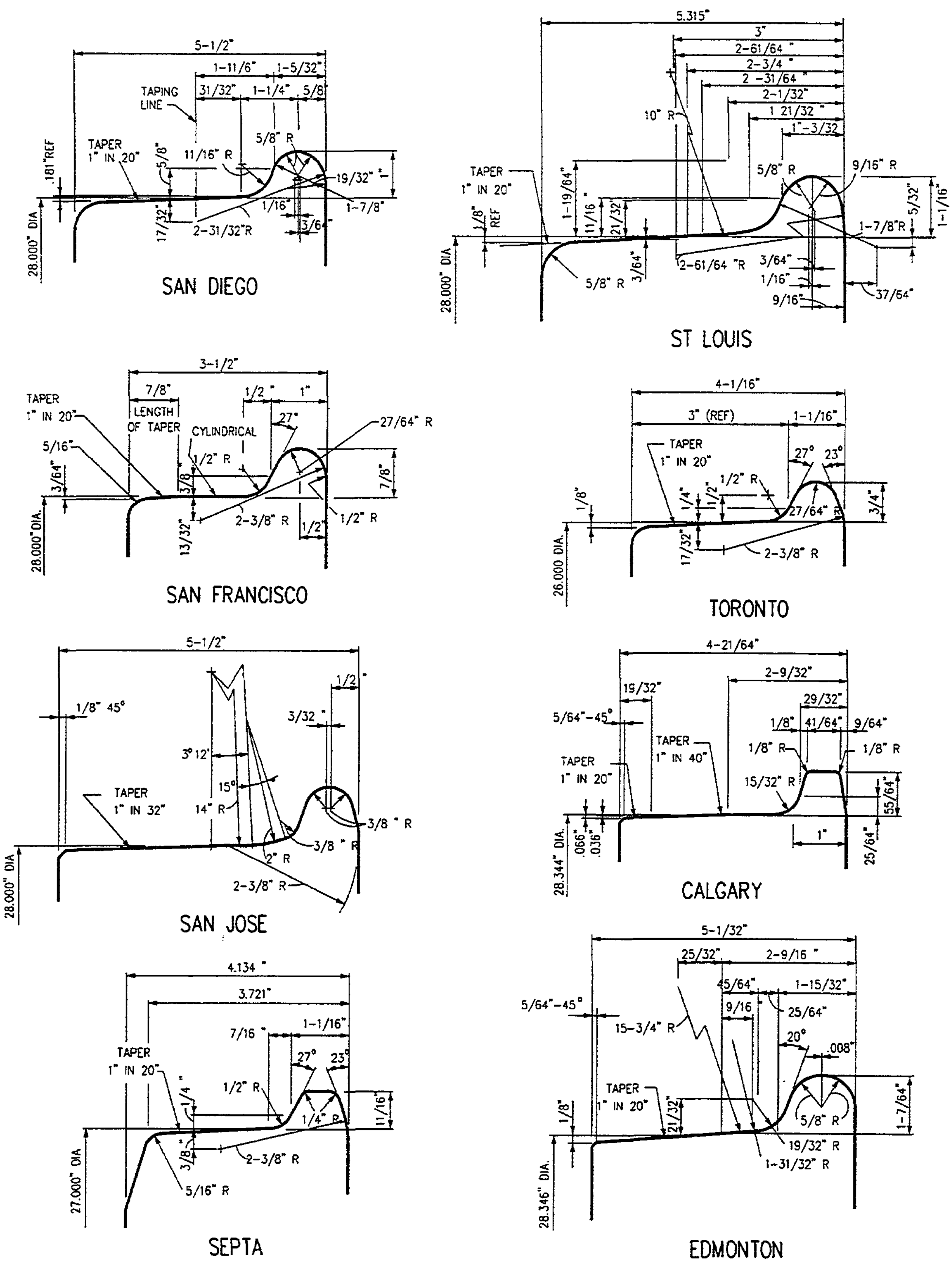

Figure 2.6.2 Wheel Profiles (U.S./North America)

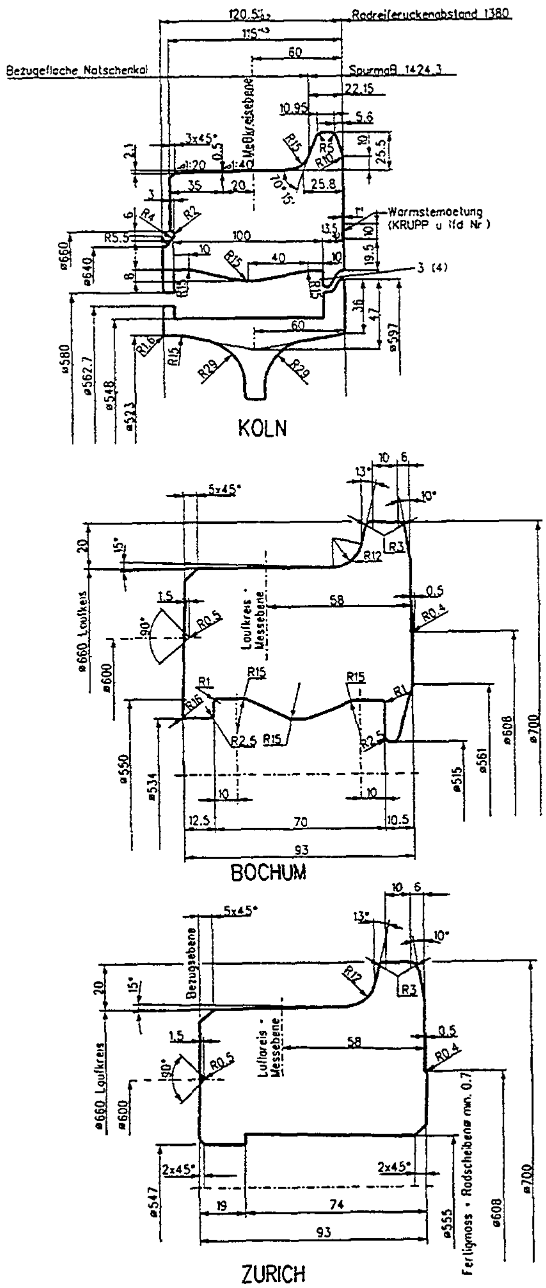

Figure 2.6.3 European Wheel Profiles

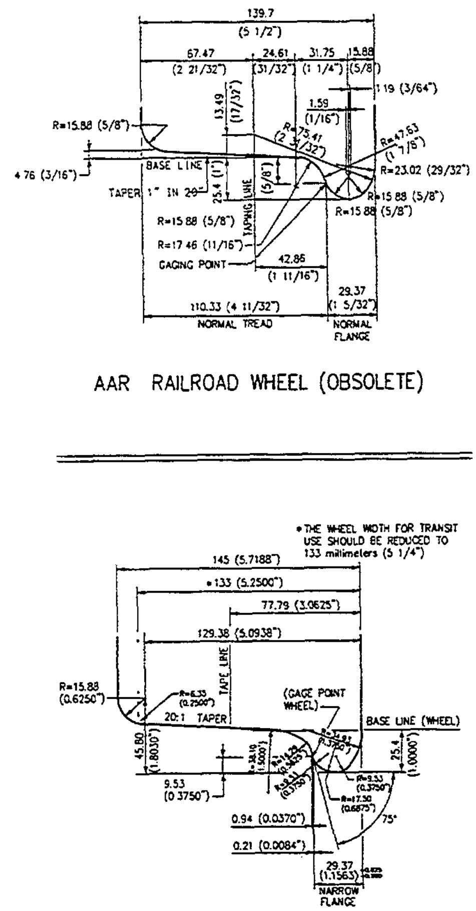

Figure 2.6.4 AAR Wheel Profiles

flexibility is not provided in the selection of standard rail profiles. Only a few standard rail sections exist for use by the transit industry. However, wheel and rail profiles must be compatible, which means that the wheel profile should conform to the rail head profile.

As with wheel profiles, the majority of the research and development on rail head profiles and rail profile grinding has been undertaken by and for the railroad industry. The transit industry can also benefit from this research. However, recommendations for heavy haul railroads may not be entirely applicable to the transit industry. A light rail vehicle weighs (AW0) approximately 44,000 kilograms (97,000 pounds). A loaded freight car weighs as much as 152,000 kilograms (335,000 pounds). This represents a significant difference in wheel loads of 5,500 kilograms (12,100 pounds) and 19,000 kilograms (41,900 pounds) for LRVs and freight cars, respectively. Obviously, rails used in transit service will not be subjected to wheel forces of the magnitude exerted by freight cars. Therefore, theories of rail gauge corner fatigue, high L/V ratios, and the threat of rail rollover that pertain to freight railroads may not be fully applicable on a transit system.[3] The contact forces at the rail gauge corner on curved tracks are usually twice as large as those between the rail crown and wheel tread.

To reduce contact stresses at the gauge corner and gauge side rail base fastening, it is important that the wheel/rail profile be compatible. The wheel profile is conformed to the rail profile if the gap between the wheel and rail profile is less than 0.5 millimeters (0.02 inches) at the center of the rail (in single-point contact) or at the gauge corner (in two-point contact).

Figure 2.6.5 illustrates various transit rail sections used on contemporary LRT systems in conjunction with the obsolete AAR wheel profile and the new AAR-1B wheel profile. The obsolete AAR wheel profile is included to show a non-conformal two-point contact wheel/rail relationship that transfers the vertical load from the gauge corner toward the centerline of the rail. This combination, shown in Figure 2.6.5 A and C, reduces the wheel radius at the contact location which is detrimental to steering and introduces accelerated gauge face wear. A secondary distinct wheel/rail profile condition, shown in Figure 2.6.5 E, is the AAR-1B wheel superimposed on the Ri59N girder groove rail. Although the wheel is conformed to the rail head, a pronounced one-point contact materializes. Although excellent for steering, the contact stresses at the gauge corner may prove to be too high and detrimental to the rail, leading to fatigue defects. Recent revisions to the rail head profile that alter the head radius introduce a surface cant in the head, and increase the gauge corner radius of the Ri59 and Ri60 rail to 13 millimeters (0 5 inches) were undoubtedly undertaken to improve the wheel-to-rail contact points.

The combinations of wheel and rail profiles shown in Figure 2.6.5 illustrate the various interface conditions generated between the wheels and rails. The old AAR wheel profile is obsolete for use on main line railroads. However, some existing transit systems may utilize this profile. To improve wheel/rail interface contact, alternate wheel shapes may be considered. During the early design stage of new transit systems, transit wheel profiles should be considered that match or conform to the rail section(s) to be used on the system. In the process of wheel design, the design engineer must consider the rail sections and the rail cant to be selected. For additional information on rail sections, refer to Section 5.2.2 of this handbook. For additional information on rail cant selection and benefits, refer to Section 4.2.4.

Many transit properties have adopted the combination of transit wheel/rail profiles proposed by Prof. Herman Heumann [4], where the wheel profile conforms to the rail head profile. This design emphasizes single point contact which improves the difference in radius between the two rail/wheel contact points leading to improved wheel set (axle) curving. Improved wheel/rail contact at the gauge corner provides improved steering and less gauge face contact. Figure 2.6.5 F illustrates a recommended transit wheel profile taking advantage of the following design concepts·

- The wheel profile is designed to conform to selected rail sections (where the transit system will not share track with freight cars). Heritage or historical vehicles to be used on the transit system for special occasions must be considered.

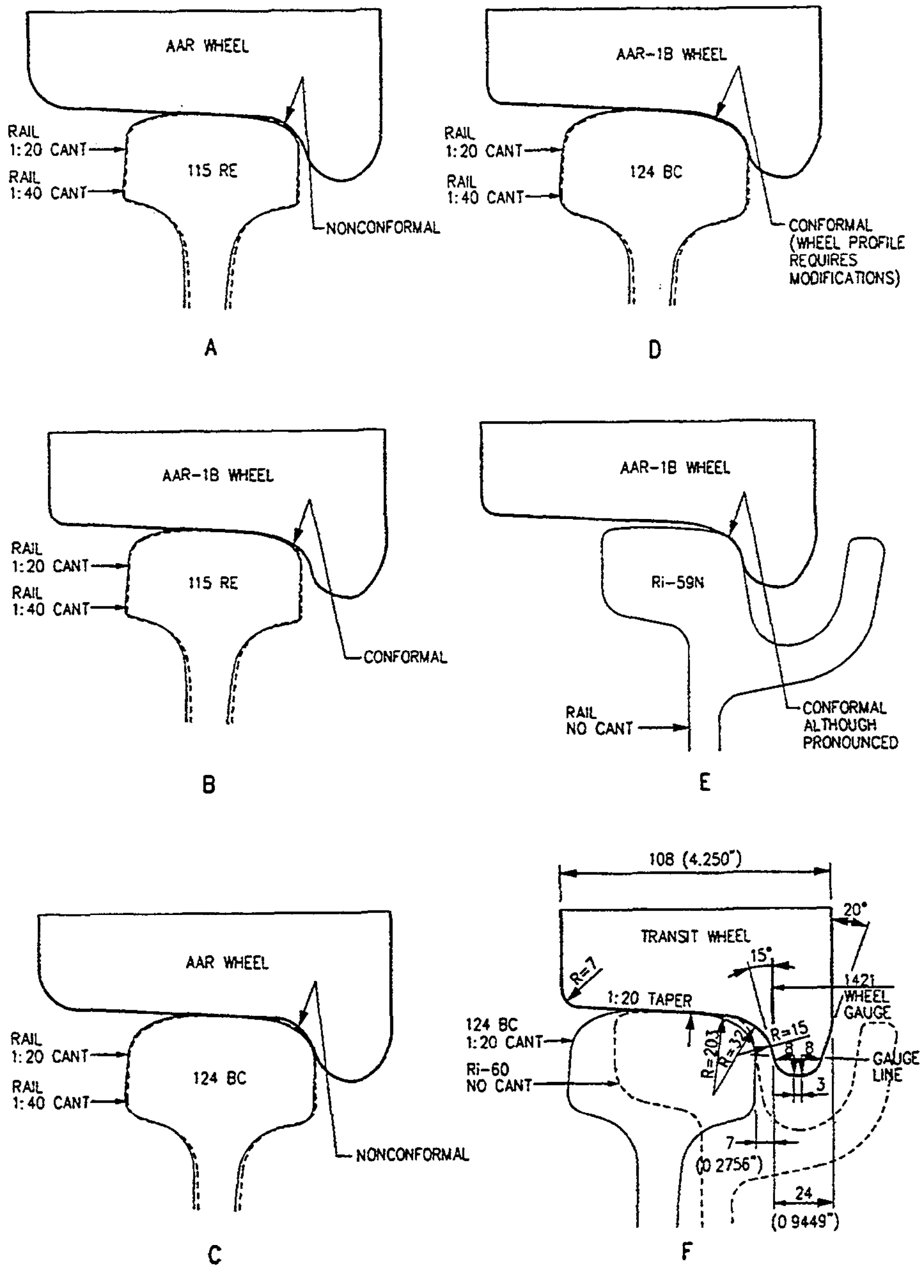

Figure 2.6.5 Wheel-Rail Interface

- The selected wheel width is 108 millimeters (4.2 inches) to reduce wheel weight and projection of wheel beyond the rail head on the field side. Special trackwork switch mates, turnouts, and crossing (diamond) frogs must be flange bearing to conform to the wheel width. The width of the wheel is 18 millimeters (0.7 inches) wider than the normal 89-millimeter (3.5-inch) width. This provides additional wheel tread for occasional wide track gauge locations in sharp curves to specifically halt the vertical wear step in the head of rail produced under these operating conditions.
- Tee rail profile is 124 BC to provide a preferred rail head profile with improved radii and additional steel in the head area.

- Girder groove rail section (Ri59N) is used to provide a narrow flangeway and increased tram or girder lip. (Note the wheel gauge must be transit width or 1,421 millimeters).
- Rail cant is 1.20 to improve wheel/rail contact location in curved track.

2.6.4.1 Wheel Profile—Widths and Flangeways

The wider clearance between AAR wheel gauge and standard track gauge governs the width of the wheel tread and affects the width of the wheel tread supporting surface through special trackwork. The larger wheel-to-rail clearance requires a wider flangeway opening through frogs and the corresponding guard rail flangeway. The wider flangeways promote increased lateral wheel positions resulting in less wheel tread contact when the wheels are furthest from the gauge face of a frog. This condition promotes rapid deterioration of the critical wing rail frog point due to improper tread support transfers between the two components. Wheels traversing the frog point area in a facing point lose the wing rail-wheel support surface resulting in premature transfer of wheel load to the frog point. This early transfer causes the load to bear on too narrow a frog point, producing frog point vertical head crushing.

Placing the wheel flange further from the gauge face of rail requires a wider wheel tread. The wider wheel tread increases the weight of the wheel, thereby increasing the unsprung mass of the truck. A narrower wheel profile of 133 millimeters (5.25 inches) with the standard AAR-1B flange profile is the recommended maximum width for transit systems sharing track with freight cars, or for special trackwork sections that do not employ a flange-bearing frog design. This width includes a 6-millimeter (0.25-inch) radius at the field side of the wheel tread. Narrower wheels used with standard railroad flangeways and wheel gauges will undoubtedly lead to improper wheel traverse through special trackwork components.

2.6.4.2 Wheel Profile—Flange Configuration

The wheel flange is an extremely important component when considering wheel/rail design compatibility. The width of the flange should be selected based on the standard girder groove or guard rail section to be used in embedded track. The standard rail sections currently available (Ri59N, Ri60N, etc.) restrict the width of the wheel flange. If only tee rail is to be used on the transit system, the flange width can be more flexible. A wheel flange with side slopes approximately 70° from vertical has been the focus of much design discussion based on the L/V wheel forces and friction levels, with rail head wear leading to potential wheel climb. The proposed wheel is based on Professor Heumann's 70° flange design. The radii at the outside edges of the wheel flange should be relatively curved, in lieu of a squarer configuration which, when worn, could lead to sharp flange corners that perpetuate potential wheel climb. The flange edge, or bottom, on a majority of transit wheels is totally curved.

Comparing standard American and European wheel profiles (Figures 2.6.1, 2.6.2 and 2.6.3), it is apparent that the European wheel design with flat wheel flanges considers flange bearing a standard practice. The majority of transit agencies in North America have not featured a flat wheel flange design, even though a limited amount of flange bearing is used on some systems. Philadelphia, Pittsburgh, and Calgary are the only North American transit agencies using a pronounced flat wheel flange design. The recommended wheel design proposes a limited flat section on the flange specifically to be compatible with flange-bearing special trackwork components.

As a guideline for improved wheel-to-rail and special trackwork performance, the wheel flange profile should be 25 millimeters (1 inch) high nominally and definitely not less than 22 millimeters (0.86 inch).

2.6.4.3 Wheel/Rail Wear Interface

As stated previously, transit systems generally rely on railroad research data for analyzing conditions when considering issues of mechanical and track maintenance, vehicle operation, and safety. Understandably, intensive research by new transit systems is not economically practical. However, conditions on railroad trackage are often different than conditions on transit trackage. Conclusions based on railroad research should be used only as a basis for clarifying and resolving transit-related conditions between vehicle and track. The following information discusses AAR research and development of the wheel/rail interface.[5]

2.6.4.3.1 Hollow Worn Wheels

AAR investigations of rail rollover derailments have ascertained that, under certain conditions, a combination of hollow worn wheels and heavy rail gauge corner grinding can generate large gauge spreading forces. The interfacing of the wheel/rail profiles can contribute to:

- Rail spalling and wear
- Wheel shelling and wear
- Damage to special trackwork
- Rail rollover and flange climb derailments
- Train resistance

The wheel and rail profile system can be considered a fundamental component of a rail vehicle's suspension system, providing proper guidance along the track.

