NOTE TO USERS

This reproduction is the best copy available.

UMI[®]

**DEVELOPMENT AND COST-EFFECTIVENESS OF NEW AND
IMPROVED 3D REVERSE HORIZONTAL CURVES WITH
INTERMEDIATE TANGENTS**

By

Amir Omar Abd El Halim, B.Eng

A thesis submitted to the
Faculty of Graduate Studies and Research
in partial fulfillment of the requirements
of the degree of

**Master of Applied Science
in Civil Engineering**

Department of Civil and Environmental Engineering
Carleton University
Ottawa, Ontario, Canada.

© 2004, Amir Omar Abd El Halim

Library and
Archives Canada

Bibliothèque et
Archives Canada

Published Heritage
Branch

Direction du
Patrimoine de l'édition

395 Wellington Street
Ottawa ON K1A 0N4
Canada

395, rue Wellington
Ottawa ON K1A 0N4
Canada

Your file Votre référence

ISBN: 0-612-97396-4

Our file Notre référence

ISBN: 0-612-97396-4

The author has granted a non-exclusive license allowing the Library and Archives Canada to reproduce, loan, distribute or sell copies of this thesis in microform, paper or electronic formats.

L'auteur a accordé une licence non exclusive permettant à la Bibliothèque et Archives Canada de reproduire, prêter, distribuer ou vendre des copies de cette thèse sous la forme de microfiche/film, de reproduction sur papier ou sur format électronique.

The author retains ownership of the copyright in this thesis. Neither the thesis nor substantial extracts from it may be printed or otherwise reproduced without the author's permission.

L'auteur conserve la propriété du droit d'auteur qui protège cette thèse. Ni la thèse ni des extraits substantiels de celle-ci ne doivent être imprimés ou autrement reproduits sans son autorisation.

In compliance with the Canadian Privacy Act some supporting forms may have been removed from this thesis.

Conformément à la loi canadienne sur la protection de la vie privée, quelques formulaires secondaires ont été enlevés de cette thèse.

While these forms may be included in the document page count, their removal does not represent any loss of content from the thesis.

Bien que ces formulaires aient inclus dans la pagination, il n'y aura aucun contenu manquant.

Canada

The undersigned recommend to the
Faculty of Graduate Studies and Research
acceptance of this thesis

**DEVELOPMENT AND COST-EFFECTIVENESS OF NEW AND
IMPROVED 3D REVERSE HORIZONTAL CURVES WITH
INTERMEDIATE TANGENTS**

Submitted by:

Amir Omar Abd El Halim, B.Eng

In partial fulfilment of the requirements

For the degree of

Master of Applied Science in

Civil Engineering

Thesis Co-Supervisor
Dr. Said Easa
Adjunct, Carleton University

Thesis Co-Supervisor
Dr. Paul Van Geel
Carleton University

Acting Chair, Dept. of Civil and Environmental Engineering
Dr. G. Hartley
Carleton University

ABSTRACT

Current design guides of the Transportation Association of Canada (TAC) and the American Association of State Highway and Transportation Officials (AASHTO) establish the minimum radius requirements for reverse horizontal curves by designing each arc of the reverse curve as a separate simple horizontal curve, on level terrain. This traditional design disregards the effects of both reverse horizontal curvature and vertical alignment. The purpose of this research is twofold: (1) to establish minimum radius and spiral length requirements for three-dimensional (3D) reverse horizontal curves with intermediate tangents based on vehicle stability and (2) to evaluate the cost-effectiveness of the new proposed guidelines for minimum radius. Using vehicle simulation software (VDM RoAD), vehicle dynamics were recorded for the base-case of 2D simple horizontal curves to enable a comparison with reverse horizontal curves with intermediate tangents, superimposed with various vertical alignments.

The results showed that an increase in the minimum radius requirements of TAC and AASHTO design guides, ranging from 3% to 29%, is required to compensate for the effects of reverse curvature and vertical alignment. The guidelines for reverse horizontal curves with intermediate tangents developed in this research, besides achieving vehicle stability requirements, were found to be more cost-effective than the traditional North American guidelines.

ACKNOWLEDGEMENTS

Many thanks to my thesis supervisor Dr. Said Easa for his generosity, guidance, direction, support, and encouragement throughout the course of this research. I would also like to thank my co-supervisor Dr. Paul Van Geel for his on-going support and encouragement. Thanks to Essam Dabbour for all his help, input, and efforts, and thanks to Steven Lee for his support, and giving me a “foot in the door”.

Finally, I would like to thank my family, for if it was not for them, I would not have had the chance, or opportunity to achieve this work. I would like to thank my father Dr. Abd El Halim, my mother Beverly, my Grandparents, my two brothers Karim and Tarek, my sister Vanessa, and my fiancé Ilona...without all your love and support, this wouldn't have been possible.

DEDICATION

This work is dedicated to my loving parents, Abd El Halim and Beverly for their support and
love, and for never giving up in me.

Thank-you for making all my dreams come true.

TABLE OF CONTENTS

ABSTRACT	iii
ACKNOWLEDGEMENTS	iv
DEDICATION	v
LIST OF TABLES	xiii
LIST OF FIGURES	xv
CHAPTER 1: INTRODUCTION	1
1.1 Background.....	4
1.1.1 Human Factors.....	4
1.1.2 Highway Geometry.....	5
1.1.3 Design Consistency.....	7
1.1.4 Vehicle Stability.....	8
1.2 Problem Definition.....	9
1.3 Research Objectives.....	12
1.4 Thesis Organization.....	13
CHAPTER 2: LITERATURE REVIEW	16
2.1 Highway Alignment.....	16
2.2 Sight Distance.....	16
2.2.1 Required Sight Distance.....	16
2.2.2 Types of Sight Distance.....	17

2.2.2.1 Stopping Sight Distance.....	17
2.2.2.2 Passing Sight Distance.....	18
2.2.2.3 Decision Sight Distance.....	19
2.2.2.4 Preview Sight Distance.....	20
2.2.2.5 Intersection Sight Distance.....	20
2.2.3 Criteria for Measuring Sight Distance.....	21
2.2.3.1 Perception-Reaction Time.....	22
2.2.3.2 Driver's Eye Height.....	23
2.2.3.3 Vehicle Operating Speed.....	23
2.2.3.4 Object Height.....	24
2.2.3.5 Coefficient of Pavement Friction.....	25
2.2.3.6 Sight Obstructions.....	25
2.3 Horizontal Alignment.....	25
2.3.1 Horizontal Curves.....	26
2.3.2 Minimum Radius Requirements.....	28
2.3.2.1 Superelevation.....	31
2.3.2.2 Side Friction Factor.....	32
2.3.3 Tangents.....	33
2.3.4 Spiral Curves.....	35
2.3.5 Safety Considerations.....	36
2.4 Vertical Alignment.....	37
2.4.1 Crest Vertical Curves.....	40
2.4.2 Sag Vertical Curves.....	41
2.5 Design Consistency.....	42

2.5.1 Design Consistency Measures.....	43
2.5.1.1 Operating Speed.....	43
2.5.1.2 Vehicle Stability.....	45
2.5.2 TAC and Design Consistency.....	46
2.6 Vehicle Stability.....	46
2.6.1 Vehicle Dynamics and Ride Characteristics.....	46
2.6.1.1 Acceleration.....	47
2.6.1.2 Braking Performance.....	48
2.6.2 Handling – Cornering.....	50
2.6.3 Vehicle Rollover.....	51
2.6.4 Vehicle Stability Models.....	52
2.6.4.1 The Point Mass Model.....	53
2.6.4.2 The Bicycle Model.....	53
2.6.4.3 Two-Axle Model.....	54
2.6.5 Vehicle Stability Simulation Software.....	55
2.6.5.1 NADSdyna.....	56
2.6.5.2 Vehicle Dynamic Analysis Nonlinear Model (VDANL).....	56
2.6.5.3 VDM RoAD.....	57
2.7 Other Considerations.....	59
2.7.1 Cross Section.....	59
2.7.2 Three Dimensional (3D) Alignment.....	59
2.7.3 Interchange Ramps.....	61
2.7.4 The Interactive Highway Safety Design Model (IHSDM).....	62
2.7.5 Truck Transport and Safety.....	64