Generally, the wheel/rail profiles have been designed and maintained separately, with the consequence that some practices may benefit one discipline but degrade overall performance One such example is the practice of grinding gauge corner relief on the high rail in curves and applying lubrication. This practice was commonly thought to reduce rail wear and extend rail life. However, investigations now indicate that this procedure may actually accelerate rail wear in curves and degrade railcar steering to the point that wheel flange forces are substantially increased. Wheel/rail conformance and maintaining that conformance on transit system track is essential in restricting these degradations.[5]

2.6.5 Profile Rail Grinding vs. Wheel Wear

Rail grinding procedures have received a substantial amount of attention in the railroad industry. The focus has been on grinding the high rail in curves to provide gauge corner relief. The theory was that avoiding overload of the gauge corner on the high rail would reduce internal rail defects. The other theory was that this relief grinding exacerbates rail and wheel wear, compared to more conformal rail profiles, by reducing the railcar steering forces and increasing the wheel flange forces.

To provide insight into the relative performance of various rail grinding practices, long-term rail grinding experiments were undertaken. New rails were installed in several curves and were being maintained using three different rail grinding practices:

- No grinding
- "Mild" high rail gauge corner relief
- "Moderate" high rail gauge corner relief

Transverse rail profiles and rail head heights were periodically measured to compare the relative wear rates in the three zones.

2.6.5.1 Wheel Profile Development

Figure 2.6.6 shows preliminary results of the rail grinding experiment cited above. The high rail gauge face wear rates are plotted for each practice. Clearly, the wear rate increased with the amount of gauge corner relief. It was established that new wheels with AAR 1:20 profiles experienced substantial wear when first put into service and that most worn wheels developed very similar profiles over time. To minimize wear on new wheels, the AAR developed a new standard wheel profile (AAR-1B) that was based on an "average" worn wheel shape (see Figure 2.6.4).

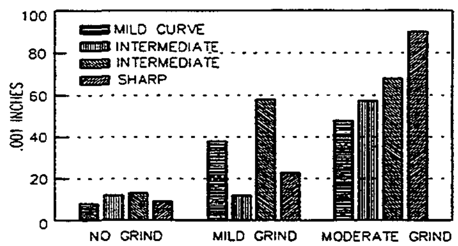

Figure 2.6.6 Preliminary High Face Gauge Wear Measurements

The implementation of the AAR-1B wheel profile has reduced the wear of new wheels. However, stricter new wheel profile maintenance practices are required to minimize deterioration of wheel profile performance from tread wear. For example, **Figure 2.6.7** shows the profile of a new AAR-1B wheel hollow worn from revenue service. Although the worn wheel tread appears to be excessively hollow, the wheel is not condemnable under current AAR limits.

The ability of worn wheels to properly guide, or steer, a railcar through curves is seriously compromised by excessive tread hollowing. The AAR has recently demonstrated that in a 233-meter (764-foot) radius track curve with heavy high rail gauge, corner grinding and wheel sets with hollow profile will actually produce forces that inhibit truck turning and cause trucks to warp.

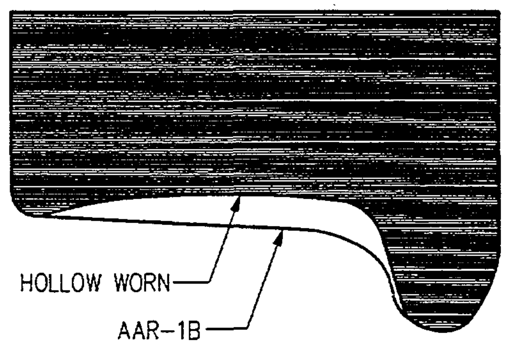

Figure 2.6.7 New AAR-1B and Hollow Worn Wheel

Truck warp occurs when the truck is skewed so much that its side frames rotate relative to the bolster in the vertical plane and both wheel sets develop large angles of attack relative to the rails. The large angles of attack from the wheel sets of a warped truck often generate large gauge spreading forces.

2.6.5.2 Wheel/Rail Interface Profiles and Potential Derailments

Wheel and rail profiles play major roles in flange climb and rail rollover or wide gauge derailments. The AAR recently performed tests to better understand the factors that influence the propensity of a wheel set to climb the rail. These factors include lateral and vertical wheel force ratios, wheel set angle of attack, wheel/rail flange contact angle, and friction.

The test demonstrations were conducted on a 233-meter (764-foot) radius track curve for the three different high rail profiles, as shown in **Figure 2.6.8**:

- "Heavy" gauge corner grinding
- "Light" gauge corner grinding
- No grinding

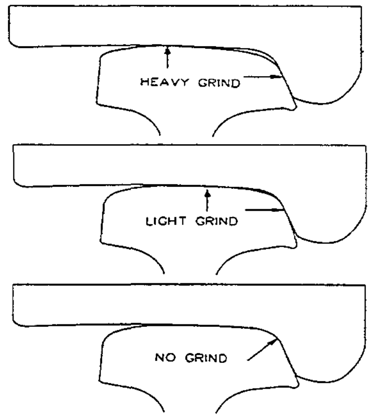

Figure 2.6.8 Three Rail Profiles Used in AAR Demonstration

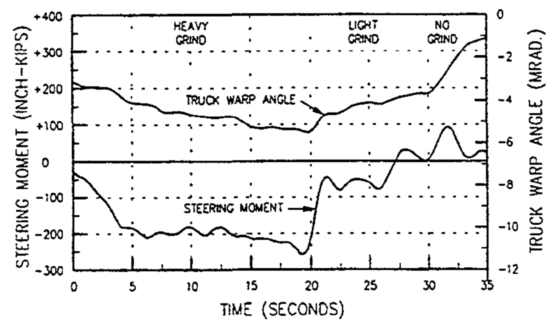

Figure 2.6.9 Truck Steering Moment and Warp Angle from Demonstration

A pair of instrumented wheel sets, with the hollow worn profiles shown in Figure 2.6.7, were used in the trailing truck of a 100-ton hopper car to measure the wheel/rail forces.

The primary measurements of interest were truck steering moments, truck warp angle, and wheel set lateral forces. Truck steering moments were measured to evaluate the steering quality of a particular wheel/rail profile combination. In **Figure 2.6.9** the bottom curve shows the truck steering moment through the three test zones when the running surfaces of the rails were dry and the gauge face of the high rail was lubricated. In the figure, a positive steering moment acts to steer the truck into the curve, while a negative steering moment acts to resist truck steering. The combination of hollow worn wheel profiles and heavy rail gauge corner grinding generated a large negative steering moment in the heavy grind zone. The steering moment improved dramatically in the mild and no-grind zones.

The large negative steering moment caused the test truck to warp in the heavy grind zone, as shown in the top curve of the figure. As the steering moment increased in the light and no-grind zones, the truck warp angle improved. At the point of maximum truck warp in the heavy grind zone, the test truck produced a trackside lateral gauge spreading force of 151,000 Newtons (34,000 pounds) Gauge spreading forces of this magnitude have the potential to cause wide gauge or rail rollover derailments in weak track under certain conditions.[5]

2.6.5.3 Special Trackwork and Hollow Worn Wheels

False flanges on hollow worn wheels cause excessive damage to switches, turnouts, crossing frogs, and grade crossings compared to properly tapered wheels. Hollow worn wheels increase noise and vibration due to excessive impacting of the false flange on the wing rails and wide special trackwork components.

European switch point design does not consider the raised switch point concept; therefore, the selection of a uniform or graduated design is not a concern. However, either raised switch point design, especially level switch point design, can best improve operations through the regular maintenance of wheel truing, eliminating the false flange and secondary batter caused by the false flange. The standards for vehicle wheel maintenance play an important part in the switch point design and must be considered when contemplating wheel special trackwork switch point interface.

For additional information on wheel false flange and special trackwork switch point design with raised switch points, refer to Section 6.5.3.

2.6.5.4 Truck Resistance with Hollow Worn Wheels

It was determined that trucks that warp in curves, so that both wheel sets run in flange contact with the high rail, have a higher rolling resistance than trucks that steer properly in curves. Also, trucks that exhibit a "diagonal" wheel wear pattern—two diagonally opposite wheels are worn hollow while the other two are not—might have an increased rolling resistance on tangent track because two diagonally opposite wheels would run in or near flange contact.

Test results indicate that, at 80 km/h (48 mph), the rolling resistance of the test truck increased in the curve from approximately 2600 to 7100 Newtons (600 to 1,600 pounds) when the wheel profile was changed from new to hollow worn.

Transit agencies generally include wheel truing machines in their requirements for maintenance facilities. Therefore, severely hollow worn wheels should not be a problem if conscientious wheel maintenance is practiced. Hollow worn wheels would also be a severe detriment to the surrounding surfaces in embedded track.

2.6.5.5 Truck Resistance—Alternate Approaches

The advantages of "radial" or "self-steering" trucks have been demonstrated in a variety of main line railroad and transit applications. These advantages usually appear as lower wear rates on both wheels and rails due to the decrease in overturning, creep, and climb forces being exerted on the running rails.

"Normal" trucks are configured as two parallel sets of wheels and axles locked in a rectangular frame. As this assembly travels through curves, the attempt by the inside and outside wheels to remain parallel results in significant forces being exerted by the wheels on the rails.

The wheels attempt to overturn the rails, climb the rails, and creep along the rails simultaneously.

Rail systems designers have recognized that if successful steerable trucks could be developed, rail and wheel wear could be reduced. A major problem in achieving a successful steerable truck or axle has been the difficulty in developing a system that not only permits steerability in curves, but also retains stability (i.e. does not "hunt") when traveling on tangent track.

The self-steering principle has been successfully implemented in main line diesel-electric freight locomotives using mechanical linkages that allow axle movement within the truck frame. Successful designs based on rubber/steel chevron primary suspension systems have been achieved on commuter, intercity, and high speed trains, notably in Sweden.

The rubber/steel chevron system has also been applied successfully to light rail vehicles both in Europe and the United States.

Some new design European vehicles, featuring 100% low-floor designs, are effectively eliminating the conventional "four-wheel" truck, as we have known it. Instead various types of single axles and independently mounted wheels are being utilized.

If a light rail system is proposed that will utilize radial steering or other unconventional designs for wheels and axles, the vehicle and track designers should cooperatively determine the impacts of such designs on wheels and rails.

2.7 WHEEL CENTER LIMITING FLANGE CONDITIONS

The standard for most LRV wheel designs includes resilient wheels such as the Bochum 54, Bochum 84, SAB, and the Acousta-Flex wheel designs.

Observation of internal wheel wear at the interface between the resilient wheel tire and the center hub has indicated substantial lateral deflection in the elastomer components as shown in **Figure 2.7.1**. Some resilient wheel designs include a limiting flange that controls the amount of lateral deflection when the outside wheel actually bears against the outside rail gauge face. On certain resilient wheel designs the limiting flange is unidirectional, controlling the lateral shift for a typical outside wheel-to-rail force. The limiting flange design does not consider the inner wheel action, as normally there is no lateral wheel restriction.

Most light rail track designs include guarded track on relatively sharp curves by providing a restraining rail adjacent to the inner rail. The guarding or restraining rail is positioned to contact the inside face of the inside wheel of the vehicle in a curve. This action, in fact, assists in steering the vehicle truck through the track curve. For additional information on guarded track, refer to Section 4.2.8. The restraining rail action results in a force on the wheel in the direction opposite to the customary wheel-rail gauge face flanging.

Figure 2.7.1 illustrates and documents the normal resilient wheel position, the lateral shift in the tire, and the distortion in the elastomer at the high rail. The limiting flange provides control of the lateral tire position. The figure also illustrates the inner wheel, wherein the restraining rail-to-wheel tire action actually opens the gap at the limiting flange Under these conditions, the wheel tire is free to shift to the limit of the elastomer distortion which is equal to the lateral outside wheel shift beyond the restraining rail flangeway width.

Wheel designers must consider transit systems design criteria for guarded track wherein the guard or restraining rail will place lateral restrictions on movement of the wheel out of the normal direction.

Notably, the resilient wheel designs for the North American PCC cars were designed with rigidity limits in both lateral directions. Whether this was by design or accident is unknown.

In addition, to accommodate the proposed heavy wheel flanging due to sharp curvature and excessive vehicle mass, the tire and wheel center component material and hardness should be re-evaluated to provide wear-resistant faces.

Wheel squeal in curves has continually been studied at the wheel/rail interface. Consideration must be given to wheel squeal caused by the limiting flange action.

2.8 VEHICLES AND STATIONS—ADA REQUIREMENTS

ADA requires that public operators of light rail transit systems make their transportation services, facilities and communication systems accessible to persons with disabilities. New vehicles and construction of facilities must provide the needed accessibility.

2.8.1 Clearance and Tolerances

To properly address ADA requirements, designers will consider all dimensional tolerances of the platform/vehicle interface, such as:

- Track-to-platform clearances
- Vehicle-to-track clearances
- Vehicle dimensional tolerances, new/old
- Vehicle load leveling

The tight horizontal and vertical clearance requirements between the vehicle door threshold and the platform edge impact the construction of track. In order to maintain these tolerances, it may be necessary to structurally connect the track and the platform This may best be accomplished using direct fixation track or embedded track with a structural slab connected to the platform structure.

Track design, station design, and vehicle design must comply with the requirements of the ADA (1990). As a guideline, new light rail transit stations should be designed taking into consideration the ultimate ADA goal of providing access for persons with disabilities. Horizontally, these requirements include providing platform edges that are within 75 millimeters (3 inches) of the edge of the vehicle floor with the door in the open position. Vertically, the vehicle floor elevation should be level with or slightly higher than the station platform elevation.

Figure 2.8.1 outlines the general configuration of the track-to-station platform interface with the desired installation tolerances. The illustration references both embedded track and direct fixation track designs that require construction of a permanent track bed in lieu of a ballasted section, which is subject to settlement and possible surface lift requirements.

2.9 REFERENCES

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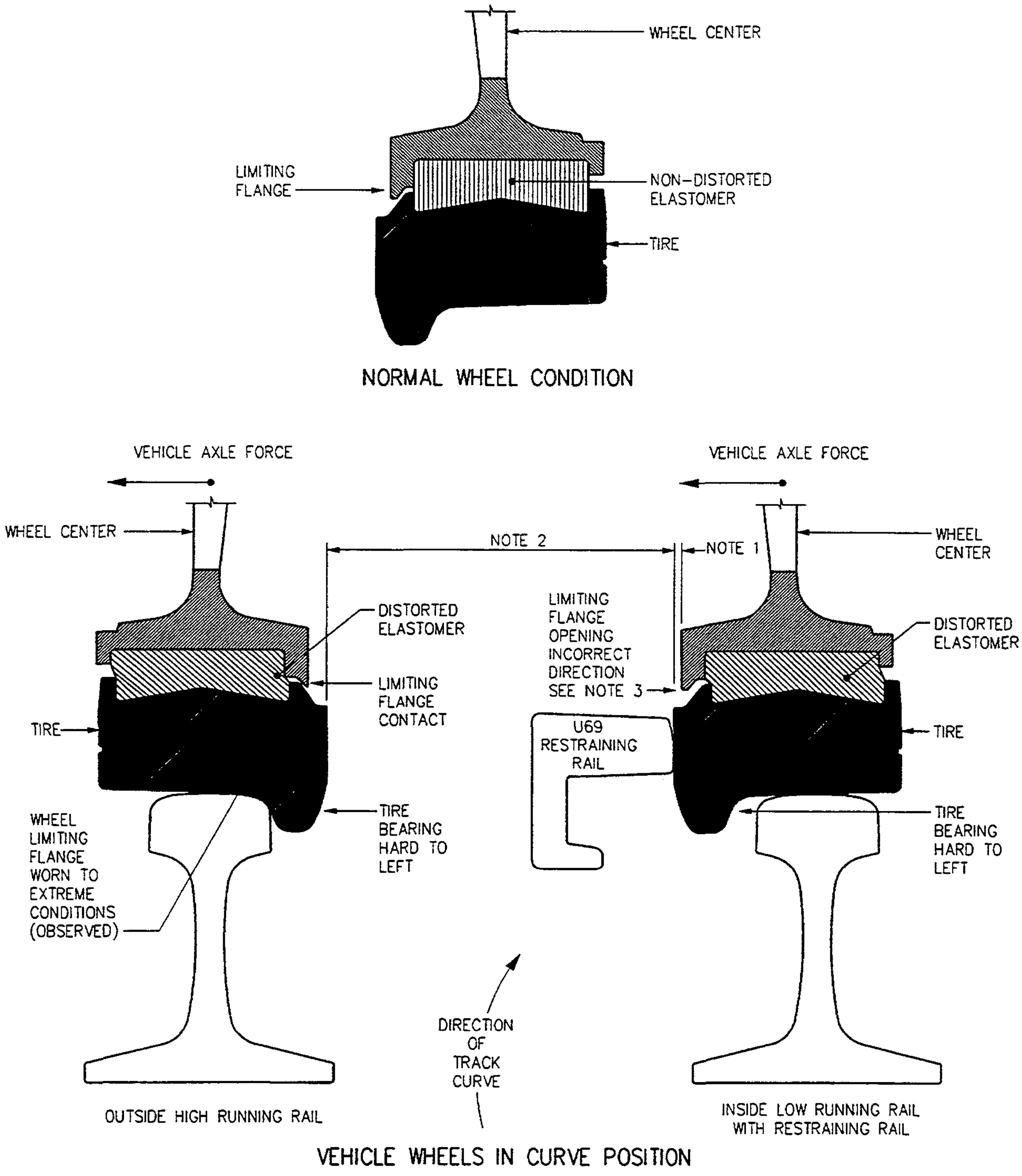
2] Leary, John F. "America Adopts Worn Wheel Profiles." *AAR Railway Gazette International*, July 1990.

[3] Kalousek, Joe & Magel, Eric, *Managing Rail Resources*, AREA Volume 98, Bulletin 760, May 1997.

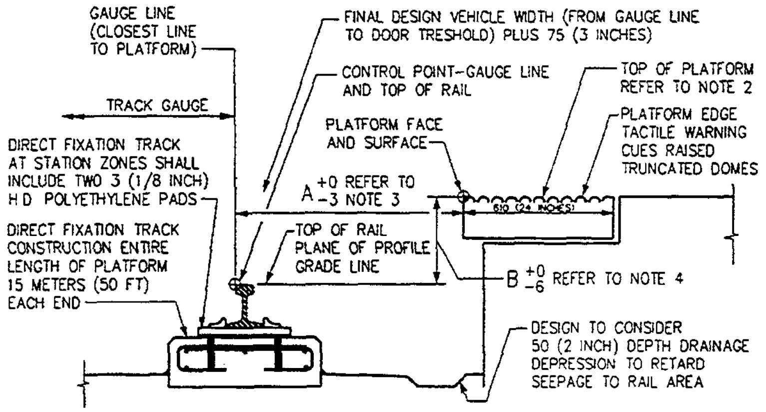
[4] Professor Hermann Heumann, Centenary Anniversary.

[5] Mace, Stephen E., *Improving the Wheel/Rail Interface*, Association of American Railroads *Railway Age*, October 1995.