CHAPTER 3: VDM RoAD SOFTWARE.....	68
3.1 Capabilities and Limitations of VDM RoAD.....	68
3.1.1 Capabilities.....	68
3.1.2 Limitations.....	71
3.2 Reasons for Selecting VDM RoAD.....	71
3.3 Input Data.....	72
3.3.1 Vehicle Type.....	74
3.3.2 Roadway Geometry.....	76
3.3.3 Driver Behaviour.....	76
3.3.4 Cargo-Load Distribution.....	77
3.4 Data Processing and Output.....	78
 CHAPTER 4: EXPERIMENTAL DESIGN AND DATA PREPARATION.....	 80
4.1 Selection of Variables.....	80
4.1.1 Minimum Radius and Superelevation.....	81
4.1.2 Vertical Alignment.....	82
4.1.3 Ratio Between Flatter and Sharper Radii for Reverse Curves.....	82
4.1.4 Design Speed.....	83
4.1.5 Selection of Design Vehicles.....	83
4.1.6 Cargo-Load Distribution.....	83
4.1.7 Tangent Length.....	84
4.2 Lateral Acceleration and Side Friction Variation.....	84
4.3 VDM RoAD and Human Factors.....	85

CHAPTER 5: DEVELOPMENT OF NEW GUIDELINES FOR REVERSE CURVES WITH INTERMEDIATE TANGENTS	87
5.1 Simulation Procedure.....	88
5.2 Development of New Guidelines.....	91
5.3 Side Friction Variation.....	94
5.4 Effect of Tangent Length on Reverse Curvature.....	95
5.5 Effect of Vertical Alignment on Reverse Curvature.....	98
5.6 Model Development.....	100
5.6.1 Development of Flat Reverse Curve Model with Intermediate Tangent...	100
5.6.2 Development of 3D Reverse Curve Model with Intermediate Tangent...	103
5.7 Sensitivity Analysis.....	107
5.7.1 Effect of Design Speed (V_D).....	107
5.7.2 Effect of Tangent Length (T).....	109
5.7.3 Effect of Superelevation (e_{max}).....	112
5.7.4 Effect of Design Vehicle (<i>WB-15, WB-20, NHTSA 3a/2a</i>).....	113
5.7.5 Effect of Ratio Between Curve Radii R_1 and R_2	114
5.7.6 Effect of First and Second Grade (G_1 and G_2).....	115
5.8 Guidelines for Spiral Curves.....	116
CHAPTER 6: COST EFFECTIVENESS ANALYSIS.....	120
6.1 Geometry of Existing and New Horizontal Curves.....	120
6.2 Analysis Methodology.....	124
6.2.1 Construction Cost and Savings.....	124
6.2.2 Collision Savings.....	125

6.2.3 Vehicle Running Cost Savings.....	126
6.2.4 Cost Associated with Speed Changes.....	127
6.2.5 Net Present Value of the Annual Savings.....	128
6.2.6 Other Considerations.....	129
6.3 Analysis of Simple Horizontal Curves.....	129
6.3.1 Case S1: Newly Constructed Simple Horizontal Curve.....	129
6.3.2 Case S2: Flattening of an Existing Simple Horizontal Curve.....	130
6.4 Analysis of Reverse Horizontal Curves ($T = 0$).....	135
6.4.1 Case R1: Newly Constructed Reverse Horizontal Curve.....	136
6.4.2 Case R2: Flattening of an Existing Reverse Horizontal Curve.....	138
6.5 Analysis of Reverse Horizontal Curves ($T = 100$ m).....	141
6.5.1 Case RT1: Newly Constructed Reverse Curve ($T = 100$ m).....	141
6.5.2 Case RT2: Flattening of an Existing Reverse Curve ($T = 100$ m).....	142
6.6 Sensitivity Analysis.....	145
6.6.1 Average Annual Daily Traffic.....	145
6.6.2 Unit Construction Cost.....	148
 CHAPTER 7: SUMMARY AND CONCLUSIONS	
7.1 Summary.....	150
7.2 Conclusions.....	152

7.3 Recommendations.....	153
REFERENCES.....	157
APPENDIX A	164
APPENDIX B.....	185

LIST OF TABLES

Table 2.1: Minimum Spiral Parameters for $e_{max} = 0.06$ (TAC, 1999).....	36
Table 2.2: Design Rates of Vertical Curvature K for Crest and Sag Vertical Curves (AASHTO, 2001).....	40
Table 2.3: Consistency Evaluation Criteria.....	44
Table 2.4: Consistency Measure for Vehicle Stability.....	45
Table 2.5: Rollover Threshold Values for Different Vehicles (Gillespie, 1992).....	52
Table 2.6: Rollover Threshold for Various Weight Ranges of Nontanker Trucks.....	67
Table 5.1: Design Speeds and Minimum Design Radii used in Analysis.....	89
Table 5.2: Required Increase in curve radius R_l for flat reverse curves for different values of ratio, and intermediate tangent lengths (NHSTA 3a/2a, $e_{max} = 0.06$).....	92
Table 5.3: Required Increase in curve radius R_l for 3D reverse curves for different values of ratio, and intermediate tangent lengths with (NHSTA 3a/2a, $e_{max} = 0.06$).....	93
Table 5.4: Minimum Lengths of Spiral Transition Curve For 3D Reverse Curves with Intermediate Tangents (NHTSA 3a/2a, Ratio = 1.5, $e_{max} = 0.06$)..	117
Table 5.5: Minimum Lengths of Spiral Transition Curve For Flat Reverse Curves with Intermediate Tangents (NHTSA 3a/2a, Ratio = 1.5, $e_{max} = 0.06$)..	117
Table 6.1: Annual Savings and Benefits of Implementing New Simple Horizontal Curves ($e_{max} = 0.04$ and AADT = 7,000).....	134
Table 6.2: NPV of Costs and Benefits of Implementing Larger Radius Simple Horizontal Curves in New and Existing Alignments over 20-year Project Life ($e_{max} = 0.04$ and AADT = 7,000).....	134
Table 6.3: Annual Savings and Benefits of Implementing New Reverse Horizontal Curves ($T = 0$ m, $e_{max} = 0.06$, and AADT = 7,000)...	140
Table 6.4: Costs and Benefits of Implementing Larger Radius Reverse Horizontal Curves in New and Existing Alignments over 20-year Project Life ($T = 0$ m, $e_{max} = 0.06$, and AADT = 7,000).....	140

Table 6.5: Annual Savings and Benefits of Implementing New Reverse Horizontal Curves ($T = 100$ m, $e_{max} = 0.06$ and AADT = 7,000).....	144
Table 6.6: NPV of Costs and Benefits of Implementing Larger Radius Reverse Horizontal Curves in New and Existing Alignments over 20-year Project Life ($T = 100$ m, $e_{max} = 0.06$, and AADT = 7,000)...	144

LIST OF FIGURES

Figure 1.1: Number of Traffic Related Fatalities in the US from 1995 to 2001..... (NTS, 2002)	1
Figure 1.2: Occupant Fatalities for Passenger Cars and Trucks in the United States..... (NTS, 2002)	2
Figure 1.3: Research Overview and Outline.....	15
Figure 2.1: Horizontal Circular Curves.....	27
Figure 2.2: Dynamics of Vehicle on Circular Curve (TAC, 1999).....	30
Figure 2.3: Different types of Simple Vertical Curves.....	38
Figure 2.4: Simple Parabolic Vertical Curve.....	40
Figure 2.5: A Late model Freightliner Transport Truck.....	65
Figure 2.6: Truck Rollover on Tangent Section of 2-Lane Rural Highway.....	66
Figure 3.1: Analytical model for truck stability in VDM RoAD.....	70
Figure 3.2: Screenshot of <i>Runs</i> Screen from VDM RoAD.....	73
Figure 3.3: Vehicle Screen in VDM RoAD for WB-12 Combination Truck.....	75
Figure 3.4: Wire Frame animation in VDM RoAD.....	78
Figure 3.5: Sample of Output plots for single run in VDM RoAD.....	79
Figure 4.1: Lateral acceleration calculated by VDM RoAD on 3-D alignment.....	85
Figure 5.1: Evaluation procedure for vehicle stability.....	90
Figure 5.2: Plot of Actual and Assumed Lateral Acceleration for WB15 truck on a reverse curve with intermediate tangent.....	94
Figure 5.3: Increase in Required Minimum Radius of Flat Reverse Curve for different tangent lengths (NHTSA 3a/2a, $e_{max} = 0.04$, and Ratio = 1.0).....	96