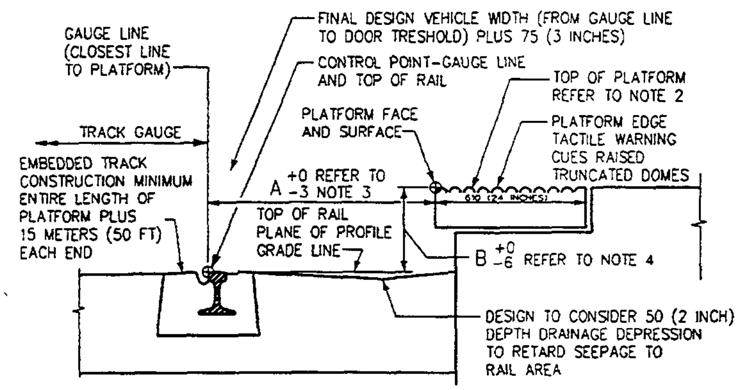


NOTES:
1. LATERAL SHIFTS AND DISTORTION OF RUBBER OF 6 millimeters (1/4 inch) IS CONSIDERED NORMAL.
2. WIDE WHEEL GAUGE DUE TO DISTORTION RESULTS IN GUARDED (RESTRAINING RAIL) SITUATIONS.
3. WHEEL CENTER TO TIRE SHIFT UNCONTROLLED AT INSIDE RUNNING RAIL.

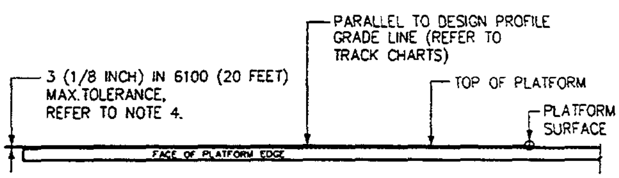
Figure 2.7.1 Resilient Wheel



DIRECT FIXATION TRACK SECTION AT STATION PLATFORM
RAIL TO PLATFORM GUIDELINES



EMBEDDED TRACK SECTION AT STATION PLATFORM
RAIL TO PLATFORM GUIDELINES



LONGITUDINAL PLATFORM TOLERANCES

NOTES:

1. DIMENSIONS A & B TO BE DETERMINED USING TRANSIT AGENCY VEHICLE WIDTH AND HEIGHT DIMENSIONS RESPECTIVELY
2. FOR THE ENTIRE LONGITUDINAL LENGTH OF PLATFORM, THE SURFACE DEVIATION SHALL HAVE A TOLERANCE OF 3 (1/8 INCH) IN 6100 (20 FEET) AND MUST BE CONSTRUCTED WITHIN THE STATED CRITICAL VERTICAL (TOP OF RAIL PLATFORM SURFACE) TOLERATED LIMITS
3. THE GAUGE LINE ALIGNMENT OF THE CLOSEST RAIL TO THE PLATFORM SHALL MAINTAIN A SPECIFIC HORIZONTAL RELATIONSHIP TO THE PLATFORM FACE WITHIN THE STATED HORIZONTAL CLEARANCE THROUGHOUT THE LENGTH OF THE STATION PLATFORM
4. THE TOP OF RAIL ALIGNMENT SHALL MAINTAIN A SPECIFIC VERTICAL RELATIONSHIP TO THE PLATFORM SURFACE WITHIN THE STATED VERTICAL CLEARANCE THROUGHOUT THE LENGTH OF THE STATION PLATFORM
5. PLATFORM EDGE OVERHANG SHALL BE OF SUFFICIENT LENGTH TO ALLOW INSTALLATION CONTRACTOR TO MEET THE REQUIREMENTS OF NOTE 1 AND A PLATFORM VEHICLE CLEARANCE OF 75 (3 INCHES) WITH VEHICLE IN NORMAL STATIC POSITION

Figure 2.8.1 Design Guidelines: Track at Station Platform

Chapter 3—Light Rail Transit Track Geometry

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CHAPTER 3—LIGHT RAIL TRANSIT TRACK GEOMETRY

3.1 INTRODUCTION

The most efficient track for operating any railway is straight and flat. Unfortunately, most railway routes are neither straight nor flat. Tangent sections of track need to be connected in a way that steers the train safely, ensuring that the passengers are comfortable and the cars and track perform well together. This dual goal is the subject of this chapter.

The primary goals of geometric criteria for light rail transit are to provide cost-effective, efficient, and comfortable transportation, while maintaining adequate factors of safety with respect to overall operations, maintenance, and vehicle stability. In general, design criteria guidelines are developed using accepted engineering practices and the experience of comparable operating rail transit systems.

Light rail transit (LRT) geometry standards and criteria differ from freight or commuter railway standards, such as those described in applicable sections of the American Railway Engineering and Maintenance-of-Way Association (AREMA) Manual, Chapter 5, in several important aspects. Although the major principles of LRT geometry design are similar or identical to that of freight/commuter railways, the LRT must be able to safely travel through restrictive alignments typical of urban central business districts, including rights-of-way shared with automotive traffic. Light rail vehicles are also typically designed to travel at relatively high operating speeds in suburban and rural settings.

The LRT alignment corridor is often predetermined by various physical or economic considerations inherent to design for urban areas One of the most common right-of-way corridors for new LRT construction is an existing or abandoned freight railway line.[1] The LRT vehicle is often required to operate at speeds of 65 to 90 kph (40 to 55 mph) through alignments that were originally designed for FRA Class 1 or 2 freight operations; i.e , less than 45 kph (30 mph)

General guidelines for the development of horizontal alignment criteria should be determined before formulating any specific criteria. This includes knowledge of the vehicle configuration and a general idea of the maximum operating speeds. An example of the latter is shown from an excerpt from the design criteria for one LRT system: [2]

> "Except for areas where the LRT operates within or adjacent to surface streets, the track alignment shall be designed to accommodate the maximum design speed of 90 kph (55 mph). Physical constraints along various portions of the system, together with other design limitations, may preclude achievement of this objective. Where the LRT operates within or adjacent to surface streets, the maximum design speed for the track alignment shall be limited to the legal speed of the parallel street traffic, but shall not exceed 57 kph (35 mph). In all areas, the civil design speed shall be coordinated with the normal operating speeds as provided on the train performance simulation program speed-distance profiles.
>
> Where the LRT system includes at-grade portions where light rail vehicles will operate in mixed traffic with rubber-tired vehicles in surface streets, the applicable geometric design criteria for such streets shall

be met in the design of the track alignment

Where the LRT system includes areas where light rail vehicles will operate in joint usage with railroad freight traffic, the applicable minimum geometric design criteria for each type of rail system shall be considered and the more restrictive shall govern the design of the track alignment and clearances."

Criteria for the design of LRT and freight railroad joint usage tracks are described later in this section.

In addition to the recommendations presented in the following sections, it should be noted that combinations of minimum horizontal radius, maximum grade, and maximum unbalanced superelevation are to be avoided in the geometric design.

The following geometric guidelines are established to consider both the limitations of horizontal, vertical, and transitional track geometry for cost-effective designs and the ride comfort requirements for the LRT passenger.

3.2 TRANSIT TRACK HORIZONTAL ALIGNMENT

The horizontal alignment of track consists of a series of tangents joined to circular curves and spiral transition curves. In yards and other non-revenue tracks, the requirement for spiral transition curve is frequently deleted. Track superelevation in curves is used to maximize vehicle operating speeds wherever practicable.

An LRT alignment is often constrained by both physical restrictions and minimum operating performance requirements. This generally results in the following effects on the LRT horizontal alignment and track superelevation designs:

- Minimum main line horizontal curve radius on new LRT systems is approximately 25 meters (82 feet), depending on physical restrictions and vehicle design.
- Superelevation unbalance ranges from 100 to 225 millimeters (4 to 9 inches), depending on vehicle design and passenger comfort tolerance.[3] Vehicle designs that can handle higher superelevation unbalance can operate at higher speeds through a given curve radius and actual superelevation combination. LRT superelevation unbalance is normally limited to 75 millimeters (3.0 inches); however, there are instances where 115 millimeters (4.5 inches) have been implemented.
- LRT spiral transition lengths and superelevation runoff rates are generally shorter than corresponding freight/commuter railway criteria.

In determining horizontal alignment, four levels of criteria may be considered.[4] These levels are based on a review of existing design criteria documents, particularly those with a combination of ballasted and embedded main line trackwork:

- *Main Line Desired Minimum*—This criterion is based on an evaluation of maximum passenger comfort, initial construction cost, and maintenance considerations on main line ballasted and direct fixation track. It is used where no physical restrictions or significant construction cost differences are encountered. An optional preferred minimum may also be indicated to define the most conservative possible future case; i.e., maximum future operating speed for given conditions within the alignment corridor

- *Main Line Absolute Minimum*—Where physical restrictions prevent the use of the main line desired minimum criterion, a main line absolute minimum criterion is often specified. This criterion is determined primarily by the vehicle design, with passenger comfort a secondary consideration.
- *Main Line Embedded Track*—Where the LRT is operated on low-speed embedded track, with or without shared automotive traffic, the physical restrictions encountered require a special set of geometric criteria that accommodates existing roadway profiles, street intersections, and narrow horizontal alignment corridors that are typical of urban construction.
- *Yard and Non-Revenue Track*—This criterion is generally less than main line track, covering low-speed and low-volume non-revenue service. The minimum criterion is determined primarily by the vehicle design, with little or no consideration of passenger comfort.

The yard and non-revenue track criteria may not be valid for relatively high-volume tracks such as yard main entrance leads. This criterion also must assume that work train equipment will use the tracks.

It should be emphasized that the use of absolute minimum geometric criteria, particularly for horizontal alignment, has several potential impacts in terms of increased annual maintenance, noise, and vehicle wheel wear, and shorter track component life. Its use should be implemented with extreme caution. One or two isolated locations of high track maintenance may be tolerated and included in a programmed maintenance schedule, but extensive use of absolute minimum design criteria can result in eventual revenue service degradation and unacceptable maintenance costs.

The recommended horizontal alignment criteria herein are based on the LRT vehicle design and performance characteristics described in Chapter 2.

3.2.1 Minimum Tangent Length Between Curves

The discussion of minimum tangent track length is related to circular curves (Section 3.2.4). The complete criteria for minimum tangent length will be developed here and referenced from other applicable sections.

The development of this criterion usually considers the requirements of AREMA Manual, Chapter 5, which specifies that the minimum length of tangent between curves is equal to the longest car that will traverse the system.[5] This usually translates into a desired minimum criterion of 30 meters (100 feet). Ride comfort criteria for transit systems must be considered, however, and the minimum length of tangent between curves is also given as:

$$L_T = 0.57V \qquad (L_T = 3V)$$

where:

L_T = minimum tangent length in meters (feet)

V = operating speed in kph (mph)

This formula is based on vehicle travel of at least 2 seconds on tangent track between two curves. This same criterion also applies to circular curves, as indicated below. This criteria has been used for various transit designs in the U.S. since BART in the early 1960s.[6] The desired minimum length between curves is thus usually expressed as an approximate car length or in accordance with the formula above, whichever is larger.

Main line absolute minimum tangent length depends on the vehicle and degree of passenger ride quality degradation that can be tolerated. One criterion is the maximum truck center distance plus axle spacing; i.e., the distance from the vehicle front axle to the rear axle. In other criteria, the truck center distance alone is sometimes used. When spiral curves are used, the difference between these two criteria is not significant.

An additional consideration for ballasted trackwork is the minimum tangent length for mechanized lining equipment, which is commonly based on multiples of 10-meter (31-foot) chords. Very short curve lengths have been noted to cause significant alignment throw errors by automatic track lining machines during surfacing operations. The 10-meter (31-foot) length can thus be considered an absolute floor on the minimum tangent distance for ballasted main line track in lieu of other criteria.

The preceding discussion is based on reverse curves. For curves in the same direction, it is preferable to have a compound curve, with or without a spiral transition curve, than to have a short length of tangent between the curves This condition, known as a "broken back" curve, does not affect safety or operating speeds, but does create substandard ride quality. As a guideline, curves in the same direction should preferably have no tangent between curves or, if required, the same minimum tangent distance as that applicable to reverse curves.

In embedded trackwork on city streets and in other congested areas, it may not be feasible to provide minimum tangent distances between reverse curves. Unless the maximum vehicle coupler angle is exceeded, one practical solution to this problem is to waive the tangent track requirements between curves if operating speeds are below 32 kph (20 mph) and no track superelevation is used on either curve. [4]

For yards and in special trackwork, it is usually not practicable to achieve the desired minimum tangent lengths AREMA Manual, Chapter 5, provides a series of minimum tangent distances based on long freight car configurations and worst-case coupler angles. The use of the AREMA table would be conservative for an LRT vehicle, which has much shorter truck centers and axle spacings than a typical freight railroad car. As speeds in yards are restricted and superelevation is generally not used, very minimal tangent lengths are required between curves. It is also noted in the AREMA Manual that turnouts and sidings can also create unavoidable short tangents between reverse curves.

Existing LRT criteria do not normally address minimum tangent lengths at yard tracks, but leave this issue to the discretion of the trackwork designer and/or the individual transit agency. To permit the use of work trains and similar rail mounted equipment, it is prudent to utilize the AREMA minimum tangent distances between reverse curves in yard tracks.

Having reviewed the various criteria for tangents between reverse curves, it is now possible to summarize typical guideline criteria for light rail transit:

Main Line Preferred

Minimum (Optional) The greater of either,

L_T = 60 meters (200 feet) or

$L_T = 0.57V$

where: L_T = minimum tangent length (meters)

V = maximum operating speed (kph)

Main Line Desired Minimum	The greater of either L_T = length of LRT vehicle over couplers (meters) or L_T = 0 57V where· L_T = minimum tangent length (meters) V = maximum operating speed (kph) Note: The LRT vehicle length over couplers is often rounded up to 30 meters (100 feet).
Main Line Absolute Minimum:	The greater of either L_T = 9.5 meters (31 feet) or L_T = (Vehicle Truck Center Distance) + (Axle Spacing)
Main Line Embedded Track	L_T = 0 meters, where vehicle coupler angle limits are not exceeded, speed is less than 32 kph (20 mph), and no track superelevation is used or L_T = main line absolute minimum
Yard and Non-Revenue Track:	The lesser of either, L_T = 9.5 meters (31 feet) or L_T = 0 meters (0 feet) for R>290 meters (955 feet) L_T = 3.0 meters (10 feet) for R>250 meters (818 feet) L_T = 6.1 meters (20 feet) for R>220 meters (716 feet) L_T = 7.6 meters (25 feet) for R>195 meters (637 feet) L_T = 9.1 meters (30 feet) for R>175 meters (573 feet) Note: Where absolutely necessary, the Main Line Embedded Track criteria may also be applied.

3.2.2 Vehicle Length Criteria

Refer to Sections 1.3 and 2.2 of this handbook for a discussion and data regarding vehicle length. Criteria for vehicle length are set not only by the vehicle capacity requirements, but also by clearance and track curvature considerations

The type of vehicle, whether articulated or low-floor, will also affect its overall length, truck center spacing, axle spacing, and center of gravity, all of which have an impact on the track alignment.

3.2.3 Speed Criteria, Vehicle and Passenger

The speed criteria for curved track is determined by carefully estimating passenger comfort and preventing undue forces on the trackwork, vehicle trucks/wheels, and vehicle frames. Vehicle stability on curved track is also an important consideration in the determination of LRT speed criteria.

In general, the limiting factors of the major alignment design components can be classified as shown in **Table 3.2.1**.

As indicated in previous sections, LRT operating speeds are generally in the range of 65 to 90 kph (40 to 55 mph), except on embedded trackwork. Separate geometric criteria are recommended for these conditions. Restricted operating speeds are always possible along the alignment corridor, but proposed design speeds below 60 kph (40 mph) generally create unacceptable constraints to the train control design and proposed operations.

Table 3.2.1 Alignment Design Limiting Factors

Alignment Component	Major Limiting Factors(s)
Minimum Length between Curves	• Passenger comfort • Vehicle truck/wheel forces
Circular Curves (Minimum Radius)	• Trackwork maintenance • Vehicle truck/wheel forces
Compound and Reverse Circular Curves	• Passenger comfort • Vehicle frame forces
Spiral Transition Curve Length	• Passenger comfort • Trackwork maintenance
Superelevation	• Passenger comfort • Vehicle stability
Superelevation Runoff Rate	• Passenger comfort • Vehicle frame forces
Vertical Tangent between Vertical Curves	• Passenger comfort
Vertical Curve/Grade (Maximum Rate of Change)	• Passenger comfort • Vehicle frame forces
Special Trackwork	• Passenger comfort • Trackwork maintenance
Station Platforms	• Vehicle clearances • ADA platform gap requirements
Joint LRT/Freight RR Usage	• Trackwork maintenance • Compatibility of LRT and freight vehicle truck/wheels

3.2.4 Circular Curves

Intersections of horizontal alignment tangents are connected by circular curves. The curves may be simple curves or spiraled curves, depending on the curve location, curve radius, and required superelevation.

LRT alignment geometry differs from freight railroad (AREMA) design in that the arc is used to define circular curves and the associated spirals. Also, curves for LRT designs are generally defined and specified by their radius rather than degree of curvature. This becomes an important distinction when designing in metric units, as the degree of curvature is defined entirely in English units and has no direct equivalent in metric units.

For conversion of existing alignment curve data calculated in English units, particularly those based on the degree of curvature, it is most efficient to determine the radius in English units, then convert to metric.

As a guideline for LRT design, curves should be specified by their radius. Degree of curvature, where required for calculation purposes, should be defined by the arc definition of curvature as determined by the following formula:

$$D = \frac{176\,379}{R} \qquad \left(D = \frac{5729\,58}{R}\right)$$

where, D = degree of curvature, in decimal degrees

R = radius of curvature, in meters (feet)

Circular curves for LRT design are, as noted above, defined by curve radius and arc of curve length. The geometric properties of the circular curve are summarized in **Figure 3.2.1**.

The minimum curve radius is determined by the physical characteristics of the vehicle. Although steerable trucks or "stiff" truck designs have an impact on minimum allowable track curve radius, the minimum radius is more severely affected by the distance between vehicle truck centers and truck axle spacing.

For most modern LRV designs, whether high- or low-floor, the most common absolute minimum radius appears to be 25 meters (82 feet). This is considerably larger than the 11- to 12-meter (36- to 40-foot) track radius that can be negotiated by a tram or PCC type vehicle The 25-meter track radius is still sufficient, however, to permit at-grade alignments in urban areas while maintaining an adequate vehicle capacity.

It is easier to maintain track on tangent alignments than on curves, and there is a curve radius threshold below which it becomes extremely expensive to maintain track components. In addition, the probability of wheel squeal increases dramatically on smaller radius curves. The use of restraining rail or girder guard rail as discussed in Chapter 4 of this handbook can reduce the severity of some of these track problems to tolerable levels, but at a relatively high initial cost.