Figure 5.4: Comparison of Effect of Reverse Curvature and Vertical Alignment on the Required Increase in Minimum Radius for Design Vehicle WB-20, $e_{max} = 0.06$ (worst case vertical alignment).....	99
Figure 5.5: Effect of Design Speed on Increase in Required Minimum Radius for design vehicle NHTSA 3a/2a, $e_{max} = 0.04$, and Ratio = 1.0 for level terrain.....	108
Figure 5.6: Effect of Tangent length on the Required Radius for WB20, $e_{max} = 0.04$, $V_D = 80\text{km/h}$, and for different values of ratio for level terrain.....	110
Figure 5.7: Effect of Tangent Length on the Required Radius for WB15, $e_{max} = 0.04$, $V_D = 90\text{ km/h}$, and for different values of ratio for level terrain.....	110
Figure 5.8: Minimum Radius of Flat Reverse Horizontal Curves for NHTSA 3a/2a, $e_{max} = 0.04$, and Ratio = 1.0.....	111
Figure 5.9: Minimum Radius of Flat Reverse Horizontal Curves for NHTSA 3a/2a, $e_{max} = 0.04$, and Ratio = 1.5.....	111
Figure 5.10: Minimum Radius of Flat Reverse Horizontal Curves for NHTSA 3a/2a, $e_{max} = 0.04$, and Ratio = 2.0.....	112
Figure 5.11: Effect of Superelevation on Increase in Required Minimum Radius for Design vehicle NHTSA 3A/2A, $V_D = 60\text{ km/h}$, Ratio = 1.0 for level terrain...	113
Figure 5.12: Effect of Design Vehicle on Increase of Required Minimum Radius for design vehicles WB15, WB20 and NHTSA 3a/2a, for $V_D = 40\text{ km/h}$, $e_{max} = 0.04$, Ratio = 1.0, and for flat terrain.....	114
Figure 5.13: Effect of ratio between Flatter and Sharper arcs on Increase of Required Minimum Radius for design vehicles WB20 for $V_D = 40\text{ km/h}$, $e_{max} = 0.04$ for level terrain.....	115
Figure 5.14: Minimum Spiral Lengths Based on TAC Spiral Parameter for $e_{max} = 0.06$, design vehicle NHSTA 3a/2a, Ratio = 1.5 with vertical alignment.....	118
Figure 5.15: Minimum Spiral Lengths Based on TAC Spiral Parameter for $e_{max} = 0.06$, design vehicle NHSTA 3a/2a, Ratio = 1.5 with vertical alignment.....	118
Figure 6.1: Geometry and Effect of Curve Flattening.....	121
Figure 6.2: Geometry of Reverse Curve.....	135

Figure 6.3: Reverse Curve with Tangent Length between two arcs.....	142
Figure 6.4: NPW of Simple Horizontal Curves with AADT = 1000.....	146
Figure 6.5: Variation of the Net Present Worth of Simple Horizontal Curves with Design Speed.....	147
Figure 6.6: Effect of AADT on the Net Present Worth Benefits for Simple Horizontal Curves for various design speeds.....	147
Figure 6.7: Construction Cost for Simple Horizontal Curve ($e_{\max} = 0.04$).....	148
Figure 6.8: Effect of 10% increase and 10% decrease in Cost on the Construction Savings of constructing a larger radius simple Horizontal curve ($e_{\max} = 0.04$).....	149

CHAPTER 1

INTRODUCTION

Highway safety is becoming a more important issue in today's societies. It is a worldwide problem as there are over a half billion motor vehicles in use around the world. As the number of motor vehicles operating on roads has continued to increase, the exposure of the population to traffic collisions has increased as well. Each year, 500,000 people are killed and 15 million people are injured in motor vehicle collisions (Lamm et al., 1999). In the United States, every year approximately 40,000 people are killed in traffic related collisions (Figures 1.1 and 1.2). Highway safety is a complex problem that involves driver behaviour vehicle characteristics, roadway features, and driving conditions.

Figure 1.1: Number of Traffic Related Fatalities in the US from 1995 to 2001 (NTS, 2002)

In the United States, motor vehicle collisions are the leading cause of death for people between the ages of 1 and 34 (Lamm et al., 1999). More people in the U.S. have been killed in traffic related collisions over the past 100 years, than have died fighting in all the major wars that the U.S. has participated in (WW1, WW2, Korea, Vietnam, and Gulf War). In 1980, more than 51,000 people were killed in traffic collisions in the United States. In 1995, 41,798 people were killed in traffic collisions representing a percent change of -18% between 1980 and 1995. In Canada, 3601 road users were killed in 1993. This figure is much lower than the total amount of traffic deaths during 1982-1992 when the number was 4,100 to 4,350 people.

Figure 1.2: Occupant Fatalities for Passenger Cars and Trucks in The United States (NTS, 2002)

The high number of vehicles operating on highways around the world and the high number of vehicle miles travelled has posed a great challenge to highway engineers and designers. It is the role of highway engineers, designers, and planners to ensure that the highway system is designed and operated in such a way as to make certain that the number of collisions is at a minimum. In other words, it is their responsibility to the public and society to design the safest highway system within economic and environmental constraints.

A strong link exists between traffic safety and geometric design. Geometric design of highways refers to the design of the visible dimensions of such features as horizontal and vertical alignments, cross sections, intersections, and bicycle and pedestrian facilities (Easa, 2003). The main objectives of geometric design are to produce a highway with safe, efficient, and economic traffic operations, while still maintaining aesthetic and environmental quality. Several design controls influence today's highway geometric design process. Some of these controls are type of design vehicle, driver performance, design speed, classification of the highway, topography, climatic conditions, traffic volumes, soil information, and projected land use.

Organizations such as the American Association of State Highway and Transportation Officials (AASHTO) and the Transportation Association of Canada (TAC) have provided engineers and designers with policies on highway geometric design. These policies are compiled into manuals and represent a set of standards or guidelines for highway design. Over the years, AASHTO has produced a number of policies on geometric design (The Red Book, Blue Book, and Green Book). TAC has also provided Canadian highway engineers with policies on geometric design. The latest edition of the AASHTO Policy on highway

geometric design is “A Policy on Geometric Design of Highways and Streets” (AASHTO, 2001). The latest edition of the TAC Policy on highway geometric design is the “Geometric Design Guide for Canadian Roads” (TAC, 1999). The role of the guidelines is to provide information and background to assist the designer in selecting appropriate combinations of features and dimensions for a given design.

In this chapter, Section 1.1 provides background information on some of the important aspects of geometric design elements. In Section 1.2, the research problem is addressed and defined. In Section 1.3, the research objectives of the work presented in this thesis are outlined. The organizational structure of the remaining chapters within this work is highlighted in Section 1.4.

1.1 BACKGROUND

The design of highways involves reaching an optimal compromise between safety, efficiency, and economy (Kanellaidis, 1996). Providing good coordination among the users, the vehicle, and the roadway helps to improve and better understand the relationships and interrelationships of traffic safety. In this section, some of the major components of highway safety are discussed.

1.1.1 Human Factors

The success of a highway facility is reflected in the extent to which it satisfies the needs of its users. As the main controlling element, the driver plays the most important role in determining the efficiency and level of safety of a highway network. The main objective

should be to minimize the gap between what users need and expect, and what they are presented with. More than 90% of all traffic collisions are attributed to human error or improper human behaviour (Lamm et al., 1999).

A driver's ability to operate a vehicle safely depends on many physical and mental characteristics. Some of these characteristics are senses such as vision and hearing, physical conditions such as health and age, and mental conditions such as fatigue and stress. There are numerous variations in the capabilities of drivers, as well as variations directly within the population. A driver's level of skill is affected by his/her age, health, experience, and a number of other factors. Since no two humans are exactly alike, and thus no two drivers exhibit the exact same level of skill or performance, a highway design must be able to complement the needs of a large proportion of the population in order to provide a high level of safety.

1.1.2 Highway Geometry

The alignment of a highway has a profound effect on the environment, economy, society, and highway users. A highway alignment is comprised of a variety of elements all interconnected to create a facility that serves all users in a safe and efficient manner, consistent with the facility's intended function (AASHTO, 2001). There are several principal elements of geometric design such as sight distance, horizontal and vertical alignments, cross section, intersections, interchanges, drainage, and traffic control devices. Each element should complement other elements to provide a consistent, safe and efficient design. In this way, the road characteristics are well balanced for the motorist along the course of the road section.

The vertical alignment of a highway is the profile of the highway that shows its elevation along the length of its path. Vertical alignment is made up of tangents and parabolic curves. A tangent can be an upgrade, downgrade, or level (flat) in nature. Parabolic curves are either crest or sag curves, and their function is to connect tangents. The horizontal alignment is the plan view or "*birds eye view*" of the highway along the length of its path. Horizontal alignment is made up of tangents and circular curves. There are four basic types of horizontal curves: simple curves, compound curves, reverse curves, and spiral curves (Easa, 2003). The cross section of a highway is another major geometric design element. Typical cross section elements include the traveled way, shoulders, curbs, medians, sideslopes and backslopes, clear zones, pedestrian facilities, and bicycle facilities. The horizontal and vertical alignments are permanent design elements that require extensive research, evaluation, and design considerations. It is extremely difficult and expensive to correct any deficiencies with a design after the highway has been constructed.