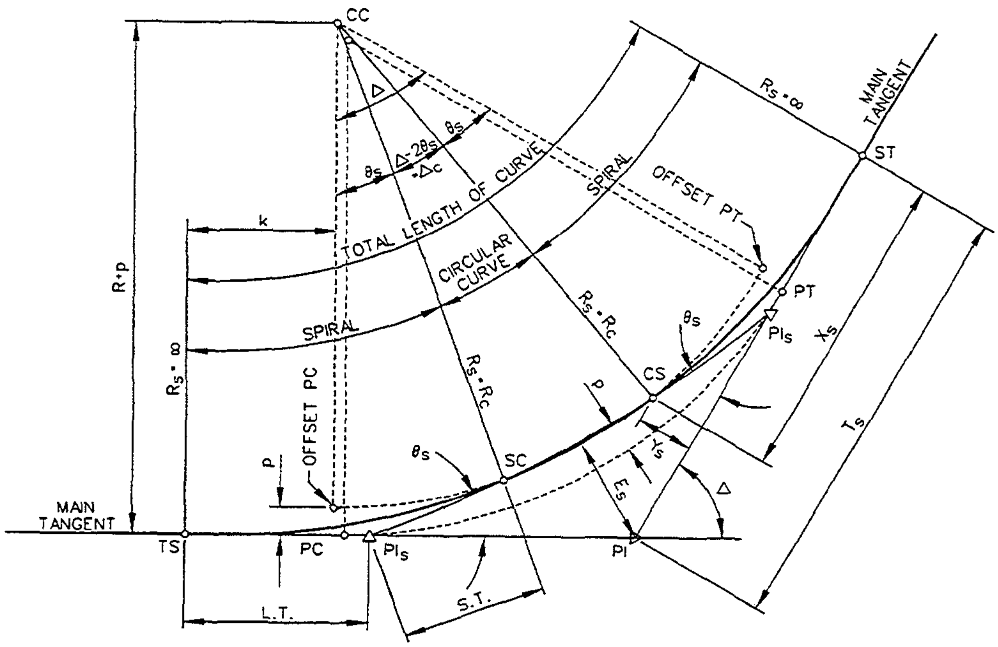
In some locations, such as aerial structures and tunnels, maintenance vehicle and equipment access must also be considered in the selection of minimum horizontal curve criteria.

The desired minimum curve radius is set at the threshold limit for restraining rail, as determined from Chapter 4 herein. In most cases, this is around 150 meters (500 feet). A secondary limit is considered for main line track, where rail guarding can control excessive maintenance and wheel squeal. Embedded track and yard track have far less rigid criteria, as vehicle speeds on these tracks are generally limited to 25 kph (16 mph).

Embedded main line track is normally permitted to be constructed at absolute minimum radii as a concession to the extreme alignment restrictions in urban areas. However, rail-mounted maintenance equipment, particularly work locomotives, must be able to operate on these tracks The use of absolute minimum radius curves should be thus restricted to main line terminal loops and yard turnaround or bypass tracks. [12]

In view of the design considerations indicated above, guideline criteria for modern LRV equipment are as follows for minimum curve radii:

Main Line Desired Minimum, except Embedded Track:	150 meters (500 feet)
Main Line Absolute Minimum, Aerial Structures and Tunnels:	150 meters (500 feet)
Main Line Absolute Minimum, Ballasted At-Grade:	90 meters (300 feet)
Main Line Embedded Track, Desired Minimum:	35 meters (115 feet)



NOTATIONS

- CC - CENTER OF CIRCULAR CURVE
- CS - POINT OF CHANGE FROM CIRCULAR CURVE TO SPIRAL
- D_c - DEGREE OF CIRCULAR CURVE, ARC DEFINITION
- E_s - TOTAL EXTERNAL DISTANCE OF A SPIRALIZED CURVE
- k - TANGENT DISTANCE FROM TS OR ST TO PC OR PT OF THE SHIFTED CIRCULAR CURVE
- L_c - TOTAL LENGTH OF CIRCULAR CURVE ARC
- L_s - TOTAL LENGTH OF SPIRAL
- L T - LONG TANGENT OF SPIRAL
- p - OFFSET FROM THE MAIN TANGENT TO THE PC OR PT OF THE SHIFTED CIRCULAR CURVE
- PC - POINT OF CHANGE FROM TANGENT TO CIRCULAR CURVE
- PI - POINT OF INTERSECTION OF MAIN TANGENTS
- PI_s - POINT OF INTERSECTION OF MAIN TANGENT WITH TANGENT THROUGH SC OR CS POINT
- PT - POINT OF CHANGE FROM CIRCULAR CURVE TO TANGENT
- R - RADIUS OF CIRCULAR CURVE
- SC - POINT OF CHANGE FROM SPIRAL TO CIRCULAR CURVE
- ST - POINT OF CHANGE FROM SPIRAL TO TANGENT
- S.T - SHORT TANGENT OF SPIRAL
- T_s - TOTAL TANGENT DISTANCE FROM TS OR ST TO PI
- TS - POINT OF CHANGE FROM TANGENT TO SPIRAL
- X_s - TANGENT DISTANCE FROM TS TO SC OR ST TO CS
- Y_s - TANGENT OFFSET AT SC OR CS
- Δ - TOTAL CENTRAL ANGLE OF SPIRAL AND CIRCULAR CURVES
- Δ_c - CENTRAL ANGLE OF THE CIRCULAR CURVE
- θ_s - CENTRAL ANGLE OF SPIRAL

CURVE FORMULAS

$$D_c = \frac{5729.578}{R}$$

$$T_s = (R+p)\tan\frac{\Delta}{2} + k$$

$$E_s = (R+p)\left(\frac{1}{\cos\frac{\Delta}{2}} - 1\right) + p$$

$$L_c = \frac{\Delta_c}{D_c} \times 100 = \frac{\Delta - 2\theta_s}{D_c} \times 100$$

SPIRAL FORMULAS

θ_s IN RADIANS

$$X_s = L_s\left(1 - \frac{\theta_s^2}{10} + \frac{\theta_s^4}{216} - \frac{\theta_s^6}{9360} \cdots\right)$$

$$Y_s = L_s\left(\frac{\theta_s}{3} - \frac{\theta_s^3}{42} + \frac{\theta_s^5}{1320} - \frac{\theta_s^7}{75600} \cdots\right)$$

$$k = L_s\left(\frac{1}{2} - \frac{\theta_s^2}{60} + \frac{\theta_s^4}{2160} - \frac{\theta_s^6}{131040} \cdot\cdot\right)$$

$$p = L_s\left(\frac{\theta_s}{12} - \frac{\theta_s^3}{336} + \frac{\theta_s^5}{15840} - \right)$$

$$L_s = 2R\theta_s$$

$$\theta_s = \frac{1}{2}\frac{L_s}{R}$$

$$L.T. = X_s - \frac{Y_s}{\tan\theta_s}$$

$$S.T. = \frac{Y_s}{\sin\theta_s}$$

Figure 3.2.1 Horizontal Curve and Spiral Nomenclature

Main Line Embedded Track, Absolute Minimum	25 meters (82 feet)
Yard and Non-Revenue Track, Desired Minimum:	30 meters (100 feet)
Yard and Non-Revenue Tracks, Absolute Minimum	25 meters (82 feet)

The minimum circular curve length is dictated by ride comfort and is hence, unlike minimum tangent length, not related to vehicle physical characteristics. The desired minimum circular curve length is generally determined by the following formula:

$$L = 0.57V \qquad (L = 3V)$$

where: L = minimum length of curve, in meters (feet)

V = design speed through the curve, in kph (mph)

For spiraled circular curves, the length of the circular curve added to the sum of one-half the length of both spirals is an acceptable method of determining compliance with the above criteria in areas of restricted geometry. The absolute minimum length of a superelevated circular curve should be 15 meters (45 feet).

Curves that include no actual circular curve segment (e.g., double-spiraled curves) should be permitted only in areas of extremely restricted geometry (such as embedded track in an urban area), provided no actual superelevation (E_a) is used and prior authority approval is obtained. This type of alignment is potentially difficult to maintain for ballasted track.

The design speed for a given horizontal curve should be based on its radius, length of spiral transition and actual and unbalance superelevation through the curve as described in the following sections.

3.2.4.1 Compound Circular Curves

The criterion for compound circular curves is similar to that of the tangent-to-curve transition described in Section 3 2.5 Although generally less severe, they must still address the dual objectives of passenger comfort and vehicle structural design in torsion

A transition spiral should be used at each end of a superelevated circular curve and between compound circular curves. Where compound curves are used, they should be connected by a spiral transition curve. The desired minimum main line spiral length is the greater of the lengths as determined by the following:

$$L_S = 0.38(E_{a2} - E_{a1}) \qquad (L_S = 31(E_{a2} - E_{a1}))$$

$$L_S = 0.006\,(E_{u2} - E_{u1})V \qquad (L_S = 0.82\,(E_{u2} - E_{u1})V)$$

$$L_S = 0.008\,(E_{a2} - E_{a1})V \qquad (L_S = 1.10\,(E_{a2} - E_{a1})V)$$

where L_S = minimum length of spiral, in meters (feet)

E_a = actual superelevation of the first circular curve in millimeters (inches)

E_{a2} = actual superelevation of the second circular curve, in millimeters (inches)

E_{u1} = superelevation unbalance of the first circular curve, in millimeters (inches)

E_{u2} = unbalanced superelevation of the second circular curve, in millimeters (inches)

V = design speed through the circular curves, in kph (mph)

The absolute minimum spiral curve on main line tracks, as well as the minimum criteria for yard and non-revenue tracks, is as follows, corresponding to LRV torsion limits:

$$L_S = 2\,(E_{a2} - E_{a1}) \qquad (L_S = 31\,(E_{a2} - E_{a1}))$$

3.2.4.2 Reverse Circular Curves

Where an extremely restrictive horizontal geometry makes it impossible to provide sufficient tangent length between reversed superelevated curves, the curves may meet at a point of reverse spiral. This tends to violate ride quality and vehicle structure criteria. As a guideline, the point of reverse spiral should be set so that:

$$L_{S1}\ E_{a2} = L_{S2}\ E_{a1}$$

where E_{a1} = actual superelevation applied to the first curve in millimeters (inches)

E_{a2} = actual superelevation of the second circular curve, in millimeters (inches)

L_{S1} = the length of the spiral leaving the first curve in meters (feet)

L_{S2} = length of the spiral entering second curve in meters (feet)

A minimum separation of 1.0 meter (3.3 feet) between the spirals is acceptable in lieu of meeting at a point of reversal.

It is entirely possible to have reverse spirals and remain within acceptable ride comfort criteria. This is indeed the practice for European interurban railway alignments and is occasionally incorporated in North American practice.[6] However, the spiral lengths required for reverse spirals to maintain ride comfort are significantly longer than normally considered in LRT design.

The superelevation transition between reversed spirals is usually accomplished by sloping both rails of the track throughout the entire transition spiral as shown on **Figure 3.2.2**. Note that through the transition, both rails will be at an elevation above the theoretical profile grade line. This method of superelevation transition creates additional design considerations, including an

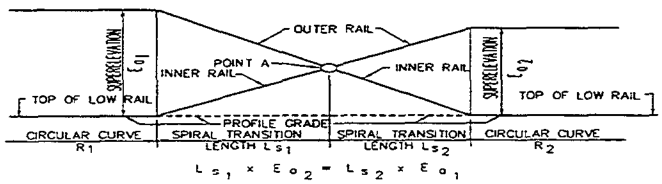


Figure 3.2.2 Supelevation Transitions for Reverse Curves

increased ballast section width at the point of the reverse spiral and possible increased clearance requirements. Such issues must be investigated in detail before incorporation in the design

In conclusion, the use of reversed spirals should be restricted to low speed operation. As a guideline, a reasonable criterion for the use of reversed spirals is given below: [2]

> "On embedded tracks in city streets, if alignment constraints make providing a tangent between two superelevated spiraled reversed curves impossible, a tangent shall not be required provided that the operating speed is limited so that the lateral acceleration is held to a maximum of 0.10 g."

Refer to Section 3.2.1 for additional discussion on minimum tangent distances between curves.

3.2.5 Superelevation and Spiral Transition Curves

The permissible speed at which a rail-mounted vehicle negotiates a curve may be increased by increasing the elevation of the outside rail of the track, creating a banking effect called superelevation. This superelevation serves to counteract the centrifugal force acting radially outward on the vehicle as it travels through the curve.[7]

For a given curve radius, the permissible operating speed can be increased by physically increasing the elevation of the outside rail of the curve, known as actual superelevation; or allowing the operating speed to exceed a lateral equilibrium force condition, known as superelevation unbalance. The latter is defined as the superelevation that would be required to restore an operating vehicle to an equilibrium steady state condition.

For vehicle operation in both actual superelevation and superelevation unbalance, there must be a transition to either zero superelevation or a different superelevation condition The logical method of accomplishing this transition on a circular curve with actual superelevation (and/or superelevation unbalance) is to utilize a spiral curve with a gradually increasing radius to tangent track, or a different horizontal curve radius

Actual superelevation is generally applied (run off) linearly throughout the length of the transition curve. As the rate of superelevation run off is necessarily limited by passenger comfort considerations, the transition curve length is determined by the length necessary to run off either the actual superelevation or superelevation unbalance.

3.2.5.1 Superelevation

Main line tracks are designed with superelevations that permit desired design speeds to be achieved without resorting to excessively large curve radii. Note that due to local constraints, the design speed may be less than either the system maximum speed or the maximum possible speed for a curve of a given radius. The design speed criteria stated below are based on a maximum lateral passenger acceleration of 0.10 g.

Equilibrium superelevation is the amount of superelevation that would be required to make the resultant force from the center of gravity of the light rail vehicle perpendicular to the plane of the two rails and halfway between them at a given speed. If a curved track is superelevated to achieve equilibrium at a given speed, a light rail vehicle passenger would experience no centrifugal force through the curve at that speed. Equilibrium superelevation is usually determined by either of the following equations:

$$E_q = E_a + E_u = 11.7\left(\frac{V^2}{R}\right) \quad \left[E_q = E_a + E_u = 3.96\left(\frac{V^2}{R}\right)\right]$$

$$E_q = 0.0067V^2 D \quad \left[E_q = 0.00069V^2 D\right]$$

where E_q = equilibrium superelevation, in millimeters (inches)

E_a = actual track superelevation to be constructed in millimeters (inches)

E_u = unbalance superelevation, in millimeters (inches)

V = design speed through the curve in kph (mph)

R = radius of curve in meters (feet)

D = degree of curve in decimal degrees

[Note previous comments on the use of degree of curvature with metric units.]

In practice, full equilibrium superelevation (E_q) is rarely installed in track. This would require excessively long spiral transition curves. It could also produce passenger discomfort on a train that is moving much slower than the design speed or stopped in the middle of a steeply superelevated curve. Therefore, only a portion of the calculated equilibrium superelevation (E_q) is commonly installed as actual superelevation (E_a). The difference between the equilibrium and actual superelevation is called superelevation unbalance (E_u). Most curves will be designed with some combination of actual and unbalanced superelevation.

Three strategies are generally employed to apply the combination of actual superelevation and superelevation unbalance:

1. No (or minimal) superelevation unbalance is applied until actual superelevation (E_a) reaches the maximum allowable level. Actual superelevation is thus equal to the equilibrium superelevation for most curves. Under ideal conditions, where all vehicles operate at the same maximum speed and do not stop (or slow down) on curves, this strategy creates the least amount of passenger and vehicle lateral acceleration for a given transition curve length. Under less than ideal operating conditions, however, the minimum superelevation unbalance strategy produces unfavorable ride comfort conditions.
2. Maximum superelevation unbalance is applied before any actual superelevation is considered. This option is used by freight and suburban commuter railroads. Where a wide variety of operating speeds are anticipated on the curved track, particularly on joint LRT-freight trackage, this strategy is usually the least disruptive to passenger comfort.
3. Actual superelevation (E_a) and superelevation unbalance (E_u) are applied equally or in some proportion. Because a certain amount of superelevation unbalance, applied gradually, is generally considered to be easily tolerated by both vehicle and passenger and tolerable superelevation unbalance increases with speed, this strategy is preferred for general usage.

One method used to apply the combination of actual and unbalanced superelevation is to find the total equilibrium superelevation (E_q) and divide the total equally between actual and unbalanced superelevation; i.e., (E_a = $E_q/2$) and (E_u = $E_q/2$). Where E_u reaches its maximum value (see below), the remaining portion of the total equilibrium superelevation (E_q) is applied to the actual superelevation (E_a).

As a practical matter for construction, curves with a large radius in comparison to the desired operating speed should not be superelevated. This can be accomplished by not applying actual superelevation (E_a) until the calculated total equilibrium superelevation (E_q) is over a minimum value, usually 12 to 25 millimeters (0.05 to 1.00 inches).

Desired values of actual superelevation (E_a) can be determined from the following formula:

$$E_a = 7.85\left(\frac{V^2}{R}\right) - 16.7 \qquad \left[E_a = 2.64\left(\frac{V^2}{R}\right) - 0.66\right]$$

The desired relationship between E_a and E_u can thus be defined as:

$$E_u = 25 - \frac{E_a}{2} \qquad \left[E_u = 1 - \frac{E_a}{2}\right]$$

Use of the above equation will result in the gradual introduction of both actual and unbalanced superelevation and avoid unnecessary lateral acceleration of light rail vehicles and their passengers. Calculated values for actual superelevation should be rounded to the nearest 5 millimeters (0.25 inch). For a total superelevation (E_a + E_u) of 25 millimeters (1 inch) or less, actual superelevation (E_a) is not usually applied. In specific cases where physical constraints limit the amount of actual superelevation (E_a) that can be introduced, a maximum of 40 millimeters (1.5 inches) of superelevation unbalance (E_u) can be permitted before applying any actual superelevation (E_a).

Actual superelevation (E_a) is usually set so that trains will have a positive superelevation unbalance (E_u) on curves where speed is likely to vary. Negative E_u is not tolerated well by passengers. **Table 3.2.2** provides desired values of actual superelevation recommended for LRT alignment calculations. Other combinations of E_a and E_u should be used only where physical restrictions make the use of desired values prohibitive or impractical

Actual superelevation (E_a) should be attained and removed linearly throughout the full length of the spiral transition curve by raising the outside rail while maintaining the inside rail at the profile grade. One exception to this method of superelevation is sometimes employed in tunnels with direct fixation tracks, where superelevation is achieved by rotating the track section about the centerline. This is undertaken to reduce vertical clearance requirements.

Maximum values of actual superelevation can be as high as 200 to 250 millimeters (8 to 10 inches). Superelevation unbalance values of 150 millimeters (6 inches) are not unreasonable for LRT vehicle designs.[7] While these values are achievable by specific light rail vehicle designs, it is much more common for actual superelevation to be limited to 150 millimeters (6 inches) and unbalanced superelevation to 115 millimeters (4.5 inches). This limit equates to the 0.1 g limit that passengers can tolerate comfortably.

As a guideline, the recommended maximum values for actual and superelevation unbalance are as follows:

Superelevation Maximum Values:
E_a = 100 mm (4 inches) desired, 150 mm (6 inches) absolute
E_u = 75 mm (3 inches) desired, 115 mm (4.5 inches) absolute

In areas of mixed traffic operation with roadway vehicles, the desired location for a pavement crown is at the centerline of track. Where this is not feasible, a maximum pavement crown of 2.0% (1/4 inch per foot) across the rails may be maintained in the street pavement to promote drainage. This practice will normally introduce a constant actual superelevation (E_a) of approximately 25 millimeters (1 inch). If, at curves, the street pavement is neither superelevated nor the crown removed, this crown-related superelevation may also dictate the maximum allowable operating speed.

On curved track, this 25 millimeters (1 inch) could be either positive or negative, depending on which side of the roadway crown line the track is located. In such cases, in order to minimize the need to extensively regrade street pavements, which could affect curb reveal heights and other civil features, the superelevation unbalance should be maximized prior to the introduction of any additional actual superelevation. Thus, a normal pavement crown would retain an actual superelevation (E_q) of 25 millimeters (1 inch) until a calculated superelevation unbalance (E_u) of 75 millimeters (3 inches) is reached. At this point, either a limit is placed on the LRT design speed or the pavement crown design is revised.