The geometric form of a highway is three-dimensional (3D) in nature, consisting of horizontal and vertical alignments. Traditionally, highways have been designed one element at a time, and are viewed as two-dimensional (2D) alignments. The main disadvantage to this type of design is that it may lead to poor design consistency along the length of a highway. Until recently, 3D analysis was generally considered infeasible because of its mathematical complexity and computer intensive requirements (Easa et al., 2002). Hassan et al. (1998) have stressed the importance and significance of considering the 3D nature of a highway alignment. A 3D-operating speed model for horizontal curves was developed by Gibreel et al. (2001) that showed that 3D models more accurately predict operating speeds than traditional 2D models. It is important that highway horizontal and vertical alignments

are designed in a 3D manner. In addition, the coordination of vertical and horizontal alignments can help designers ensure a safe, comfortable, efficient, and aesthetically pleasing design (Smith and Lamm, 1994)

1.1.3 Design Consistency

One of the most important factors in geometric design and highway safety is how well a designer reduces or eliminates uncertainty for the driver. The more consistent road designs are over a wide geographic area, the more effective the designer's contribution will be to reducing collision occurrence (TAC, 1999). It is the responsibility of the designer to achieve "design consistency" in the design of different classes of roads and changes in topography and landscape. Design consistency is the conformance of a highway's geometry with driver expectancy, and its importance to road safety is justified through an understanding of the driver-vehicle-roadway interaction (Ng and Sayed, 2003).

Highway designers must always take into consideration the relevant standards involved in selecting geometric elements. It is important to note that motorists may not expect major changes in design speed, cross section, or alignment standards between adjacent sections along a rural highway (Messer, 1980). Any abrupt geometric changes may be so inconsistent with the driver's basic expectations that delayed driving responses, and incorrect decisions may occur and result in unsafe driving. The primary inputs to the driving task are roadway geometry, traffic conditions, and the roadside environment. These inputs affect the driving task and determine the workload requirement on the driver. If an inconsistency exists that violates a driver's expectation, the driver may adopt an inappropriate speed or manoeuvre

that may lead to a collision. When design consistency is carried out through a design, abrupt changes in geometric features are eliminated, and the level of safety is improved.

Most of the research and work in design consistency has focused on identifying quantitative measures for design consistency evaluation and on developing models for their estimation (Ng and Sayed, 2003). These measures are classified into four categories: operating speed, driver workload, alignment indices, and vehicle stability. These four categories are widely used to evaluate the design consistency of a highway. Anderson et al. (1999) found several measures of design consistency that have statistically significant relationships to collision frequency. TAC has taken a leading role in North America by incorporating design consistency in its geometric design policy (Easa, 2003).

1.1.4 Vehicle Stability

Vehicle stability is an important measure of design consistency and safety. When insufficient side friction is provided at a horizontal curve, vehicles may skid out, roll over, or become involved in head-on collisions (Ng and Sayed, 2003). Such a design violates the ability of the drivers to guide and control their vehicle in a safe manner. Locations that do not provide sufficient side friction can be considered as geometric design inconsistencies. The difference between the assumed side friction and the side friction demanded (Δf_R) is a measure of vehicle stability and has been used as an evaluation criterion for design consistency (Lamm et al., 1999).

As a vehicle traverses a horizontal curve, the driver and vehicle experience a centripetal acceleration that acts towards the centre of the horizontal curve. The centrifugal force acts

outward from the curve centre. For the vehicle to maintain stability as it navigates the horizontal curve, the centrifugal force must be resisted. This is accomplished by a component of the vehicle's weight that acts perpendicular to the roadway superelevation, and by the side friction developed between the vehicle's tires and the pavement surface. The side friction demanded by a vehicle is directly related to the amount of lateral acceleration required to travel at a speed V on a curve of radius R and superelevation rate e (Bonneson, 2001). The frictional forces are highly dependent on the condition of the tire, road surface conditions, weather and climatic conditions, and vehicle kinematics. The component of the vehicles weight that acts parallel to the road surface depends on the superelevation (or cross slope) of the roadway.

This simplified approach to modelling a vehicle as it traverses a horizontal curve is based on the laws of mechanics and kinematics, and is referred to as the point-mass model. This model was adopted by the North American design guides due to its simplicity. The point-mass model will be further discussed in the literature review chapter. For 3D alignments, traditional design guides such as AASHTO (2001) and TAC (1999) calculate the minimum radius of a horizontal curve assuming a side friction on a horizontal plane using the point-mass model, thus ignoring the effects of vertical alignments and complex curvature. This approach simplifies cornering dynamics by reducing the vehicle into a point mass travelling on a 2D horizontal alignment.

1.2 PROBLEM DEFINITION

Studies have demonstrated an alarming trend that a large proportion of the total traffic related fatalities occur on curved sections, and more importantly, on horizontal curves of two-lane

rural highways (Lamm et al., 1999). Also, statistics have revealed that truck rollover rates on complex horizontal curves such as highway interchanges and on and off ramps are very high. This trend clearly indicates that there is a limitation in the design of horizontal curves. The minimum horizontal curve radius in the current design guides is related to the lateral acceleration and side friction through the point-mass formula.

Current geometric design guides address each geometric design element separately and in two-dimensions rather than collectively and in three-dimensions. This design practice does not optimize the requirements for safe, consistent, and aesthetically pleasing highway design. For example, in the current design practices, in the design of a reverse horizontal curve, the design guides separate the reverse curve into two separate simple horizontal curves and does not account for the vehicle dynamics experienced as a result of the complex curvature. In addition, for a reverse horizontal curve with an intermediate tangent, current design guides assume that the tangent length has no effect on the required minimum radius. A limitation arises when a complex curve is superimposed with a vertical alignment. In this case, the current design practice deals with the horizontal and vertical alignments separately, and ignores the 3D nature of the alignment.

The preceding limitations of the current design guides are caused by the point mass model that has been adopted in establishing required minimum radius requirements. The main drawback of this model is that it does not account for the 3D nature of highway alignments. The model simplifies a vehicle to a single point of mass, and thus does not consider the distribution of frictional forces between the inner and outer or front and rear tires (Easa and Dabbour, 2003). Another drawback is that the model is theoretically as applicable to trucks

as it is to passenger cars (Harwood et al., 1994). This assumption is not accurate as cars and trucks experience completely different magnitudes of forces due to differences in size, ride height, length, weight, number of tires, types of suspensions, and many other factors. Other limitations to the point mass model exist and will be discussed in the literature review chapter.

The inadequacy of the current North American design guides as related to vehicle stability is summarized as follows:

1. Current design guidelines separate complex curves such as reverse or compound curves into two simple horizontal curves, and thus do not provide adequate design requirements for those complex horizontal curves.
2. The effect of an intermediate tangent on the required minimum radius of reverse horizontal curves has not been addressed in the current design guides.
3. The current design guidelines deal with any highway alignment as two separate horizontal and vertical alignments, with little or no requirements for 3D combined alignments.
4. Due to the lack of guidelines for complex horizontal curves in the current design guides, there have been increased incidents of truck rollover on highway curves and interchanges.

Several of the preceding aspects of vehicle stability have been addressed by researchers, including simple horizontal curves (Easa and Dabbour, 2003) and (Furtado et al., 2002),

compound horizontal curves (Easa and Dabbour, 2004), and reverse horizontal curves (Dabbour et al., 2004). This thesis addresses vehicle stability of reverse horizontal curves with intermediate tangents and their cost-effectiveness.

1.3 RESEARCH OBJECTIVES

The main objectives of this research thesis are to:

1. To conduct a literature review on the current design guidelines and to review previous research and work.
2. To use VDM RoAD simulation software to examine truck stability on complex geometric alignments such as:
 - i. Simple horizontal curves that are used as a base case for comparison.
 - ii. Reverse horizontal curves with no vertical alignment.
 - iii. Reverse horizontal curves combined with vertical alignment.
3. To develop minimum radius requirements for reverse horizontal curves with intermediate tangents.
4. To calibrate mathematical models for the required minimum radius for flat and 3D reverse curves with an intermediate tangent.
5. To investigate the effects of an intermediate tangent on the required length of the spiral curve by comparing it with the TAC (1999) recommended lengths.

6. To conduct an analysis comparing the costs and benefits of implementing the new and improved curve design approach with traditional design approaches (TAC, 1999).