3.2.5.3 Spiral Transition Curves

Spiral transition curves are used to gradually build into the superelevation of the track and limit lateral acceleration during the horizontal transition of the light rail vehicle as it enters the curve.

Table 3.2.2a Desired Superelevation and Minimum Spiral Curve Length (Metric Units)

		CURVE RADIUS (meters)																												
VEL. (kph)		25	27	30	35	40	45	50	55	60	65	70	75	80	85	90	95	100	110	120	130	140	150	160	170	180	200	220	240	260
15	Ea	55	50	45	35	30	25	20	20	15	15	10	10	10	5	5	5	5	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	22	20	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18
20	Ea	110	100	90	75	65	55	50	45	40	35	30	30	25	25	20	20	15	15	10	10	10	5	5	5	5	0	0	0	0
	Ls	42	40	36	30	26	22	20	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18
25	Ea	Min. R =		150	125	110	95	85	75	70	60	55	50	45	45	40	35	35	30	25	25	20	20	15	15	15	10	10	5	5
	Ls			58	48	42	38	34	30	28	24	22	20	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18
30	Ea	Min. R = 43 m					145	125	115	105	95	85	80	75	70	65	60	55	50	45	40	35	35	30	25	25	20	20	15	15
	Ls						56	48	44	40	38	34	32	30	28	26	24	22	20	18	18	18	18	18	18	18	18	18	18	18
35	Ea	Min. R = 58 m								145	135	125	115	105	100	95	85	80	75	65	60	55	50	45	40	40	35	30	25	25
	Ls									56	52	48	44	40	40	38	34	32	30	26	24	22	20	18	18	18	18	18	18	18
40	Ea	Min. R = 76 m												145	135	125	120	110	100	90	80	75	70	65	60	55	50	45	40	35
	Ls													56	52	48	46	42	40	36	32	30	28	26	24	22	20	18	18	18
45	Ea	Min. R = 96 m																145	130	120	110	100	90	85	80	75	65	60	50	45
	Ls																	56	50	46	42	40	36	34	32	30	26	24	20	18
50	Ea	Min. R = 118 m																		150	135	125	115	110	100	95	85	75	70	60
	Ls																			62	56	52	48	46	42	40	36	32	30	26
55	Ea	Min. R = 143 m																					145	135	125	120	105	95	85	75
	Ls																						64	60	56	54	48	42	38	34
60	Ea	Min. R = 170 m																							150	145	125	115	105	95
	Ls																								74	70	62	56	52	46
65	Ea	Min. R = 199 m																									150	135	125	115
	Ls																										80	72	66	60
70	Ea	Min. R = 231 m																											145	135
	Ls																												82	76
75	Ea	Min. R = 265 m																												
	Ls																													
80	Ea	Min. R = 302 m																												
	Ls																													
85	Ea	Min. R = 341 m																												
	Ls																													
90	Ea	Min. R = 382 m																												
	Ls																													
95	Ea	Min. R = 425 m																												
	Ls																													
100	Ea	Min. R = 471 m																												
	Ls																													

VEL. (kph)		CURVE RADIUS (meters)																												
		280	300	320	340	360	380	400	420	440	460	480	500	520	540	560	580	600	620	640	660	680	700	720	740	760	780	800	850	900
15	**Ea**	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18
20	**Ea**	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18
25	**Ea**	5	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18
30	**Ea**	10	10	10	5	5	5	5	5	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18
35	**Ea**	20	20	15	15	15	10	10	10	10	5	5	5	5	5	5	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18
40	**Ea**	30	30	25	25	20	20	15	15	15	15	10	10	10	10	10	5	5	5	5	5	5	5	5	5	0	0	0	0	0
	Ls	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18
45	**Ea**	45	40	35	35	30	30	25	25	20	20	20	20	15	15	15	15	10	10	10	10	10	10	10	5	5	5	5	5	5
	Ls	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18
50	**Ea**	55	50	45	45	40	35	35	35	30	30	25	25	25	20	20	20	20	15	15	15	15	15	15	10	10	10	10	10	10
	Ls	24	22	20	20	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18
55	**Ea**	70	65	60	55	50	50	45	40	40	35	35	35	30	30	30	25	25	25	25	20	20	20	20	20	15	15	15	15	10
	Ls	32	30	28	26	24	24	20	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18
60	**Ea**	85	80	75	70	65	60	55	55	50	45	45	40	40	40	35	35	35	30	30	30	25	25	25	25	25	20	20	20	15
	Ls	42	40	38	34	32	30	28	28	26	22	22	20	20	20	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18
65	**Ea**	105	95	90	85	80	75	70	65	60	60	55	50	50	45	45	45	40	40	40	35	35	35	30	30	30	30	25	25	25
	Ls	56	50	48	46	42	40	38	34	32	32	30	28	28	24	24	24	22	22	22	20	20	20	18	18	18	18	18	18	18
70	**Ea**	125	115	105	100	95	85	80	75	75	70	65	65	60	55	55	50	50	50	45	45	40	40	40	40	35	35	35	30	30
	Ls	72	66	60	58	54	48	46	44	44	40	38	38	34	32	32	30	30	30	26	26	24	24	24	24	20	20	20	18	18
75	**Ea**	145	135	125	115	110	100	95	90	85	80	80	75	70	70	65	60	60	55	55	55	50	50	45	45	45	40	40	40	35
	Ls	88	82	76	70	68	62	58	56	52	50	50	46	44	44	40	38	38	34	34	34	32	32	28	28	28	26	26	26	22
80	**Ea**	Min. R =		145	135	125	120	110	105	100	95	90	85	80	80	75	70	70	65	65	60	60	60	55	55	50	50	50	45	40
	Ls			94	88	82	78	72	68	66	62	58	56	52	52	50	46	46	42	42	40	40	40	36	36	34	34	34	30	26
85	**Ea**	R= 341 m				145	135	130	120	115	110	105	100	95	90	85	85	80	75	75	70	70	65	65	60	60	60	55	55	50
	Ls					100	92	90	82	80	76	72	70	66	62	58	58	56	52	52	48	48	46	46	42	42	42	38	38	36
90	**Ea**	R = 382 m						145	135	130	125	120	115	110	105	100	95	90	90	85	80	80	75	75	70	70	65	65	60	55
	Ls							106	98	94	92	88	84	80	76	74	70	66	66	62	58	58	56	56	52	52	48	48	44	40
95	**Ea**	R = 425 m								145	140	135	125	120	115	110	110	105	100	95	95	90	85	85	80	80	75	75	70	65
	Ls									112	108	104	96	92	88	84	84	80	78	74	74	70	66	66	62	62	58	58	54	50
100	**Ea**	R = 471 m										150	145	135	130	125	120	115	110	110	105	100	100	95	90	90	85	85	80	75
	Ls											122	118	110	106	102	98	94	90	90	86	82	82	78	74	74	70	70	66	62

VEL. (kph)		950	1000	1050	1100	1150	1200	1250	1300	1350	1400	1450	1500	1600	1700	1800	1900	2000	2100	2200	2300	2400	2500	2600	2700	2800	2900	3000	>3000
		CURVE RADIUS (meters)																											
15	Ea	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	0
20	Ea	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	0
25	Ea	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	0
30	Ea	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	0
35	Ea	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	0
40	Ea	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	0
45	Ea	5	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	0
50	Ea	5	5	5	5	5	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	0
55	Ea	10	10	10	5	5	5	5	5	5	5	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	0
60	Ea	15	15	15	10	10	10	10	10	5	5	5	5	5	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	0
65	Ea	20	20	15	15	15	15	10	10	10	10	10	10	5	5	5	5	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	0
70	Ea	25	25	20	20	20	20	15	15	15	15	10	10	10	10	5	5	5	5	5	5	0	0	0	0	0	0	0	0
	Ls	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	0
75	Ea	30	30	30	25	25	25	20	20	20	15	15	15	15	10	10	10	10	5	5	5	5	5	5	0	0	0	0	0
	Ls	20	20	20	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	0
80	Ea	40	35	35	30	30	30	25	25	25	20	20	20	15	15	15	10	10	10	10	10	5	5	5	5	5	5	5	0
	Ls	26	24	24	20	20	20	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	0
85	Ea	45	45	40	35	35	35	30	30	30	25	25	25	20	20	15	15	15	15	10	10	10	10	10	5	5	5	5	0
	Ls	32	32	28	24	24	24	22	22	22	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	18	0
90	Ea	55	50	45	45	40	40	35	35	35	30	30	30	25	25	20	20	20	15	15	15	10	10	10	10	10	10	5	0
	Ls	40	38	34	34	30	30	26	26	26	22	22	22	20	20	18	18	18	18	18	18	18	18	18	18	18	18	18	0
95	Ea	60	55	55	50	45	45	40	40	40	35	35	35	30	25	25	25	20	20	20	15	15	15	15	10	10	10	10	0
	Ls	46	42	42	40	36	36	32	32	32	28	28	28	24	20	20	20	18	18	18	18	18	18	18	18	18	18	18	0
100	Ea	70	65	60	55	55	50	50	45	45	40	40	40	35	30	30	25	25	25	20	20	20	15	15	15	15	15	10	0
	Ls	58	54	50	46	46	42	42	38	38	34	34	34	30	26	26	22	22	22	18	18	18	18	18	18	18	18	18	0

Table 3.2.2b Desired Superelevation and Minimum Spiral Curve Length (English Units)

		CURVE RADIUS (feet)																												
VEL. (mph)		82	90	100	110	120	130	150	175	200	225	250	275	300	350	400	450	500	550	600	650	700	750	800	900	1000	1100	1200	1300	1400
10	Ea	2.50	2.25	2.00	1.75	1.50	1.25	1.00	0.75	1.50	0.50	0.50	0.25	0.25	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	80	70	65	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60
15	Ea	Min.	6.00	5.25	4.75	4.25	4.00	3.25	2.75	2.25	2.00	1.75	1.50	1.25	1.00	0.75	0.75	0.50	0.50	0.25	0.25	0.25	0.25	0	0	0	0	0	0	0
	Ls		190	165	150	135	125	105	90	70	65	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60
20	Ea	Min. R = 159 ft.							5.50	4.50	4.00	3.50	3.25	2.75	2.25	2.00	1.75	1.50	1.25	1.00	1.00	0.75	0.75	0.50	0.50	0.50	0.25	0.25	0.25	0
	Ls								175	140	125	110	105	90	70	65	60	60	60	60	60	60	60	60	60	60	60	60	60	60
25	Ea	Min. R = 248 ft.										6.00	5.25	5.00	4.00	3.50	3.00	2.75	2.25	2.00	2.00	1.75	1.50	1.50	1.25	1.00	0.75	0.75	0.50	0.50
	Ls											190	165	160	125	110	95	90	70	65	65	60	60	60	60	60	60	60	60	60
30	Ea	Min. R = 357 ft.														5.25	4.50	4.00	3.50	3.25	3.00	2.75	2.50	2.25	2.00	1.75	1.50	1.25	1.25	1.00
	Ls															175	150	135	120	110	100	95	85	75	70	60	60	60	60	60
35	Ea	Min. R = 486 ft.																5.75	5.25	4.75	4.25	4.00	3.75	3.25	3.00	2.50	2.25	2.00	1.75	1.75
	Ls																	225	205	185	165	155	145	130	120	100	90	80	70	70
40	Ea	Min. R = 635 ft.																			6.00	5.25	5.00	4.50	4.00	3.50	3.25	3.00	2.50	2.25
	Ls																				265	235	225	200	180	155	145	135	115	100
45	Ea	Min. R = 803 ft.																							5.25	4.75	4.25	3.75	3.50	3.25
	Ls																								260	240	215	190	175	165
50	Ea	Min. R = 991 ft.																								6.00	5.25	4.75	4.50	4.00
	Ls																									335	290	265	250	225
55	Ea	Min. R = 1199 ft.																										6.00	5.50	5.00
	Ls																											365	335	305
60	Ea	Min. R = 1427 ft.																												
	Ls																													
65	Ea	Min. R = 1675 ft.																												
	Ls																													

VEL. (mph)		CURVE RADIUS (feet) 1500	1600	1700	1800	1900	2000	2100	2200	2300	2400	2500	2600	2700	2800	2900	3000	3200	3400	3600	3800	4000	4200	4400	4600	4800
10	Ea	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60
15	Ea	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60
20	Ea	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60
25	Ea	0.50	0.25	0.25	0.25	0.25	0.25	0.25	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60
30	Ea	1.00	0.75	0.75	0.75	0.50	0.50	0.50	0.50	0.50	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0	0	0	0	0	0	0	0	0
	Ls	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60
35	Ea	1.50	1.25	1.25	1.25	1.00	1.00	1.00	0.75	0.75	0.75	0.75	0.50	0.50	0.50	0.50	0.50	0.25	0.25	0.25	0.25	0.25	0	0	0	0
	Ls	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60
40	Ea	2.25	2.00	1.75	1.75	1.50	1.50	1.25	1.25	1.25	1.00	1.00	1.00	1.00	0.75	0.75	0.75	0.75	0.50	0.50	0.50	0.50	0.25	0.25	0.25	0.25
	Ls	100	90	80	80	70	70	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60
45	Ea	3.00	2.75	2.50	2.25	2.25	2.00	2.00	1.75	1.75	1.50	1.50	1.50	1.25	1.25	1.25	1.00	1.00	1.00	0.75	0.75	0.75	0.50	0.50	0.50	0.50
	Ls	150	140	125	115	115	100	100	90	90	75	75	75	65	65	65	60	60	60	60	60	60	60	60	60	60
50	Ea	3.75	3.50	3.25	3.00	2.75	2.75	2.50	2.25	2.25	2.00	2.00	2.00	1.75	1.75	1.50	1.50	1.50	1.25	1.25	1.00	1.00	1.00	0.75	0.75	0.75
	Ls	210	195	180	170	155	155	140	125	125	115	115	115	100	100	85	85	85	70	70	60	60	60	60	60	60
55	Ea	4.75	4.25	4.00	3.75	3.50	3.25	3.25	3.00	2.75	2.75	2.50	2.50	2.25	2.25	2.00	2.00	1.75	1.75	1.50	1.50	1.50	1.25	1.25	1.00	1.00
	Ls	290	260	245	230	215	200	200	185	170	170	155	155	140	140	125	125	110	110	95	95	95	80	80	65	65
60	Ea	5.75	5.25	5.00	4.75	4.50	4.00	4.00	3.50	3.50	3.25	3.25	3.00	3.00	2.75	2.75	2.50	2.25	2.25	2.00	2.00	1.75	1.50	1.50	1.50	1.25
	Ls	380	350	335	315	300	265	265	235	235	215	215	200	200	185	185	170	150	150	135	135	120	100	100	100	85
65	Ea	R = 1675 ft.		6.00	5.50	5.25	5.00	4.50	4.50	4.25	4.00	3.75	3.75	3.50	3.25	3.25	3.00	3.00	2.75	2.50	2.25	2.25	2.00	2.00	1.75	1.75
	Ls			430	395	380	360	325	325	305	290	270	270	255	235	235	215	215	200	180	165	165	145	145	130	130

VEL. (mph)		5000	5200	5400	5600	6600	6800	7000	7500	8000	8500	9000	9500	10000	12000	>12000
10	Ea	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	60	60	60	60	60	60	60	60	60	60	60	60	60	60	0
15	Ea	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	60	60	60	60	60	60	60	60	60	60	60	60	60	60	0
20	Ea	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	60	60	60	60	60	60	60	60	60	60	60	60	60	60	0
25	Ea	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	60	60	60	60	60	60	60	60	60	60	60	60	60	60	0
30	Ea	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	60	60	60	60	60	60	60	60	60	60	60	60	60	60	0
35	Ea	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	60	60	60	60	60	60	60	60	60	60	60	60	60	60	0
40	Ea	0.25	0.25	0	0	0	0	0	0	0	0	0	0	0	0	0
	Ls	60	60	60	60	60	60	60	60	60	60	60	60	60	60	0
45	Ea	0.50	0.50	0.25	0.25	0.25	0.25	0	0	0	0	0	0	0	0	0
	Ls	60	60	60	60	60	60	60	60	60	60	60	60	60	60	0
50	Ea	0.75	0.50	0.50	0.50	0.25	0.25	0.25	0.25	0.25	0	0	0	0	0	0
	Ls	60	60	60	60	60	60	60	60	60	60	60	60	60	60	0
55	Ea	1.00	1.00	0.75	0.75	0.50	0.50	0.50	0.50	0.50	0.25	0.25	0.25	0.25	0	0
	Ls	65	65	60	60	60	60	60	60	60	60	60	60	60	60	0
60	Ea	1.25	1.25	1.00	1.00	0.75	0.75	0.75	0.50	0.50	0.50	0.50	0.50	0.25	0.25	0
	Ls	85	85	70	70	60	60	60	60	60	60	60	60	60	60	0
65	Ea	1.50	1.50	1.50	1.50	1.00	1.00	1.00	0.75	0.75	0.75	0.50	0.50	0.50	0.25	0
	Ls	110	110	110	110	75	75	75	60	60	60	60	60	60	60	0

Horizontal spiral curves are broadly defined as curves with a constantly decreasing or increasing radius proportional between either a tangent and curve (simple spiral) or between two curves (compound spiral).

There are many formulae that describe or approximate the alignment that conforms to the above definition. Various types of spirals found in railway alignment design include AREMA Ten Chord, PTC/SEPTA, Cubic, Bartlett, Hickerson, clothoid, and ATEA. For the spiral lengths and curvatures found in LRT, all of the above spiral formulae will generally describe the same physical alignment laterally to within several millimeters. The choice of spiral easement curve type is thus not critical.

It is important, however, to utilize only one of the spiral types, and define it as succinctly as possible. Vague terms such as "clothoid spiral" should be clarified as more than one formula describes this type of spiral curve. A spiral transition curve that is most commonly used in transit work is the Hickerson spiral. Its main advantage is that it is well-defined in terms of data required for both alignment design and field survey work.

Spiral curve length and superelevation runoff are directly related to passenger comfort At this point, it is useful to review the basis of both superelevation theory and runoff rate. There are a number of good explanations of the derivation of runoff theory; the references at the end of this section contain extensive background on the subject. [8,11]

While passenger comfort is a major consideration, the designer must also limit the rate of change in superelevation in a transition curve to avoid overstressing the vehicle frame through twisting. In order to accomplish this, the superelevation differential between truck centers should not exceed 25 mm (1 inch). As a guideline, for a car with 7-meter (23-foot) truck centers, the minimum transition length for a 75-mm (3-inch) superelevation is 21 meters (69 feet).

3.2.5.3.1 Spiral Transition Curve Lengths

For LRT design, it is recommended that spiral transition curves should be clothoid spirals as depicted in Figure 3.2.1 and as mathematically defined by Hickerson. [13] Spirals should be used on all main line track horizontal curves with radii less than 3,000 meters (10,000 feet) wherever practicable.

As a guideline, the recommended criteria for the LRT transition spiral length, based on the theoretical development in the previous section, are presented herein.

It is recommended that the length of spiral be at least 20 meters (60 feet). Where geometric conditions are extremely restricted, such as in unsuperelevated embedded track in a CBD area, the spiral length may be reduced to the absolute minimum of 10 meters (31 feet). The minimum length of spiral should be the greater of the lengths determined from the following formulae, rounded to the next even meter (or 5 feet).

$$L_s = 0.38\,E_a \qquad (L_s = 31\,E_a)$$
$$L_s = 0.006\,VE_u \qquad (L_s = 0.82\,E_uV)$$
$$L_s = 0.008\,VE_a \qquad (L_s = 1.10\,E_aV)$$

where: E_q = equilibrium superelevation in millimeters (inches)

L_s = length of spiral in meters (feet)

E_a = actual track superelevation to be constructed in millimeters (inches)

E_u = unbalance superelevation in millimeters (inches)

V = design speed through the curve, in kph (mph)

A spiral is preferred, but not required, for yard and secondary tracks where design speeds are less than 16 kph (10 mph). Curves on yard lead and secondary tracks that have design speeds greater than 16 kph (10 mph) should have spiral transition curves and superelevation

Under normal design conditions, superelevation should be introduced and run off uniformly throughout the length of a spiral transition curve In extraordinary cases, the superelevation may be developed along the tangent preceding the point of curvature (PC), or run off in the tangent immediately beyond the point of tangency (PT). The transition length is then determined from the minimum spiral length formulae presented herein. The maximum amount of superelevation that is run off in tangent track should be no more than 25 millimeters (1 inch).

3.2.6 Speed, Curvature, and Superelevation: Theory and Basis of Criteria

This section summarizes the basis of design for speed, curvature, and superelevation. This material is based on information provided by Nelson [7], but has been condensed and modified as necessary for the specific application to current LRT designs and to include the use of metric units.

3.2.6.1 Design Speed in Curves

The background for recommended standards for actual superelevation, allowable superelevation unbalance, easement curves, and the length of superelevation runoffs will be reviewed in this section.

It takes more than 1 kilometer (0.62 miles) for a light rail vehicle to decelerate from 110 kph (70 mph) to 90 kph (55 mph), run through a 300-meter (1000-foot) circular curve and accelerate back to 110 kph (70 mph). The same curve designed for a reduction to 70 kph (45 mph) requires a length of about 1 2 kilometers (0 75 miles). Therefore, it is generally desirable to eliminate as many speed restrictions as possible and to maximize the design speed of all curves that must be designed with speed restrictions

3.2.6.2 Superelevation Theory

The design speed at which a light rail vehicle can negotiate a curve is increased proportionally by increasing the elevation of the outside rail of the track, thus creating a banking effect called superelevation.

When rounding a curve, a vehicle is subject to centrifugal force acting radially outward. The forces acting on the vehicle are illustrated in **Figure 3.2.3**. To counteract the effect of the centrifugal force (F_c), the outside rail of a curve is raised by a distance 'e' above the inside rail. A state of equilibrium is reached in which both wheels exert equal force on the rails; i.e., where 'e' is sufficient to bring the resultant force (F_r) to right angles with the plane of the top of the rails.

AREMA Manual, Chapter 5, gives the following equation to determine the distance that the outside rail must be raised to reach a state of equilibrium, where both wheels bear equally on the rails.

$$e = \frac{BV^2}{gr}$$

where, e = equilibrium superelevation in meters (feet)

B = bearing distance of track in meters (feet) usually 1.5 meters (5 feet).

V = velocity in meters (feet) per second

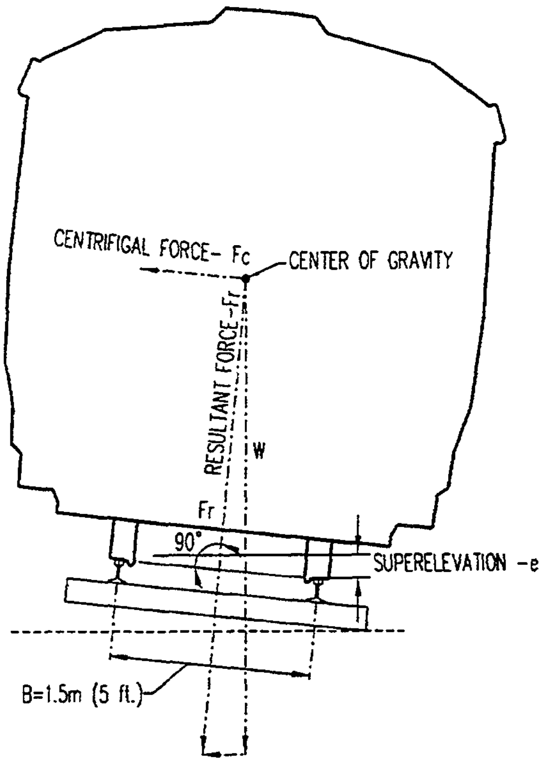


Figure 3.2.3 LRT Vehicle on Superelevated Track

g = force of gravity in meters per second per second, or meters/sec^2 (feet per second per second, or feet/sec^2)
r = radius in meters (feet)

To convert these terms to common usage, 'e' in meters (feet) is expressed as 'E' in millimeters (inches), 'B' is usually considered to be 1524 millimeters (60 inches) on standard gauge track. 'V' in meters per second (feet per second) is changed to 'V' in kph (mph). 'g' is equal to 9.8 meters/sec^2 (32.2 feet/sec^2), and 'r' is replaced by 1746.379/D (5730/D) in meters (feet), where 'D' is equal to the decimal degree of curvature. The revised formula is as follows:

$$E=\frac{1{,}524V^2}{(9.8)\left(\frac{1{,}746.379}{D}\right)\left(\frac{3{,}600}{1{,}000}\right)^2}=\frac{1{,}524V^2}{\frac{221{,}804}{D}}$$

$$\left[E=\frac{60V^2}{(32.2)\left(\frac{5{,}730}{D}\right)\left(\frac{3{,}600}{5{,}280}\right)^2}=\frac{60V^2}{\frac{85{,}772}{D}}\right]$$

thus;

$$E=\frac{V^2D}{1430}\text{ or }E=0\ 0069V^2D\quad\left[E=\frac{V^2D}{1430}\text{ or }E=0\ 0007V^2D\right]$$

and conversely;

$$V^2=\frac{145.5E}{D}\text{ or }V=\sqrt{\frac{145.5E}{D}}\quad\left[V^2=\frac{1430E}{D}\text{ or }V=\sqrt{\frac{1430E}{D}}\right]$$

These are the standard equations for equilibrium superelevation most commonly used in track design.

3.2.6.3 Actual Superelevation

Most railway route design texts recommend an absolute limit of 200 millimeters (8 inches) of actual superelevation for passenger operations unless slow moving or freight traffic is mixed with passenger traffic. As noted previously, LRT superelevation is generally limited to 150 millimeters (6 inches) or less.

All railroads administered by the Federal Railroad Administration (FRA) are limited to 150 millimeters (6 inches) of superelevation, primarily because the FRA mandates that all track that is a part of the nation's general railroad system must be capable of handling mixed traffic. Track that is not part of the general railroad system, or is used exclusively for rapid transit service in a metropolitan or suburban area, generally does not fall with the jurisdiction of the FRA. This includes the vast majority of LRT systems.

In view of the foregoing, railways that are not administered by the FRA may, when appropriate, use up to 200 millimeters (8 inches) of actual superelevation on curved track. This has been applied to at least two North American transit systems. However, it is more common to limit maximum actual superelevation to 150 millimeters (6 inches) on LRT systems, as it becomes more difficult to consistently maintain ride comfort levels at higher actual superelevations.

3.2.6.4 Superelevation Unbalance

The equations in the previous section are expressed in terms of a single equilibrium speed. Light rail vehicles often run at different speeds on the same segment of track. The variance from the so-called balanced speed concept is termed superelevation unbalance.

Superelevation unbalance may be defined as the difference between actual superelevation and that superelevation required for true equilibrium of the LRT vehicle traversing a curve.

If we call the superelevation unbalance E_u and the actual applied superelevation E_a, the formulae from the previous section may be restated as:

$$V^2 = \frac{145\ 5(E_a + E_u)}{D} \qquad \left[V^2 = \frac{1430(E_a + E_u)}{D}\right]$$

or

$$V = \sqrt{\frac{145.5(E_a + E_u)}{D}} \qquad \left[V = \sqrt{\frac{1430(E_a + E_u)}{D}}\right]$$

and;

$$E_a = 0.0069\ V^2D - E_u\ [E_a = 0\ 0007\ V^2D - E_u]$$

Limited superelevation unbalance is intentionally incorporated in most curve design speed calculations to avoid the effects of persistent underspeed operation—including passenger discomfort and excessive rail flow on the low (inside) rail of the curve.

Allowable superelevation unbalance varies among transit facilities. For instance, MTA New York City Transit only allows 25 millimeters (1 inch), while the Delaware River Port Authority (Lindenwold High Speed Line) allows 115 millimeters (4.5 inches). Generally, it is recognized that 75 to 115 millimeters (3 to 4.5 inches) of superelevation unbalance is acceptable for LRT operations, depending upon the vehicle design.

It should also be noted that Amtrak, with the approval of the FRA, raised its superelevation unbalance limit from 75 millimeters (3 inches) to 115 millimeters (4.5 inches) for intercity passenger trains.

In Sweden, Norway, West Germany, and France, intercity railways commonly employ from 100 to 150 millimeters (4 to 6 inches) of superelevation unbalance, and occasionally use even higher unbalance for specific applications.

The AREMA Manual for Railway Engineering (1985-86) states:

> "Equipment designed with large center bearings, roll stabilizers and outboard swing hangers can negotiate curves comfortably at greater than 75 millimeters (3 inches) of unbalanced superelevation because there is less body roll." ... " If the roll angle is less than 1°-30' experiments indicate that cars can negotiate curves comfortably at 115 millimeters (4.5 inches) of unbalanced elevation."

The preceding comments also generally apply to LRT vehicles as well.

In other words, a curve without any actual superelevation (E_a) can be safely and comfortably negotiated at a velocity requiring 115 millimeters (4.5 inches) of superelevation. A greater operating speed would result in an uncomfortable ride. Hence, a speed requiring no more than 115 millimeters (4.5 inches) of additional superelevation for equilibrium than is actually used is within a range for comfortable speed. Actual superelevation for maximum comfortable speed (E_c) may be expressed as:

$$E_c = 0.0069\,V^2D - 115\ [E_c = 0.0007V^2D - 4.5]$$

Thus, if an LRT vehicle is of modern design, it is appropriate to use up to 115 millimeters (4.5 inches) of superelevation unbalance as a parameter in the design of track curves.

It also should be noted, however, that a greater superelevation unbalance creates an increased impact on maintenance of vehicles and track. Conversely, operation closer to balance speed results in a more comfortable ride and less impact on the vehicle and track. Therefore, given equal speeds and circumstances it is preferable to maximize actual superelevation and minimize superelevation unbalance to reduce the effects of centrifugal force upon the passengers, vehicles, track structures, and roadbed.

3.2.6.5 Determination of Curve Design Speed

The calculation of design speed in curves is dependent on the vehicle design and passenger comfort. In addition to the preceding guidelines, curve design speed can be determined from the following principles if specific vehicle performance characteristics are known. This analysis is also necessary if the vehicle dimensions are significantly different than the LRT vehicles described in Chapter 2.

3.2.6.5.1 Categories of Speeds in Curves

Speed in curves may be categorized as follows;

- Overturning Speed: The speed at which the vehicle will derail or overturn because centrifugal force overcomes gravity.
- Safe Speed: The speed limit above which the vehicle becomes unstable and in great danger of derailment upon the introduction of any anomaly in the roadway.
- Maximum Authorized Speed (MAS): The speed at which the track shall be designed utilizing maximum allowable actual superelevation and superelevation unbalance.
- Signal Speed: The speed for which the signal speed control system is designed Ideally, signal speed should be just a little faster than the speed at which an experienced operator would normally operate the vehicle so that the automatic overspeed braking system is not deployed unnecessarily.

3.2.6.5.2 Overturning Speed

When the horizontal centrifugal forces of velocity and the effects of curvature overcome the vertical forces of weight and gravity, causing the resultant to rotate about the center of gravity of the vehicle and pass beyond the bearing point of the track, derailment or overturning of the vehicle will occur. This is diagrammed in **Figure 3.2.4**.

Overturning speed is dependent upon the height of the center of gravity above the top of the rail (h) and the amount that the center of gravity moves laterally toward the high rail (x)

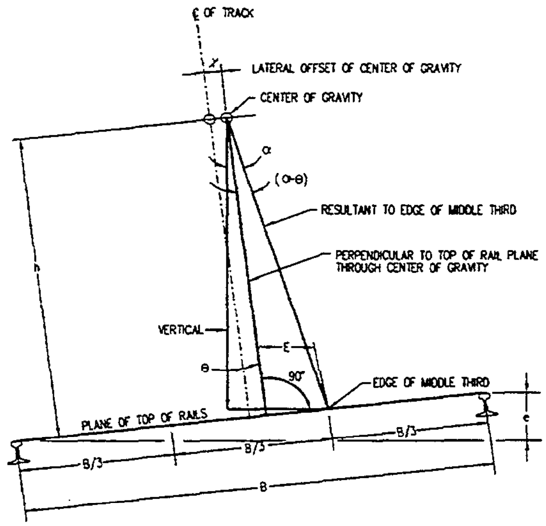


Figure 3.2.4 Force Diagram of LRT Vehicle on Superelevated Track

The formula for computing superelevation unbalance for 'Overturning Speed E_u' is derived from the theory of superelevation:

$$\text{Overturning Speed } E_u = Be/h$$

where: B = rail bearing distance = 1524 millimeters (60 inches)
e = B/2 – x
h = height of center of gravity = 1270 millimeters (50 inches)

If 'x' = 50 mm (2 in.), then e = (1524/2) - 50 = 712 millimeters (28 inches)
then:

$$\text{Overturning Speed } E_u = \frac{(1524)(712)}{1270} = 854 \text{ millimeters } (33.6 \text{ inches})$$

and

$$\text{Overturning Speed } V = \sqrt{\frac{145.5(E_a + E_u)}{D}}$$

For example, if 'E_a' is given as 150 millimeters (6 inches) and the decimal degree of curvature 'D' is equal to 5.00°, then

$$\text{Overturning Speed } V = \sqrt{\frac{(145.5)(150 + 854)}{5}}$$

$$= 170.9 \text{ kph } (106 \text{ mph})$$

Obviously, overturning speed should be far in excess of the curve's maximum authorized speed

3.2.6.5.3 Safe Speed

It is generally agreed that a rail vehicle is in a stable condition while rounding a curve if the resultant horizontal and vertical forces fall within the middle third of the distance between the wheel contact points. This equates to the middle 508 millimeters (20 inches) of the 1524-millimeter (60-inch) bearing zone 'B' indicated in Figure 3.2.4.

Safe speed is that arbitrary condition where the vehicle force resultant projection stays within the one-third point of the bearing distance. That speed is entirely dependent upon the location of the center of gravity, which is the height above the top of rail 'h' and the offset 'x' of the center of gravity toward the outside rail. From the theory of superelevation, we derive the formula for computing superelevation unbalance for maximum safe speed 'E_u':

$$\text{Safe Speed } E_u = Be/h$$

where: B = rail bearing distance = 1524 millimeters (60 inches)
e = B/6 - x. If 'x' = 50 mm (2 in.),
then: e = (1524/6) - 50 = 204 millimeters (8 inches)
h = height of center of gravity = 1270 millimeters (50 inches)

then.

$$\text{SafeSpeed } E_u = \frac{(1524)(204)}{1270} = 245 \text{ millimeters } (9.6 \text{ inches})$$

and

$$\text{MaximumSafeSpeed } V = \sqrt{\frac{145\ 5(E_a + E_u)}{D}}$$

Using the example given for overturning speed, if 'E_a' is given as 150 millimeters (6 inches) and the decimal degree of curvature 'D' is equal to 5.00°, then:

$$\text{Maximum Safe Speed } V = \sqrt{\frac{(145.5)(150 + 245)}{5}}$$
$$= 107\text{kph } (66.5 \text{ mph})$$

3.2.6.5.4 Determination of Superelevation Unbalance Values for Safe and Overturning Speeds

Table 3.2.3 lists reasonable values for 'E_u' for safe speed and overturning speed for various equipment characteristics. For reference, a typical transit car has a typical center of gravity shift (x) and height (h) of 63.5 mm and 1270 mm, respectively, and a freight train diesel locomotive has a typical 'x' and 'h' values of 75 mm and 1575 mm, respectively.

Using the example of a typical transit car with a center of gravity shift/height of 63.5 mm/1270 mm, an 'E_u' of 229 millimeters (9 inches) for safe speed and an 'E_u' of 838 millimeters (33 inches) for overturning speed are calculated. MAS and signal speed can then be determined from the safe speed results.

3.2.6.6 Easement Curves

Superelevated circular curves usually require easement curves to control the rate of lateral acceleration exerted upon the track, the passengers, and the vehicle. Easement curves are usually spirals with radii changing from infinity to the radius of the circular curve. Spiral curves also provide the ramp for introducing superelevation into the outside rail of the curve. Superelevation is normally runoff entirely within the spiral curve.

3.2.6.6.1 Length of Easement Curves

Safety and comfort will usually limit operating speed and dictate the length of transition spirals. As a general rule, any speed and

Table 3.2.3
Safe and Overturning Speed E_u Limits

		SAFE: e = 254 – x						OVERTURNING: e = 762 - x					
	x (mm)	0	25	50	64	75	100	0	25	50	64	75	100
h (mm)	e (mm)	250	225	200	190	175	150	760	735	710	698	685	660
1016	E_u	381	343	305	287	267	229	1,143	1,105	1,067	1,049	1,029	991
1270	E_u	305	274	244	229	213	183	914	884	853	838	823	792
1524	E_u	254	229	203	190	178	152	762	737	711	698	686	660
1575	E_u	246	221	196	185	173	147	737	714	688	676	663	640
1778	E_u	218	196	175	163	152	130	653	632	610	600	587	566
2032	E_u	190	173	152	142	135	114	572	554	533	523	516	495
2134	E_u	180	163	145	137	127	109	544	526	508	498	490	472

transition that provides a comfortable ride through a curve is well within the limits of safety.

Determining easement curve length allows for establishment of superelevation runoff within the allowable rate of increase in lateral acceleration due to cant deficiency (superelevation unbalance). Also, the transition must be long enough to limit possible racking of the vehicle frame and torsional forces from being introduced to the track structure by the moving vehicle.

When an LRT vehicle operating on straight (tangent) track reaches a circular path, the vehicle axles must be set at a new angle, depending upon the radius of the curve. This movement is not done instantly but over a measurable time interval, thus creating the need for a transitional curve, the length of which equals speed multiplied by time.

3.2.6.6.1.1 Length Based upon Passenger Comfort and Superelevation Unbalance. It is generally recognized by FRA, AREMA, Amtrak, OSHA, and many other applicable authorities that the maximum acceptable rate of acceleration of cant deficiency, or superelevation unbalance, for passenger comfort is 0.10 g, where 'g' is 9.8 meters per second per second (32.2 feet per second per second).

The change in the rate of acceleration from zero to 0.10 g should not exceed 0.03 g per second. Thus the minimum time needed to attain the maximum lateral acceleration will be:

$$\frac{\text{Max. Rate of Accel.}}{\text{Max. Rate of Change}} = \frac{0.10g}{0.03g} = 3.33 \text{ seconds}$$

Therefore the time factor for determining the length of the spiral required is 3.33 seconds multiplied by the speed of the vehicle.