1.4 THESIS ORGANIZATION

An organized research outline representing the various stages and tasks of the thesis is presented in Figure 1.3. The chapters of this thesis are organized as follows:

- Chapter 2 contains a detailed literature review on some of the main components of geometric design - the horizontal and vertical alignments, cross section, superelevation, interchange ramps, and 3D alignments. Also covered in this section are a review of design consistency and a brief overview of different vehicle stability models and the Interactive Highway Safety Design Model (IHSDM Project).
- Chapter 3 discusses the simulation software used for the experimental portion of this research. It provides an overview of the simulation processes in VDM RoAD including data input, simulation processing, and data output. The capabilities and limitations of VDM RoAD are discussed in this section.
- Chapter 4 outlines the different variables used in the simulation process for the research. Some of the variables discussed are the design vehicles, test alignments, and all other considerations in the testing selection criteria.
- Chapter 5 provides details on the simulation procedures for reverse horizontal curves with and without intermediate tangents for flat (level) and 3D-alignments. New

guidelines for the minimum required radius of reverse horizontal curves with intermediate tangents are developed in this chapter. Using the experimental output and test alignments, two mathematical models (Flat and 3D) were developed. A sensitivity analysis for each variable involved in the simulation is conducted

- Chapter 6 investigates the economic effectiveness of implementing the new reverse horizontal curve design approach that considers vehicle stability and vertical alignment. The costs, savings, and benefits of constructing larger radii simple and reverse horizontal curves are evaluated.
- Chapter 7 highlights the main findings and results of this research. Conclusions and recommendations are provided. Also outlined are areas of further research and study.

For a more complete overview of the structure of this thesis, a flowchart representing the organization and structure of the research and tasks is presented in Figure 1.3.

Figure 1.3: Research Overview and Outline

CHAPTER 2

LITERATURE REVIEW

2.1 HIGHWAY ALIGNMENT

The alignment of a highway or street produces a significant impact on the environment, community, and the highway users. The alignment of a highway design is composed of several major components, the most important ones being: sight distance, the horizontal and vertical alignments, cross-section/alignment, and combined 3D alignments. The topography of the land in which the highway is situated has a profound influence on the alignment of highways, roads and streets (AASHTO, 2001).

2.2 SIGHT DISTANCE

One of the most important aspects in highway and traffic safety, and more specifically in geometric design is sight distance. Sight distance is the length of the roadway ahead that is visible to a driver. A driver's ability to see ahead is of the utmost importance in the safe and efficient operation of a vehicle on a highway (AASHTO, 2001). It is a fundamental design element in the safe and efficient operation of a highway.

2.2.1 Required Sight Distance

Sight distance is highly variable and can be a very challenging quantity to measure since the power and characteristics of vehicles vary greatly and the ability of a driver to operate, perceive, and react is different from person to person. Thus, it is very important that

highway designers provide sight distance of adequate length that would allow drivers to control their vehicles and avoid crashing into an unexpected object on the roadway. For some roads and highways such as two-lane rural highways, adequate sight distance should be provided to enable drivers to occupy the opposing lane when initiating a passing manoeuvre without risk of collision. On two-lane rural highways, sufficient sight distance should be provided frequently over the length of the highway.

2.2.2 Types of Sight Distance

In highway design, several types of sight distances must be considered in the design process. The following types of sight distances should be considered in the design: stopping sight distance (SSD), passing sight distance (PSD), decisions sight distance (DSD), preview sight distance (PVSD) and intersection sight distance. The uses of the different types of sight distances are dependent on the type of highway (multi-lane, or 2-lane) and the complexity of the design.

2.2.2.1 Stopping Sight Distance

Stopping sight distance (SSD) is the sum of two distances. The first distance is the distance traveled during the perception reaction time, when the driver first sights an object requiring a stop to the instant the brakes are applied. The second distance is defined as the distance that it takes to stop a vehicle once the brakes have been applied. It is from the sum of the distance traveled during the perception reaction time and the braking distance that makes up the stopping sight distance. A time of 2.5 seconds is typically used as the perception reaction

time, which represents the 90th percentile of drivers under various situations (Fambro et al., 2000).

The deceleration rate recommended by AASHTO (2001) is 3.4 m/s^2 . This value is recommended since approximately 90 percent of all drivers decelerate at rates greater than 3.4 m/s^2 . This rate of deceleration allows a driver to maintain steering control of their vehicle in their lane under wet conditions. The friction available on most wet pavement surfaces and the capabilities of most vehicle braking systems can provide a coefficient of friction that exceeds this deceleration rate. Fambro et al. (2000) developed a new stopping sight distance model. Their justification for a new SSD model was that past models were based on common sense, engineering judgment, and the laws of physics. One of the major concerns was the parameters in the previous models were not representative of the true driving environment (Fambro et al., 2000).

2.2.2.2 Passing Sight Distance

On most two-lane rural highways, faster moving vehicles often overtake slower moving vehicles. This passing manoeuvre must be accomplished on lanes in which on-coming vehicles are operating. It is a major requirement of safety that sufficient sight distance be provided in order to complete the passing manoeuvre without cutting off the passed vehicle or colliding with the on-coming vehicle. In some instances, a driver may abort a pass, and return back to their original position in the lane.

When designing a road or highway in which passing will be performed, passing sight distance should be determined on the basis of the length required to complete normal passing

manoeuvres in which the passing driver can determine that there are no potentially conflicting vehicles ahead before beginning the manoeuvre (AASHTO, 2001). The passing sight distance is determined for a one-vehicle pass. In instances where multiple passes may take place, longer sight distances are to be provided in the design. When determining passing sight distance for design of two lane rural highways, several assumptions are made concerning driver behaviour during the passing manoeuvre (AASHTO, 2001):

- 1) The overtaken vehicle travels at a constant speed.
- 2) The passing vehicle reduces speed and trails the overtaken vehicle as it enters a passing section.
- 3) When the passing section is reached, the passing driver needs a short period to perceive the clear passing section and to react to start their manoeuvre.
- 4) Passing is accomplished under what may be termed a delayed start and a hurried return in the face of opposing traffic. The overtaking vehicles speed increases to 15 km/h more than the overtaken vehicle.
- 5) When the passing vehicle returns to its lane, there is a suitable clearance length between it and an on-coming vehicle in the other lane.

2.2.2.3 Decision Sight Distance

Decision sight distance is the distance needed for a driver to detect an unexpected or otherwise difficult-to-perceive information source or condition in a roadway environment that may be visually cluttered, recognize the condition or its potential threat, select an appropriate speed and path, and initiate and complete the manoeuvre safely and efficiently (AASHTO, 2001). The major benefit of decision sight distance is that it offers drivers

“room” for additional margin for error, and allows them sufficient length to manoeuvre their vehicles, and make adjustments. As a result, the values for decision sight distance are substantially greater than the values for stopping sight distance. Drivers require decision sight distances whenever there is a chance for error in either information reception, decision-making, or control actions (AASHTO, 2001). Easa (2003) outlines some complex locations where provisions of DSD are desirable. These locations include complex interchanges and intersections, toll plazas, lane drops, and areas where sources of information (such as signs, signals and traffic control) are present.

2.2.2.4 Preview Sight Distance

Preview sight distance is the distance required by a driver to perceive a horizontal curve and properly react to it. AASHTO implicitly recognized PVSD by recommending that sharp horizontal curvature should not be introduced at or near the top of a pronounced crest vertical curve or near the low point of a pronounced sag vertical curve. For 3D-alignments, the preview sight distance is the sum of two components: the tangent component, and the curve component (Easa, 2003).

2.2.2.5 Intersection Sight Distance

Intersection sight distance is the distance provided at an intersection to allow approaching vehicles to see any potentially conflicting vehicles in sufficient time to slow or stop safely and to allow stopped vehicles to enter or cross the intersection safely. The approaching vehicle may be approaching from an uncontrolled or yield-controlled intersection, and the stopped vehicle may be at a stop or signal-controlled intersection (Easa, 2000).

The intersection sight triangle must be clear of obstructions that might block the driver's view of potentially conflicting vehicles. The dimensions of the sight triangle depend on the design speed of the road and the type of traffic control at the intersection. Two types of sight triangles must be considered in the design of the intersection – approach and departure sight triangles. For the approaching sight triangle, the length of the legs should be sufficient to allow the driver to see any potential conflicting vehicle, and to allow for sufficient time to slow or stop before reaching the intersection. For the departure sight triangles, the clear sight triangle provides a sight distance sufficient for a stopped driver on the minor-road approach to leave the intersection and cross the major road. AASHTO (2001) recommends a driver's eye height of 1.08 m and an object height of 1.08 m above the roadway to construct the intersection sight triangles. If an object's height within the sight triangle is larger than the respective height of the sight triangle it can be considered an obstruction.