Converting to kilometers per hour (miles per hour) the formula may be expressed as:

$$L_S(\text{meters}) = V(\text{kph})\frac{1000}{3600} \times 3.33$$
$$= 0.925V(\text{kph})$$
$$[L_S(\text{feet}) = 4.89V(\text{mph})]$$

Assuming that 115 millimeters (4.5 inches) is the maximum allowable superelevation unbalance, a formula to determine the length of the spiral necessary to ensure passenger comfort can be stated as:

$$L_S = \frac{0.925}{115}VE_U \text{ or } L_S = 0.008VE_U$$
$$\left[L_S = \left(\frac{4.89}{4.5}\right)VE_U \text{ or } L_S = 1.09VE_U\right]$$

3.2.6.6.1.2 Length based upon Superelevation. AREMA Manual, Chapter 5, gives the following formula for determining the length of an easement spiral curve:

$$L_S(\text{meters}) = 0.75E_U(\text{millimeters})$$
$$[L_S(\text{feet}) = 62E_a(\text{inches})]$$

In this equation, 'L_s' equals the length of the spiral and 'E_a' equals actual superelevation. The only criterion for establishing minimum spiral length is actual superelevation with no consideration for speed. For 150 millimeters (6 inches) of elevation, this produces a spiral 113 meters (372 feet) long.

This formula is based on the long-term structural integrity of a 26-meter (85-foot) long intercity passenger car. Most LRT vehicles can easily tolerate twice this rate of change. Therefore, a normal value for the minimum spiral length due to vehicle consideration is:

$$L_s = 0.38\ E_a \qquad [L_s = 31\ E_a]$$

The AREMA Manual criteria is somewhat conservative for LRT design in this respect.

As indicated above, the AREMA Manual determination of spiral length as a function of the runoff of actual superelevation is based on a 26-meter (85-foot) length car with 19-meter (62-foot) truck centers. This indicates that, for a 1,435-millimeter (4 35-foot) gauge, the minimum ratio of superelevation change across truck centers is 1:744. This is an empirical value that accounts for track cross-level tolerances, car suspension type, and fatigue stresses on the vehicle sills. Also note that the AREMA Manual formula is applicable to both passenger and freight cars.

Light rail vehicles have a far greater range of suspension travel than freight or intercity passenger cars. The magnitude of the LRV frame twist is relatively small compared to the nominal LRV suspension movement. The maximum actual superelevation runoff rate and minimum ratio of superelevation change across truck centers are thus not fixed values, but are functions of the LRV truck center distance.

Another service proven, although conservative, approach to establishing minimum criteria for spiral length can be derived from Amtrak's Specification for Construction and Maintenance of Track, MW-1000. Amtrak uses 75 to 115 millimeters (3 to 4.5 inches) of superelevation unbalance on curves, comparable to many LRT systems. MW-1000, Part I, Paragraph 213.63 states that for Class 3 Track, the maximum rate of superelevation runoff may not be more than 188:1 (2 inches in 31 feet). MW-1000, Part II, Paragraph 59.2 also states that the rate of change should not be more that 744:1 (0.5 inch per 31 feet) at 80 kph (50 mph). With the maximum rate of elevation as 744:1 and maximum rate of change of 188:1, the length of the spiral is 57 meters (186 feet) with 150 (6 inches) of superelevation.

Therefore, L_s can be derived from:

$$L_s = 0.0046\ Ve_a \qquad [L_s = 0.62\ VE_a]$$

where: L_s = spiral length in meters (feet)
V = speed in kph (mph)
E_a = actual superelevation in millimeters (inches)

Amtrak's MW-1000 Manual also shows that, for Class 5 track, the maximum rate of superelevation runoff may not be more than 3372:1 (1 inch in 31 feet) and that the maximum rate of change of elevation should not exceed 1488:1 (0.25 inch per 31 feet) for 160 kph (100 mph). With the maximum rate of elevation as 372:1 and maximum rate of change of 1488:1, the length of the spiral is 76 meters (248 feet) with 100 millimeters (4 inches) of superelevation.

Therefore again:

$$L_s = 0.0046\ Ve_a\ [L_s = 0.62\ VE_a]$$

If 'E_a' is increased to 150 millimeters (6 inches) and 'V' remains at 162 kph (100 mph) then:

$$L_s = (0.0046)(162)(150) = 112 \text{ meters}$$
$$[L_s = (0.62)(100)(6) = 372 \text{ feet}]$$

This shows that the AREMA formula is safe and conservative for speeds up to 162 kph (100 mph), but that other methods for determining spiral length should be used when shorter lengths are required for cases of lower operating speed.

3.2.6.6.1.3 Comparison of Spiral Lengths Based Upon Actual vs Unbalanced Elevation. From Section 3.2.6.6.1.1, based on superelevation unbalance, minimum spiral curve length is determined by:

$$L_s = 0.008\ V\ E_u \quad [L_s = 1.09\ VE_u]$$

An example using the above equation where V = 80 kph (50 mph) and E_u = 115 millimeters (4.5 inches) yields:

$$L_s = (0.008)(80)(115) = 74 \text{ meters } (242 \text{ feet})$$

From Section 3.2.6.6.1.2, based on actual superelevation runoff, minimum spiral curve length is determined by:

$$L_s = 0\ 0046\ V\ E_a \qquad [L_s = 0.62\ VE_a]$$

An example using the formula above, where V = 80 kph (50 mph) and E_a = 150 millimeters (6 inches) yields:

$$L_s = (0.0046)(80)(150) = 56 \text{ meters } (186 \text{ feet})$$

If E_a = 200 millimeters (8 inches), the minimum spiral length values would be very close for the two cases above. In LRT design, the vehicle can generally handle twice the actual superelevation runoff indicated in the above example. Therefore, it can be said that passenger comfort criteria will generally be the main factor in determining minimum spiral length.

3.3 VERTICAL ALIGNMENT

The vertical alignment of an LRT alignment is composed of constant grade tangent segments connected at their intersection by parabolic curves having a constant rate of change in grade. The nomenclature used to describe vertical alignments is illustrated in **Figure 3.3.1**.

The percentage grade is defined as the rise or fall in elevation, divided by the length. Thus a change in elevation of 1 meter over a distance of 100 meters would be defined as a 1% grade.

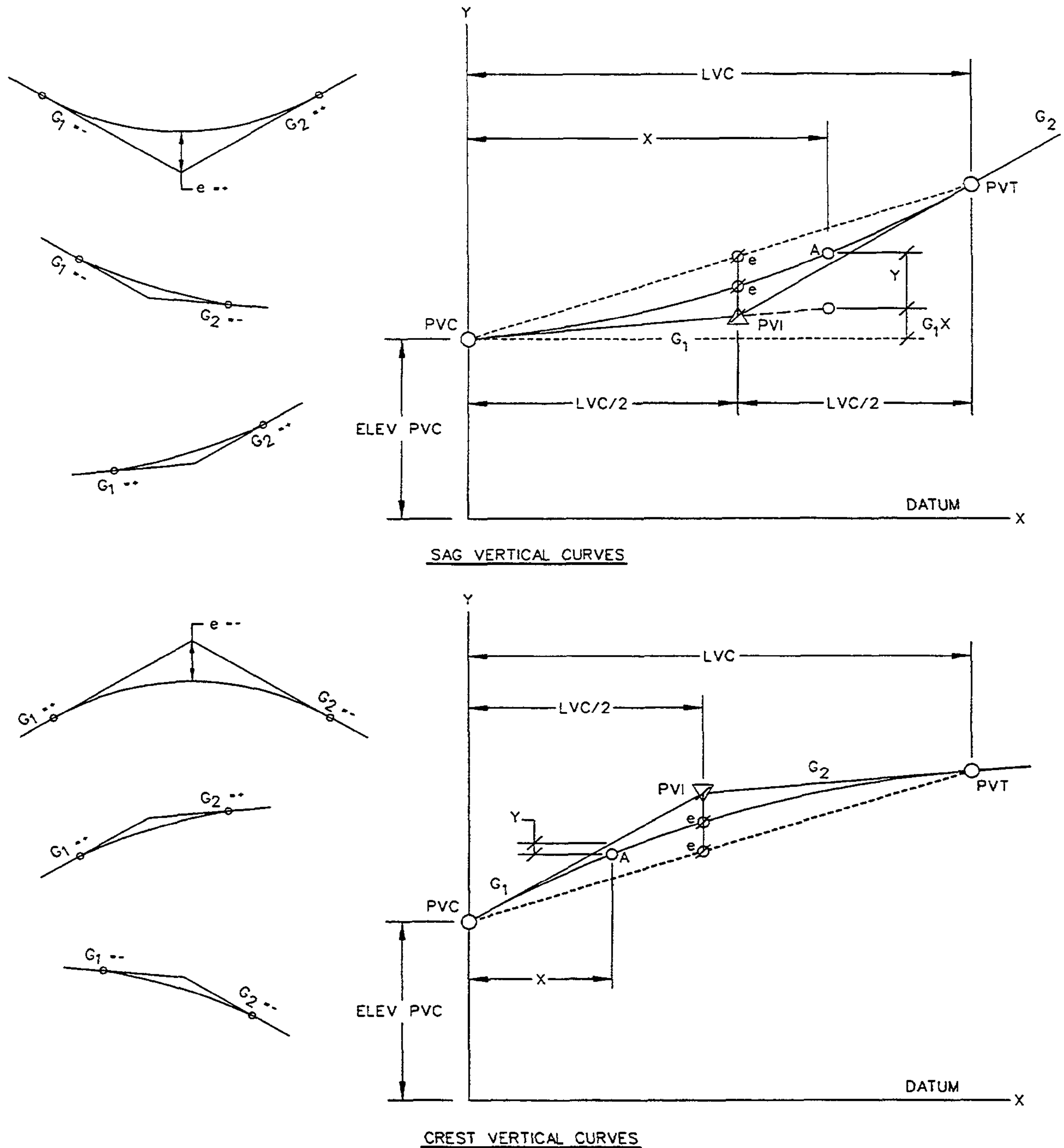
The profile grade line in tangent track is usually measured along the centerline of track between the two running rails and in the plane defined by the top of the two rails. In superelevated track, the inside rail of the curve normally remains at the profile grade line and superelevation is achieved by raising the outer rail above the inner rail. One exception to this recommendation is in tunnels, where the superelevation may be rotated about the centerline of track in the interest of improved vertical clearances.

The vehicle's performance, dimensions, and tolerance to vertical bending stress dictate criteria for vertical alignments. The following criteria are used for proposed systems using a modern low-floor vehicle. It can be used as a basis of consideration for general use.

3.3.1 Vertical Tangents

The minimum length of constant profile grade between vertical curves should be as follows:

Condition	Length
Main Line Desired Minimum	30 meters (100 feet) or 0.57V, three times the design speed in kph (mph), whichever is greater
Main Line Absolute Minimum	12 meters (40 feet)



PARABOLIC VERTICAL CURVE FORMULAS:

$$e = \left(\frac{G_2 - G_1}{8}\right) LVC = \frac{1}{8} A\,(LVC); \qquad r = \left(\frac{G_2 - G_1}{LVC}\right) = \text{RATE OF CHANGE IN GRADE}$$

$$y = \frac{1}{2}\left(\frac{G_2 - G_1}{LVC}\right) x^2 = \frac{1}{2} r x^2$$

$$\text{ELEV A} = \left(\frac{r}{2}\right) x^2 + XG_1 + \text{ELEV PVC}$$

Figure 3.3.1 Vertical Curve Nomencalture

In embedded track in urban areas, where the need to conform to existing street profiles makes compliance with the above criteria impracticable, the above requirement is usually waived Where a tangent between vertical curves is shorter than 12 meters (40 feet), consideration should be given to using reverse or compound vertical curves. This avoids abrupt changes in vertical acceleration that could result in both passenger discomfort and excessive vehicle suspension system wear.

3.3.2 Vehicle Length Criteria

This topic is covered in Section 2.4 of this handbook.

3.3.3 Vertical Grades

Maximum grades in track are controlled by vehicle braking and tractive efforts. On main line track, civil drainage provisions also establish a minimum recommended profile grade. In yards, shops, and at station platforms, there is usually secondary or cross drainage available. Thus, grades in the range of 0.00% to 0.04% are acceptable.

As a guideline, the following profile grade limitations are recommended for general use in LRT design:

Main Line Tracks

Maximum Sustained Grade, Unlimited Length	4.0%
Maximum Sustained Grade with Up to 750 Meters (2500 feet) between PVIs of Vertical Curves	6.0%
Maximum Short Sustained Grade with No More than 150 Meters (500 Feet) between PVIs of Vertical Curves	7.0%
Minimum Grade for Drainage on Direct Fixation Track	0.2%

No minimum grade is specified at passenger stations provided adequate track drainage can be maintained. In urban areas, the existing street profile may govern the profile grade within the station. In this case, the profile grade may exceed 2.0%, but should be restricted to a maximum of 3.5%.

Yard Tracks	
Desired	0 0%
Maximum	1.0%
Yard Storage & Pocket Tracks	
Desired	0.0%
Maximum	0.2%

All tracks entering a yard should either be level, sloped downward away from the main line, or dished to prevent rail vehicles from rolling out of the yard onto the main line. For yard secondary tracks, a slight grade, usually between 0.35% and 1.00%, is recommended to achieve good track drainage at the subballast level.

Through storage tracks generally have a sag in the middle of their profile to prevent rail vehicles from rolling to either end. It is recommended that the profile grade of a stub end storage track descend toward the stub end and, if it is adjacent to a main line or secondary track, it should be curved away from that track at its stub end. If it is necessary for the profile grade of a storage track to slope up toward the stub end, the grade should not exceed 0.20%.

Tracks located within maintenance shops and other buildings are generally level.

3.3.4 Vertical Curves

All changes in grade are connected by vertical curves. Vertical curves are defined by parabolic curves having a constant rate of change in grade. Parabolic curves are, for all practical purposes, equivalent to circular curves for LRT design, but parabolic curves are easier to calculate and are thus preferable for this purpose.

As a guideline, the following vertical curve criteria are recommended for general use in LRT designs:

3.3.4.1 Vertical Curve Lengths

The length of vertical curves can be determined as follows:

- Desired Length: LVC = 60A (LVC = 200A)
- Preferred Minimum Length: LVC = 30A (LVC = 100A)
- Absolute Minimum Length:
  - Crest Curves:

$$LVC = \frac{AV^2}{215} \qquad \left[LVC = \frac{AV^2}{25}\right]$$

  - Sag Curves:

$$LVC = \frac{AV^2}{387} \qquad \left[LVC = \frac{AV^2}{45}\right]$$

where: LVC = length of vertical curve, in meters (feet)

A = $(G_2 - G_1)$ algebraic difference in gradients connected by the vertical curve, in percent

G_1 = percent grade of approaching tangent

G_2 = percent grade of departing tangent

V = design speed, in kph (mph)

Both sag and crest vertical curves should have the maximum possible length, especially if approach and departure tangents are long. Vertical broken back curves and short horizontal curves at sags and crests should be avoided.

The minimum equivalent radius of curvature for vertical curves located on main line tangent track should not be less than 250 meters (820 feet) for crests and 350 meters (1150 feet) for sags. This equivalent radius of curvature can be calculated from the following formula:

$$R_v = \frac{LVC}{0.01(G_2 - G_1)} \qquad \left[R_v = \frac{LVC}{0.01(G_2 - G_1)}\right]$$

where: R_v = minimum radius of curvature of a vertical curve in meters (feet).

Minimum vertical curve length and/or design speed may be governed by the overhead contact system (OCS) due to the maximum permissible rate of separation or convergence between the track grade and the contact wire gradient. Coordination with the OCS designer is strongly recommended to ensure compliance with this limitation.

3.3.5 Vertical Curves, Special Conditions

3.3.5.1 Reverse Vertical Curves

Reverse vertical curves are feasible, provided each curve conforms to the requirements stated in Section 3.3.4 and the restrictions imposed by the LRT vehicle design.

3.3.5.2 Combined Vertical and Horizontal Curvature

Where possible, areas of combined vertical and horizontal curvature should be avoided. Where areas of combined vertical and horizontal curvature cannot be avoided, the geometry should not be more severe than a 25-meter (82-foot) radius horizontal combined with a 250-meter (820-foot) equivalent radius vertical crest curve. Again, this criterion must be conformed with the vehicle design.

3.3.6 Station Platform Alignment Considerations

In addition to the stringent track installation tolerances imposed by the Americans with Disabilities Act (ADA), there are alignment considerations that must be included in LRT trackwork. All LRT systems must provide level boarding. This applies whether the LRT vehicle uses a high- or low-floor system.

Consequently, a horizontal curve cannot be located within a vehicle length of the platform; otherwise, the ADA platform gap requirements will be virtually impossible to achieve.

3.3.6.1 Horizontal Alignment of Station Platforms

At station platforms, the horizontal alignment should be tangent throughout the entire length of the platform. The tangent should be extended beyond both ends of the platform as follows:

Condition	Minimum Tangent Length
Desired Minimum	25 meters (75 feet)
Preferred Minimum	20 meters (60 feet)
Absolute Minimum	15 meters (45 feet)

3.3.6.2 Vertical Alignment of Station Platforms

The profile at stations should be on a vertical tangent that extends 12 meters (40 feet) beyond each end of the platform.

Station Area Grades

Desired: 0.0%

Maximum: 2.0%

No minimum grade is necessary at passenger stations, provided that adequate track drainage can be maintained.

3.3.7 Joint LRT-Railroad/Freight Tracks

Railroad tracks to be relocated or in joint usage areas are designed in conformance with the requirements of the operating railroad and the AREMA Manual, except as recommended herein As a guideline, recommended criteria are as follows:

3.3.7.1 Horizontal Alignment

The horizontal alignment for joint LRT-railroad/freight tracks consists of tangent, circular curves, and spiral transitions based on the preferred maximum LRV design speed and the required FRA freight class of railroad operation Lead tracks and industrial spurs generally do not require spiral transitions.

Curves adjacent to turnouts on tracks that diverge from the main track should be designed for the maximum allowable speeds of the adjoining turnouts.

Yard track should be designed for a minimum of 25 kph (15 mph). Lead track and industrial sidetracks should be designed for a minimum of 16 kph (10 mph).

3.3.7.2 Tangent Alignment

For joint LRT-railroad/freight main tracks, the desired tangent length between curves should be 90 meters (300 feet), with an absolute minimum of 30 meters (100 feet). For lead tracks and industrial spurs, a minimum tangent distance of 15 meters (50 feet) should be provided between curve points. All turnouts should be located on tangents.

3.3.7.3 Curved Alignment

The maximum desired degree of curvature for railroad main line tracks should be either 3° or the maximum presently in use along the route, but should not in any case exceed 9° 30'. The maximum curvature for lead tracks and industrial sidetracks should be 12°. In extreme cases, revisions to existing industrial sidetracks may be designed with sharper curves that match the existing values. Exceptions to the above criteria may be permitted as authorized by both the transit authority and the operating freight railroad. The minimum length of circular curves for main line tracks should be 30 meters (100 feet).

3.3.7.4 Superelevation

Superelevation should be provided on main line and secondary line tracks only, based on the following formula:

$$E_a = 100\left(\frac{V^2}{R}\right) - 10 \qquad \left[E_a = 3.17\left(\frac{V^2}{R}\right) - 0.4\right]$$

where: E_a = actual superelevation in millimeters (inches)
V = curve design speed, in kph (mph)
R = radius of curve in meters (feet)

Values of actual superelevation (E_a) should be rounded to the nearest 6 millimeters (0.25 inch). In cases where the calculated value is less than 12 millimeters (0.5 inch), no actual superelevation (E_a) need be applied.

Under joint freight and LRT operating conditions, E_a should be obtained from the above formula until the calculated value reaches 75 millimeters (3 inches). E_a can be further increased to 100 millimeters (4 inches) to achieve desired speed with the approval of transit authority and the operating railroad.