2.2.3 Criteria for Measuring Sight Distance

Sight distance is a very important design requirement that must be addressed and provided in the design of safe roads and highways. As previously mentioned, sight distance is the distance along a roadway throughout which an object of specified height is continuously visible to the driver (AASHTO, 2001). Several variables are required for calculating sight distance. The variables required based on the TAC and AASHTO guidelines are: perception-reaction time, the height of the driver's eye above the road's surface, the vehicles operating speed, the specified object height above the road's surface, the coefficient of pavement friction and the height and lateral position of sight obstructions within the driver's line of

sight. In some situations (night time), the height of a vehicle headlight will have an effect on the sight distance.

2.2.3.1 Perception-Reaction Time

When calculating sight distance, the perception and reaction time is used to describe the period between the occurrence or appearance of a “signal” and the driver’s physical reaction to it (TAC, 1999). Complex and unexpected decisions require considerably longer time, than a basic, anticipated decision. Perception and reaction is considered to comprise four elements (TAC, 1999):

1. Perception: The use of vision capabilities to see a visual signal.
2. Identification: The driver identifies the signal and understands the stimulus.
3. Emotion: The driver decides the proper action to take in response to the stimulus.
4. Volition: The driver decides what action to take in response to stimulus.

Perception reaction time is difficult to quantify as humans exhibit so much variation in their abilities. When conducting field tests, results are usually suspect as a driver is in a heightened state of anticipation under test conditions, and may exhibit shorter perception and reaction times. Typically, 2.5 seconds is used as the perception reaction time, which is representative of the 90th percentile of drivers and situations. The perception reaction time for un-alerted drivers to complex or inconspicuous stimuli is 3.0 to 4.5 seconds (TAC, 1999).

2.2.3.2 Driver's Eye Height

Driver's eye height is a criterion that is highly dependent on the type and design of a motor vehicle. In a study by the *National Committee on Uniform Traffic Control*, it was found that 99% of the total vehicle population has a driver eye height of at least 1.05 m (TAC, 1999). For sight distance calculations, driver's eye height is dependent on vehicle type. For a passenger vehicle, AASHTO (2001) gives a driver's eye height of 1.08 m, while TAC recommends 1.05 m. The values of driver's eye height for passenger vehicles have decreased over the decades. This can be attributed to the more "lowered and sleeker" designs of passenger vehicles. One issue that has arisen with the decrease in driver eye height is the relatively small increases in the lengths of vertical curves that would result for changes that do occur. Designs that were once deemed safe and up to standard, may in the future be considered unsafe due to a decrease in the driver's eye height of passenger vehicles. This can lead to liability and legal issues for highway designers and planners, and governmental organizations. For other vehicles such as buses and single unit vehicles as defined by TAC, driver's eye height is recommended to be 1.8 m above the roadway. For larger trucks and truck trailer combinations, the height of an eye can vary from 1.9 m to 2.4 m.

2.2.3.3 Vehicle Operating Speed

The operating speed of a vehicle is an important factor used when calculating sight distance. For a given individual, the operating speed is generally lower than the desired speed, as operating conditions are very seldom ideal. The operating speed of all vehicles at a particular location is reported as 85th to 90th percentile operating speed. A typical driver is able to recognize a suitable operating speed for a given roadway based on knowledge of the system,

posted speed limits, topography, traffic volumes and densities, and past experiences (TAC, 1999). Generally, for increasing operating speeds, a driver and vehicle will require a longer length of road to come to a full stop.

2.2.3.4 Object Height

When calculating sight distance, a highway engineer or designer must determine what type of object is likely to be encountered on the roadway. It is this object that the driver is trying to avoid by stopping or manoeuvring. Over the past decades, several values of object heights ranging from zero up to 1.67 m have been suggested and used for different applications (TAC, 1999). For stopping sight distance, TAC recommends a height of 0.38 m. This height represents a vehicle tail or brake light. AASHTO defines the height of object to be 600 mm above the road surface. This value was selected based on an arbitrary rationalization the size of object that might potentially be encountered in the road and of a driver's ability to perceive and react to such situations. It is considered that a 600 mm object could potentially cause risk to drivers and can be recognized by a driver in time to stop before colliding with it. Using object heights lower than 600 mm would result in an increase in the length of vertical curves, and an increase in construction costs. For passing sight distance calculations, an object height of 1080 mm and 1300 mm is recommended by AASHTO and TAC, respectively. AASHTO bases the object height of 1080 mm as the height of an on coming vehicle. This height represents the 15th percentile of vehicle heights in the current passenger car population.

2.2.3.5 Coefficient of Pavement Friction

The friction of pavement is represented by a coefficient of friction (f_L), and is considered to be one of the most sensitive parameters in calculating stopping sight distance (TAC, 1999). The coefficient of friction of pavement is highly variable and can vary depending on the speed of a vehicle prior to braking, the type of tire tread, amount of tire wear, condition of pavement, presence of water, mud, snow, or ice, etc. When calculating stopping sight distance, the value of f_L is assumed to vary only with design speed, and wet pavement conditions are assumed.

2.2.3.6 Sight Obstructions

On a tangent or relatively flat roadway, the obstruction that limits the driver's sight distance is the surface of the road at some point on a crest vertical curve. On horizontal curves, the obstruction that limits the driver's sight distance may be the road surface at some point on crest vertical curve, or some physical features outside of the traveled way (AASHTO, 2001). Some examples of sight obstructions are barriers, trees, vegetation, and bridge approach fill slopes.

2.3 HORIZONTAL ALIGNMENT

The design elements of the horizontal alignment are the tangent, the circular curve, and the transition curve. Straight roadway sections or tangents are connected by horizontal curves, which are normally circular curves with or without transition (spiral curves). TAC (1999) defines the horizontal alignment as "The configuration of the roadway as seen in plan and

generally consists of tangent sections, circular curves, and in some instances, spiral transitions.” The horizontal and vertical alignments are major components of a highway alignment, however, any deficiencies in the horizontal alignment tend to be more critical if design dimensions cannot be attained. Deficiencies in the horizontal alignment may result in some reduction in the posted speed on the facility, and create or introduce inconsistencies that the driver must deal with.

The horizontal alignment of a highway is a permanent feature of the facility, and is difficult and expensive to modify after it has been constructed. TAC lists six items that have specific relevance to horizontal alignment design: safety, design speed, topography, climatic conditions, traffic volumes, and location of existing major utilities. Horizontal circular curves are used to connect two tangents. In some instances, a transition curve is placed between a tangent and a horizontal curve to help provide a smoother and more comfortable transition from tangent to curve. The basic design features of horizontal alignment include minimum radius, transition curves, superelevation, and sight distance.

2.3.1 Horizontal Curves

There are four types of horizontal curves: simple, compound, reverse, and spiral (Figure 2.1). The circular curve is a segment of a circle with radius R . The point at which the curve begins is denoted as the point of curvature (PC), and the point at which it ends is known as the point of tangent (PT). The point at which the two tangents intersect is known as the point of intersection (PI) or vertex (V). A simple horizontal curve is typically described by its radius or by the degree of the curve (arc and chord). Compound horizontal curves consist of two or more curves in succession, turning in the same direction, with any two successive curves

having a common tangent point (Garber and Hoel, 1997). The function of compound curves is to maintain desirable shapes of the horizontal alignment at complex locations such as interchanges, intersections, and in difficult topographic locations. The AASHTO (2001) guidelines recommend that the radii of any two successive simple curves should not vary greatly to avoid any abrupt changes in the alignment. Reverse horizontal curves are useful in effecting a change of direction when conditions do not permit the use of simple curves. The alignment reversal reduces the needed lateral clearance on the first curve (Easa, 2003).

Figure 2.1 Horizontal Circular Curves

Reverse curves usually consist of two simple curves with equal radii turning in opposite directions with a common tangent. Reverse curves are seldom recommended in design practice as sudden changes to the alignment may result in driver error (Garber and Hoel, 1997). In instances where a reverse in the alignment is required, it may be preferable to add a tangent of sufficient length between the two simple horizontal curves. Easa (1994) developed some design considerations for sight distance on highway reverse curves.

A horizontal transition (spiral) curve is placed between tangents and circular curves or between two adjacent circular curves having substantially different radii. A transition curve is a curve whose radius continuously changes. Research has shown that vehicles shift laterally in the traffic lane during their entry to (or exit from) a horizontal curve (Bonneson, 2000). As well, most drivers momentarily adopt a path radius that is sharper than that of the roadway curve. The use of transition curves provide a vehicle path that gradually increases or decreases the radial force as the vehicle enters or exists a circular curve. For simple transition curves, the radius varies from infinity at the tangent end to the radius of the circular curve at the curve end. For segmental spirals, the radius varies from that of the first circular curve to that of the second circular curve (Easa, 2003). Transition curves are advantageous in highway and roadway design because they provide a natural smooth path, provide a length for attaining superelevation, facilitate pavement widening on curves, and enhance the appearance of a highway.

2.3.2 Minimum Radius Requirements

The minimum radius of a horizontal curve is typically the limiting value of curvature for a given design speed and is determined from the maximum superelevation and the maximum

side friction factor for that given design. The use of sharp curves for a particular design speed would require superelevation rates that are beyond the limit considered practical or for operation with tire friction and lateral acceleration beyond what is considered comfortable by many drivers (AASHTO, 2001). The minimum radius of the curve is based largely on the level of driver comfort rather than safety.

As a vehicle moves in a circular path, it undergoes a radial acceleration that acts towards the centre of curvature. This acceleration is sustained by the friction between the tire and pavement and, if the road is superelevated, by a component of gravity. Lateral acceleration represents the portion of centripetal acceleration sustained by friction. The design of horizontal curves should be based on an appropriate relationship between the design speed and curvature, and on their joint relationships with superelevation and side friction. The relationship between the radius, speed, superelevation, and the lateral friction factor as defined by (TAC, 1999):

$$e + f = \frac{V^2}{127R} \quad (2.1)$$

where,

- e = Pavement superelevation
- f = Lateral friction force factor between the vehicle tire and roadway pavement
- V = Speed of vehicle (km/h)
- R = Radius of curve (m)

A diagram of a vehicle operating on a superelevated circular curve of constant radius at a constant speed is presented in Figure 2.2. When $F = 0$, the total resistance to the centrifugal force is provided by superelevation. This situation may occur on a large radius horizontal curve, with slow moving vehicles under icy weather conditions. If the values for e , V , and R are such that the pavement and tires cannot supply enough lateral friction, the vehicle will lose stability and start to skid (TAC, 1999). The main objective of horizontal curve design is to provide a radius and superelevation rate that combine to yield a safe and comfortable lateral acceleration for a reasonably large percentage of drivers (Bonneson, 2001).

Figure 2.2: Dynamics of Vehicle on Circular Curve (TAC, 1999)

where,

- W Weight of vehicle
- P Centripetal force (horizontal)
- F Frictional force between tires and roadway ($F = fW \cos \alpha$)
- α Superelevation ($e = \tan \alpha$)

2.3.2.1 Superelevation

The transition from a tangent, normal crown section to a curved superelevated section must be accomplished without any appreciable reduction in speed and in such a manner to ensure the safety and comfort of all occupants in the vehicle. The maximum superelevation that can be applied in the design of highways is affected by climatic conditions, terrain, type of environment, frequency of slow moving vehicles, and maintenance (TAC, 1999). In areas where snow and ice are a factor, the rate of superelevation should not exceed the rate on which a stationary or slow moving vehicle would slide toward the centre of the curve under icy conditions. For vehicles travelling at high rates of speeds on curves with poor drainage, hydroplaning can occur. Some vehicles with a high centre of gravity such as sport utility vehicles (SUVs) and trucks have a high percentage of their weight carried by the inner tires while traversing a circular curve. This can cause a rollover if the vehicle is travelling at slow speeds.

In Canada, the maximum values for superelevation used are 0.04 m/m, 0.06 m/m, and 0.08 m/m (4%, 6%, and 8%) depending on environment and the degree of surface icing that is likely to occur (TAC, 1999). In the United States, the highest superelevation rate for highways in common use is 10 percent (0.1 m/m), and in some instances, 12 percent (0.12 m/m) is used. In areas where snow and icy conditions are likely to occur, superelevation rates greater than 8 percent should not be used (AASHTO, 2001).

Generally, a curve of minimum radius requires superelevation and side friction factors at their maximum, while a curve with a very large radius (very flat radii), no superelevation is required (i.e., normally sloped cross section). Bonneson (2001) compared the methods for

the distribution of superelevation of four international highway agencies to the theoretical upper bounds of superelevation rate. The results showed that a wide range of guidance is provided by international agencies and suggests that a range of superelevation rates will likely provide safe and efficient operation for a given curve radius. Nicholson (1998) found that having the superelevation and friction varying linearly with the degree of curvature leads to greater alignment consistency and improved safety. A method for selecting the optimum superelevation rates for a system of horizontal curves has been developed (Easa, 2003).

2.3.2.2 Side Friction Factor

The side friction factor (f) is the ratio of the lateral friction force and the component of the weight of the vehicle perpendicular to the pavement (Figure 2.2). This force is applied to the vehicle at the tires and is toward the centre of the curve producing a radial acceleration (TAC, 1999). Since the speeds that drivers operate their vehicles vary greatly on curves, there is generally an unbalanced force that is produced. This force results in side thrust, that is counterbalanced by the friction between the tires and the pavement surface (AASHTO, 2001). The coefficient of friction, f is the frictional force divided by the component of the weight perpendicular to the pavement surface. The coefficient f is also referred to as the lateral ratio, cornering ratio, unbalanced centrifugal ratio, friction factor, and side friction factor. The upper limit of the side friction factor is the point at which the tire begins to lose traction or skid. This is referred to as the point of impending skid. Since horizontal curves on highways are designed with a high degree or margin of safety to avoid any chances of skidding, the f values used in the design process are substantially lower than the value of friction at the point of impending skid. The side friction factor at the point of impending skid

is dependent on factors such as vehicle speed, roadway surface conditions, type of vehicle, and tire type and pressure. Morrall and Talarico (1994) conducted a study that measured the side friction demanded and lateral acceleration of several different vehicles at the moment of impending skid conditions on horizontal curves.

The skid resistance condition of the pavement surface is very critical to highway safety. Emergency or evasive manoeuvres such as braking, sudden lane changes, and directional adjustments within a single lane, can significantly add to the frictional demands on the roadway geometry (AASHTO, 2001). When these manoeuvres are performed, it is usually for only a short period of time, and although high friction demands may exist, the amount of time required may not be enough to facilitate a corrective response from the driver resulting in unsafe operating conditions.

One major issue with the side friction factors used by the major international highway agencies is that the factors were developed almost 50 years ago. The vehicle and tire technology, as well as pavement surfaces have all evolved with the time and thus the relevance of these factors is now questioned. Today's drivers have become more tolerant of lateral acceleration through changes in attitude and vehicle cornering capabilities (Bonneson, 2001).

2.3.3 Tangents

Tangents are straight sections of the horizontal alignment, and can be beneficial to a highway design. Tangents can be used to achieve passing sight distance on two-lane highways, and for adapting the alignment to railroad sections, canals, and other man-made constraints

(Lamm et al., 1999). Long tangents having a constant grade have several drawbacks. Some of the disadvantages are they lead to excessive speeding, increase the effects of glare from on-coming vehicles at night, and can be monotonous causing driver fatigue and boredom. Tangents are typically separated into two separate groups depending on the length of the tangent.

An independent tangent is defined as tangents having a length long enough so that a driver exceeds the 85th percentile speed differences. Nonindependent tangents are tangents that are too short to exceed the possible 85th percentile speed difference. The maximum speed achieved on a tangent section is largely dependent on the tangent length, the sharpness of the curves on either end of the tangent, and driver's desires based on the general character of the roadway (Ottesen, et al., 2000). When designing a tangent, one critical factor that must be addressed is the length of the tangent. Due to glare caused by headlights at night, and the danger of drowsiness, tangent lengths (m) should not exceed 20 times the design speed V_D (km/h). Some highway agencies suggest that desirable tangent lengths should be at least six times the design speed.

Independent tangents are an issue in highway safety as it affects driver consistency. In situations where an independent tangent is used between a reverse curve, a driver is able to accelerate or decelerate in such a way that even the maximum allowable operating speed changes for fair design levels (Lamm et al., 1999). This can cause a critical driving manoeuvre to occur, such as over steer or under steer, loss of control, or in extreme situations, rollover. Therefore, when considering placing an intermediate tangent between two circular curves in the opposite direction (reverse curves), the engineer or designer must

carryout an evaluation of tangents in the design process. The Swiss Association of Road Specialists (1991) developed an evaluation process for identifying nonindependent and independent tangents.

2.3.4 Spiral Curves

As a vehicle operates on a circular horizontal curve, it follows a transition path as it enters or leaves the curve. On most curves, drivers can follow a suitable path within the limits of normal lane width. A combination of high speed and sharp curvature can cause a driver to take a longer transition path, and can result in shifts in lateral position and sometimes encroachment on adjoining lanes (AASHTO, 2001). As a result, spiral curves are used between the tangent and the sharp circular curve, and between circular curves of substantially different radii. A spiral curve is a curve with a constantly varying radius. Spiral curves help provide a natural, easy-to-follow path for drivers, such that the lateral force increases and decreases gradually as a vehicle enters and leaves a circular curve. A spiral curve provides a convenient and desirable arrangement for developing superelevation runoff (TAC, 1999).