3.3.7.5 Spiral Transitions

Spiral transition curves are generally used for railroad/freight main line and secondary line tracks only. Low-speed yard and secondary tracks without superelevation generally do not require spirals. Spirals should be provided on all curves where the superelevation required for the design speed is 12 millimeters (0.5 inch) or more. The maximum E_u for freight traffic is 37 millimeters (1.5 inches). Note that allowable LRT and railroad operating speeds along a given track may differ due to the difference in the maximum unbalance superelevation allowed for each mode and specific operating requirements.

As a guideline, the minimum length of a spiral in railroad track and joint use railroad and LRT track should be determined from the following formulae, rounded off to the next meter (or 5 feet), but preferably not less than 18 meters (60 feet).

$$L_s = 0.75\,E_a \qquad (L_s = 62\,E_a)$$
$$L_s = 0.009\,E_u\,V \qquad (L_s = 1.22\,E_u\,V)$$
$$L_s = 0.0083\,E_a\,V \qquad (L_s = 1.13\,E_a\,V)$$

where: L_s = minimum length of spiral, in meters (feet)
E_a = actual superelevation in millimeters (inches)
E_u = unbalanced superelevation in millimeters (inches)
V = curve design speed in kph (mph)

3.3.7.6 Vertical Alignment of Joint Use Tracks

3.3.7.6.1 General

The profile grade is defined as the elevation of the top of the low rail. Vertical curves should be defined by parabolic curves having a constant rate of grade change.

3.3.7.6.2 Vertical Tangents

The desired minimum length of vertical tangents is 90 meters (300 feet) with an absolute minimum value of 60 meters (200 feet). Turnouts should be located only on tangent grades.

3.3.7.6.3 Vertical Grades

On main line tracks, the preferred maximum grade should be 1.0%. This value may only be exceeded in cases where the existing longitudinal grade is steeper than 1.0%. Grades within horizontal curves are generally compensated (reduced) at a rate of 0.04% per horizontal degree of curvature. Locations where freight trains may frequently stop and start are compensated at a rate of 0.05% per degree of curvature. This compensation reduces the maximum grade in areas of curvature to reflect the additional tractive effort required to pull the train.

For yard tracks and portions of industrial sidetracks where cars are stored, the grades should preferably be 0.20% or less, but should not exceed 0.40%. Running portions of industrial sidetracks should have a maximum grade of 2.5%, except that steeper grades may be required to match existing tracks. Grade compensation is usually not required in railroad yard and industrial tracks.

3.3.7.6.4 Vertical Curves

Vertical curves are usually provided at all intersections of vertical tangent grades, except for where the total grade difference is less than 0.5%.

The lengths of vertical curves in railroad trackage should provide a rate of change of grade not exceeding 0.05% per station in sags and 0.10% per station in summits (rounded off to the next largest 30 meters, or 100 feet). Situations where this proves impossible to achieve may use shorter curves using the following formulae:

Crests: $LVC = 76A$ ($LVC = 250A$)
Sags: $LVC = 150A$ ($LVC = 500A$)

where: LVC = length of vertical curve in meters (feet)
$A = (G_2 - G_1)$ = algebraic difference in gradients connected by the vertical curve, in percent.
G_1 = percent grade of approaching tangent
G_2 = percent grade of departing tangent

If an existing railroad vertical curve is below the desired length, a replacement vertical curve with a rate of change of grade not exceeding that of the existing curve may be acceptable.

3.4 VEHICLE CLEARANCES AND TRACK CENTERS

This section discusses the minimum dimensions that must be established to ensure minimum clearances between the light rail vehicles and transit structures or other obstructions and to establish a procedure for determining minimum track center distances.

The provision of adequate clearances for the safe passage of vehicles is a fundamental concern in the design of transit facilities. Careful determination of clearance envelopes and enforcement of the resulting minimum clearance requirements during design and construction are essential to proper operations and safety.

The following discussion concentrates on the establishment of new vehicle clearance envelopes and minimum track centers. On existing LRT systems, this is normally established in the initial design criteria or by conditions in the initial sections of the transit system

3.4.1 Clearance Envelope

The clearance envelope (CE) is defined as the space occupied by the maximum vehicle dynamic envelope (VDE), plus effects due to curvature and superelevation, construction and maintenance tolerances of the track structure, construction tolerances of adjacent wayside structures, and running clearances. The relationship between the vehicle and clearance envelope can thus be expressed as follows: [14]

$$CE = VDE + TT + C\&S + RC$$

where:
CE = Clearance Envelope
VDE = Vehicle Dynamic Envelope
TT = Trackwork Construction and Maintenance Tolerances
C&S = Vehicle Curve and Superelevation Effects
RC = Vehicle Running Clearance

The clearance envelope represents the space into which no physical part of the system, other than the vehicle itself, must be placed, constructed, or protrude.

A second part of the clearance equation is what is termed structure gauge, which is basically the minimum distance between the centerline of track and a specific point on the structure.

Although structure gauge and clearance envelope elements are often combined, it is not advisable to construct a clearance envelope that includes wayside structure clearances and tolerances, as the required horizontal or vertical clearances to different structures may vary significantly.

The factors used to develop the clearance envelope are discussed in further detail in the following sections. It should be noted that in some LRT designs, some of the factors listed above are combined; for example, the trackwork construction and maintenance tolerances are frequently included in the calculation of the vehicle dynamic envelope. [2] Regardless of how the individual factors are defined, it is important that all of these items are included in the determination of the overall clearance envelope.

3.4.1.1 Vehicle Dynamic Envelope

Determination of the VDE begins with the cross sectional outline of the static vehicle. The dynamic outline of the vehicle is then developed by making allowances for car body movements that occur when the vehicle is operating on level tangent track. These movements represent the extremes of car body displacement that can occur for any combination of rotational, lateral, and vertical car body movements when the vehicle is operating on level tangent track.

The following items are typically included in the development of the VDE: [15,16]

1. Static vehicle outline
2. Dynamic motion (roll) of springs and suspension/bolsters of vehicle trucks
3. Vehicle suspension side play and component wear
4. Vehicle wheel flange and radial tread wear
5. Maximum truck yaw (fishtailing)
6. Maximum passenger loading
7. Suspension system failure
8. Wheel and track nominal gauge difference
9. Wheel back-to-back tolerance
10. Rail fastener loosening and gauge widening during revenue service
11. Dynamic rail rotation
12. Rail cant deficiency

Some of these items, particularly Items 10 to 12, are relatively minor and are often combined into a single value.

The development of the VDE is typically the responsibility of the vehicle designer. The trackwork designer may have to estimate the values of Items 10 to 12. It is imperative that the vehicle designer include maintenance tolerances as well as the initial installation tolerances in the determination of the VDE. Typical values for vehicle-based maintenance factors include the following:

- Lateral wheel wear: 7.5 millimeters (0.30 inch)
- Nominal wheel-to-rail sideplay: 10.5 millimeters (0.405 inch)
- Vertical radial wheel wear: 25 millimeters (1 inch)

The VDE is usually represented as a series of exterior coordinate points with the reference origin at the track centerline at the top of rail elevation. The static vehicle outline is generally not used in track design except for the establishment of station platforms and associated station trackwork design at these locations.

3.4.1.2 Track Construction and Maintenance Tolerances

Track construction and maintenance tolerances should be included in the determination of the clearance envelope, whether as part of the VDE or as a separate clearance item. The track maintenance tolerances are generally far greater than the initial construction tolerances and thus take precedence for the purpose of determining clearances.

It should also be noted that direct fixation and ballasted trackwork have different track maintenance tolerances. It is possible to determine separate clearance envelopes for ballasted and direct fixation track, or to use the more conservative clearance envelope based on the ballasted trackwork case. Both options have been used in actual practice.

Trackwork-based factors to be considered in the development of the clearance envelope, with typical values, include the following:

- Lateral Rail Wear: 13 millimeters (0.50 inch)
- Lateral Maintenance Tolerance, Direct Fixation Track: 13 millimeters (0 50 inch)
- Lateral Maintenance Tolerance, Ballasted Track: 25 millimeters (1.00 inch)
- Vertical Maintenance Tolerance: 13 millimeters (0.50 inch)
- Cross Level Variance, Direct Fixation Track: 13 millimeters (0.50 inch)
- Cross Level Variance, Ballasted Track: 25 millimeters (1.00 inch)

Cross level variance creates a condition of vehicle rotation rather than lateral shift. Effects on the clearance envelope are similar to superelevation effects noted below.

3.4.1.3 Curvature and Superelevation Effects

In addition to the VDE and track maintenance factors, track curvature and superelevation have a significant effect on the determination of the clearance envelope. These effects will be covered separately. Some authorities consider the effects of curvature and superlevation as part of the VDE, and calculate separate VDE diagrams for each combination of curvature and superelevation. As a guideline, this handbook considers only one VDE and determines curvature and superelevation effects separately to establish multiple clearance envelopes.

3.4.1.3.1 Curvature Effects

In addition to the dynamic car body movements described above, car body overhang on horizontal curves also increases the lateral displacement of the VDE relative to the track centerline. For design purposes, both mid-car inswing (mid-ordinate) and end-of-car outswing (end overhang) of the vehicle must be considered.

The amount of mid-car inswing and end-of-car outswing depends primarily on the vehicle truck spacing, vehicle end overhang, and track curve radius. The truck axle spacing also has an effect on clearances, although it is relatively small and frequently ignored.[6] Refer to Section 2.3.2 for vehicle dynamic outline.

To determine the amount of vehicle inswing and outswing for a given curve radius, one of two formulas are generally used, depending on whether the vehicle axle spacing is known. Both methods are sufficiently accurate for general clearance envelope determinations for LRT vehicles.

If truck axle spacing effects are ignored, the effects of vehicle inswing and outswing are determined from the assumption that the vehicle truck centers.are located at the center of track, as shown on **Figure 3.4.1**. In this case, the vehicle inswing and outswing can be found from:

$$\text{Inswing} = M_O = R(1 - \cos a) \text{ and } a = \sin^{-1}\left(\frac{L_2}{2R}\right)$$

where: M_o = mid-ordinate of vehicle chord
R = track curve radius
L_2 = vehicle truck spacing

$$\text{Outswing} = R_O - R \quad R_O = \frac{L}{\cos b} \quad b = \tan^{-1}\left(\frac{L}{R - M_O}\right)$$

where: R = track curve radius
L = half of overall vehicle length

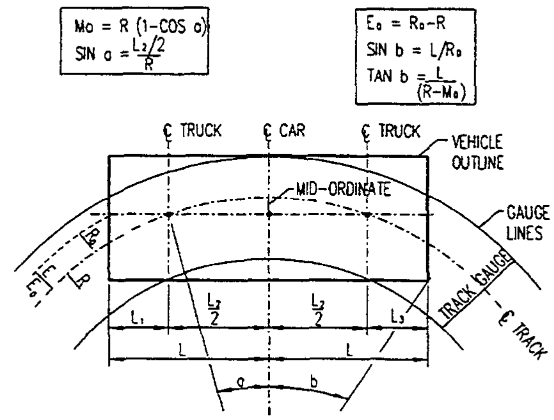


Figure 3.4.1 Horizontal Curve Effects on Vehicle Lateral Clearance

A somewhat more accurate calculation is provided from UIC 505-5, Enclosure VI, which is calculated by placing the four vehicle axles on the track centerline. In this publication, the vehicle inswing and outswing are determined from:

$$\text{Inswing} = M_O = \frac{(L_2/2)(L + L_2/2) - (P^2/4)}{2R}$$

$$\text{Outswing} = E_O = \frac{(L_2/2)(L - L_2/2) - (P^2/4)}{2R}$$

where: P = vehicle axle spacing

For single axle vehicles, such as those on low-floor articulated vehicles, the value of P in the UIC formulae is 0.

In determining the outswing of the vehicle, it must be noted that some vehicles have tapered ends, and that the clearance diagram is based on the worst-case between the vehicle end section and the full vehicle section away from the vehicle end.

When calculating the CE for horizontal curves with spirals, it is necessary to end the tangent clearance envelope at some distance, usually 15 meters (50 feet), before the track tangent-to-spiral (TS) point. The full curvature CE should begin 7.5 meters (25 feet) before the track spiral-to-curve (SC) point and after the curve-to-spiral (CS) point. Horizontal offsets of the CE are calculated by linear interpolation with sufficient accuracy for clearance purposes. For simple circular curves, the full curvature CE begins 15 meters (50 feet) before the point of curve (PC) and ends 15 meters (50 feet) beyond the point of tangency (PT). These distances are for a 25- to 28-meter (82- to 92-foot) long vehicle, very short LRT vehicles would require shorter distances.

The CE through turnouts is calculated based on the centerline radius of the turnout.

It is of interest to note that the vehicle designer does not always provide the calculations for the effects of horizontal curvature clearance, and that this task is frequently left to the trackwork or civil alignment engineer.

3.4.1.3.2 Superelevation Effects

Superelevation effects are limited to the vehicle lean induced by a specific difference in elevation between the two rails of the track, and should be considered independently of other effects on the CE. In determining the effects of superelevation, the shape of the VDE is not altered, but is rotated about the centerline of the top of the low rail of the track for an amount equal to the actual track superelevation.

This rotation is illustrated in **Figure 3.4.2**. For any given coordinate on the VDE, the equations indicated in Figure 3.4.1 are sufficiently accurate to convert the original VDE coordinate (x_T, y_T) into a revised clearance coordinate (x_2, y_2) to account for superelevation effects.

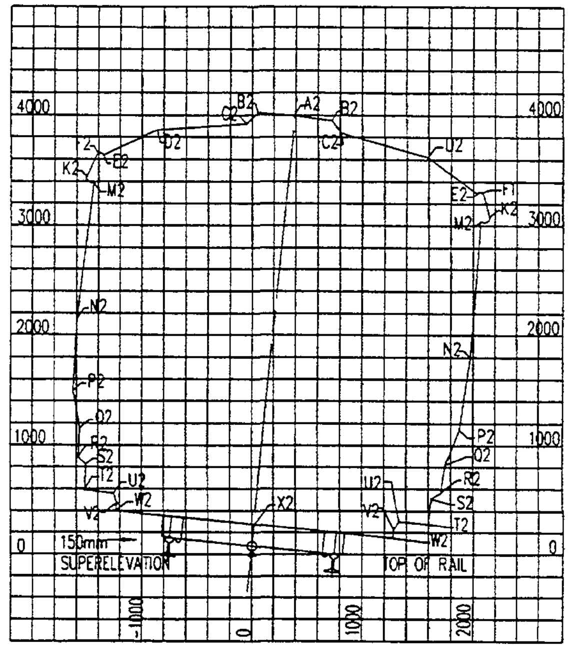


Figure 3.4.2 Dynamic Vehicle Outline Superelevation Effect on Vertical Clearances

3.4.1.5 Vehicle Running Clearance

The clearance envelope must include a minimum allowance for running clearance between the vehicle and adjacent obstructions or vehicles. Running clearance is generally measured horizontally (laterally) to the obstruction, although some clearance envelopes are developed with the running clearance added around the entire perimeter of the vehicle.

The most common general value assigned to running clearances is 50 millimeters (2 inches). Except at station platforms, which are special cases in LRT design, the 50 millimeters (2 inches) represents a minimum running clearance value.

Some items are occasionally assigned a higher minimum running clearance. These include structural members and adjacent vehicles. A typical assignment of running clearance criteria includes the following data:

- Minimum running clearance to signals, signs, platform doors, and other non-structural members: 50 millimeters (2 inches)
- Minimum running clearance to an emergency walkway envelope: 50 millimeters (2 inches)
- Minimum running clearance along an aerial deck parapet, walls, and all structural members: 150 millimeters (6 inches)
- Minimum running clearance to adjacent LRT vehicles: 150 millimeters (6 inches)

3.4.2 Structure Gauge

The second part of the clearance equation is what is termed structure gauge, which is basically the minimum distance between the centerline of track and a specific point on the structure. This is determined from the CE above, plus structure tolerances and minimum clearances to structures. Thus:

$$SG = CE + SC + ST + AA$$

where, SG = structure gauge
CE = clearance envelope
SC = required clearance to wayside structure
ST = wayside structure construction tolerance
AA = acoustic allowance

The required clearance to wayside structures may be specified separately from the running clearance described above. In other words, the running clearance envelope is stated as a constant value, usually 50 millimeters, and a separate required clearance criteria is specified for each type of wayside structure. Values of 50 to 150 millimeters (2 to 6 inches) are normally specified as minimum clearance from structures in the clearance envelope.

Construction tolerances for wayside structures include the construction and maintenance tolerances associated with structural elements outside of the track. These can include walls, catenary poles, and signal equipment. A minimum construction tolerance for large structural elements is normally 50 millimeters (2 inches), although soldier pile and lagging type walls may have a much larger tolerance requirement.

A second item that must be considered in construction tolerances is an allowance for chorded construction of tunnel walls, large precast aerial structure sections, and walkways. In lieu of exact construction information, a general guideline of a 15-meter (50-foot) chord for curve radii greater than 750 meters (2,500 feet), and 7.5-meter (25-foot) chords for smaller radius curves can be used as a basis for design.

Finally, provisions for present or future acoustical treatments are often required on walls and other structures. Typical values for this range from 50 to 75 millimeters (2 to 3 inches).

3.4.3 Station Platforms

Station platforms require special clearance considerations, especially since regulations such as the American with Disabilities Act cover the maximum permissible gap between the vehicle floor and platform edge.

It should be noted that current ADA regulations require a maximum vehicle-platform gap of 75 millimeters (3 inches) with the static vehicle located at the centerline of track. For high platforms or high block portions of station platforms, where applicable, this is usually not in conformance with other clearance criteria. Therefore, clearance at station platforms should be considered separate from all other structural clearances

This topic is also covered in the discussion of vehicle/track installation tolerances in Chapter 2 herein.

3.4.4 Vertical Clearances

Vertical clearances are normally set with a 100- to 150-millimeter (4- to 6-inch) allowance from the clearance envelope, including superelevation effects. Actual LRT operations normally do not require this amount of vertical clearance, but an allowance is usually required to accommodate future maintenance, particularly on ballasted trackwork.

3.4.5 Track Centers and Fouling Points

The minimum allowable spacing between tracks and the location of fouling points are determined using the same principles as those used for determining clearances to structures. Referring to the previous discussion on clearances, minimum track centers can be determined from the following equation, if catenary poles are not located between tracks:

$$TC = T_t + T_a + 2(OWF) + RC$$

where: TC = minimum track centers
T_t = half of vehicle CE toward curve center
T_a = half of vehicle CE away from curve center
RC = running clearance
OWF = other wayside factors (see structure gauge)

Where catenary poles are located between tracks, the minimum track centers are determined from:

$$TC = T_t + T_a + 2(OWF + RC) + P$$

where: TC = minimum track centers
T_t = half of vehicle CE toward curve center
T_a = half of vehicle CE away from curve center
RC = running clearance
OWF = other wayside factors (see structure gauge)
P = maximum allowable catenary pole diameter

Where the LRT track is designed for joint usage with freight railroads, the clearances mandated by the operating freight railroad generally predominates. The AREMA Manual contains useful information on general freight railway clearances, but the individual railroads also have specific clearance requirements that will supersede the AREMA recommendations.

3.5 REFERENCES

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[5] AREMA Manual, Chapter 5.

[6] Parsons Brinckerhoff-Tudor-Bechtel, "Basis of Geometrics Criteria," submitted to the Metropolitan Atlanta Rapid Transit Authority (Atlanta: MARTA, Aug. 1974), p. 3.

[7] Harvey S. Nelson, "Speed and Superelevation on an Interurban Electric Railway," presentation at APTA Conference, Philadelphia, PA, June 1991.

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