In geometric design, the spiral form most commonly used is the clothoid. A clothoid has the relationship where R varies with the reciprocal of L , where R is the radius of curve at a distance L from the beginning of spiral ($R \propto 1/L$). The spiral relationship is given by (TAC, 1999):

$$A^2 = RL \quad (2.2)$$

where A is a constant called the spiral parameter and has units of length.

Table 2.1: Minimum Spiral Parameters for $e_{\max} = 0.06$ (TAC, 1999)

V_D (km/h)	$R_{\min}$ (m)	A (m)
40	55	50
50	90	65
60	130	85
70	190	110
80	250	125
90	340	160
100	440	190
110	600	220

The spiral parameter is a measure of the flatness of the spiral; the larger the parameter, the flatter the spiral. The minimum spiral parameters for various design speeds and for $e_{\max} = 0.06$ are presented in Table 2.1. The spiral parameter is developed based on the following criteria; comfort, superelevation runoff, and aesthetics. The minimum spiral requirement for design is the highest of the three values (TAC, 1999). It is important to note that for smaller radii, or when the minimum radius is used, the comfort criterion controls. When a reverse curve is used TAC recommends using successive simple spiral curves in which a spiral curve of increasing radius is followed immediately by a spiral curve of decreasing radius without a length of tangent between them. However, the alignment will have an improved appearance if an intermediate tangent is inserted between the two spirals.

2.3.5 Safety Considerations

Several research studies have revealed some interesting relationships regarding horizontal circular curves and collision rates (Choueiri et al., 1994). A review of collision spot maps

normally shows that collisions tend to cluster on curves, particularly on very sharp curves. The general trend viewed today is that the collision risk decreases as the radius of the curve increases or as the degree of the curve decreases (Lamm et al, 1999). A study by Rumar (1985) analyzed 14,000 collisions on 9,000 km of two-lane roads in Sweden. His research demonstrated a reduction in collision rates occurred with increasing radii of horizontal curves and a significantly higher number of collisions occurred on sharper curves. A study by Krebs and Kloeckner (1977) in Germany and Lamm and Choueiri (1987) in the United States determined that road sections with curve radii of less than 200 m have collision rates that are twice as high as those on sections with curve radii greater than 400 m. Also, a radius of 400 m provides a cross point in safety, and that for radii greater than this cross point (400 m), the gain in safety is relatively small.

2.4 VERTICAL ALIGNMENT

The vertical alignment is characterized as essentially a 2D longitudinal profile or section cut vertically through the centreline of a road. The vertical alignment of a highway is the profile of the highway that shows its elevation along the length of its path and consists of grades or tangents connected by vertical curves. Vertical curves are parabolic in function, with the rate of change of slope being constant. Some of the basic design features of the vertical alignment are grades, critical length of grade, climbing lanes, emergency escape ramps, and vertical curve length. Various types of vertical curves are shown in Figure 2.3.

Vertical curves should be simple in application and should result in a design that is safe and comfortable in operation, pleasing in appearance, and adequate for drainage (AASHTO, 2001).

Figure 2.3: Different types of Simple Vertical Curves (AASHTO, 2001)

For driver comfort, the rate of change of grade (slope) should be kept within tolerable limits. This is important in sag vertical curves where gravitational and vertical centrifugal forces act in opposite directions (AASHTO, 2001). Vertical curves are classified as crest vertical curves (CVC) or sag vertical curves (SVC). In vertical curve design, a symmetrical curve has the vertical axis centred on the vertical point of intersection (PVI). The distance from the start of the curve VPC to the point of vertical intersection (PVI) is half the total length of the curve (L). The distance from the point of intersection (PVI) to the end of the vertical curve (vertical point of tangent, VPT) is equal to half the total length of the vertical curve. The entry and exit tangents can have either positive or negative grades (G).

In some instances such as critical clearance (overpass or bridge) or other controls, an unsymmetrical vertical curve may be used. Unsymmetrical curves consist of two consecutive (unequal) parabolic curves with a common tangent at the vertical point of intersection. The curve is described by the parameter Q , which is the ratio of the shorter curve to the total curve length (Easa, 2003). Easa (1994) has developed a vertical curve with two equal arcs to improve the comfort and smoothness of unsymmetrical vertical curves. The two-arc unsymmetrical vertical curve provides greater comfort, sight distance and aesthetics when compared to the traditionally designed unsymmetrical vertical curves. Easa and Hassan (1998) developed the length requirements of the equal arc unsymmetrical curve.

A vertical curve is traditionally designed as a parabolic curve that directly connects two tangents. A vertical transition curve has been developed for use before and after the parabolic curve (Easa and Hassan, 2000). The authors called the resulting vertical curve a transitioned vertical curve. This vertical curve consists of transition-parabolic-transition segments. The function of the transitioned vertical curve is similar to the spiralled horizontal curve, and is useful for sharp vertical alignments.

The selection of a minimum curve length or K factor for a particular design provides geometric control for vertical curves. For crest vertical curves, the constant K is dependent on the sight distance used for design, driver's eye height above the road surface H_e , and object height above the road surface H_o . For sag vertical curves, the design is generally based on a headlight criterion, and the constant K depends on stopping sight distance, headlight height H , and the upward divergence of the light beam from the longitudinal axis of the vehicle α (1°).

Figure 2.4: Simple Parabolic Vertical Curve

Table 2.2: Design Rates of Vertical Curvature K for Crest and Sag Vertical Curves (AASHTO, 2001)

Design Speed (km/h)	<u>K Value for Crest Curves (m)</u>		<u>K Value for Sag Curves (m)</u>
	Stopping Sight Distance (m)	Passing Sight Distance (m)	Stopping Sight Distance (m)
40	50	200	50
50	65	270	65
60	85	345	85
70	105	410	105
80	130	485	130
90	160	540	160
100	185	615	185
110	220	670	220

2.4.1 Crest Vertical Curves

Stopping sight distance (SSD) is an important factor in the design of vertical curves. Sight distances on vertical curves can be based on the average grade, G , over the deceleration distance. The minimum length of vertical curves is controlled by the required stopping sight distance, driver eye height, and object height. The required length of a curve is such that, at a minimum, the stopping sight distance is available at all points along the vertical curve. Two conditions arise when calculating the length of a crest vertical curve. The first is when the

sight distance is less than the length of the curve, and second, when the sight distance is greater than the length of the curve.

Taignidis et al. (2001) and Hassan (2003) address one major area of concern with the current crest vertical curve model. The problem is that the stopping sight distance is calculated based on the assumption that the grade of the braking section is zero ($G=0$). They recommend a method for calculating the required stopping sight distance partly or wholly on a crest vertical curve by using the average grade over the braking distance. Hassan (2003) developed a vertical curve model that can act as an extension to the traditional North American analytical models. Using these models along with other models for available SSD on vertical curves, it was shown that the current design practice might yield segments of the vertical curve that violate the sight distance design basis. Most researchers agree that providing sufficient sight distance on vertical curves will make for a safer design. However, one study conducted by Fitzpatrick et al. (2000) reported that the crash rates on rural two-lane highways with limited stopping sight distance are similar to crash rates on all two-lane rural highways.

2.4.2 Sag Vertical Curves

For sag vertical curves, AASHTO (2001) defines four criteria that have been established in the design of sag vertical curves: headlight sight distance, passenger comfort, drainage control, and general appearance. The stopping sight distance on a sag vertical curve is based on the distance illuminated by the headlights at night. Design standards are based on an assumed headlight height of 600 mm, and an upward divergence of the headlight beam of 1° . For overall safety on highways, sag vertical curves should be long enough that the light beam

is nearly the same length as the stopping sight distance. To determine the length of a sag vertical curve, the stopping sight distance must be determined.

2.5 DESIGN CONSISTENCY

One of the most important factors in geometric design and highway safety is how well a designer reduces or eliminates uncertainty for the unexpected driver. The more consistent road designs are over a wide geographic area, the more effective the designer's contribution will be to reducing collision occurrence (TAC, 1999). It is the responsibility of the designer to achieve "design consistency" in the design of different classes of roads and changes in topography and landscape. Design consistency is the conformance of a highway's geometry with driver expectancy, and its importance and significant contribution to road safety is justified with an understanding of the driver-vehicle-roadway interaction (Ng and Sayed, 2003).

When highway engineers are designing highways, they must always take into consideration the high degree of standards involved with selecting geometric elements. It is also very important to note that today's motorist may not expect major changes in design speed, cross section, or alignment standards between adjacent sections along a rural highway (Messer, 1980). Any abrupt geometric changes may be so inconsistent with the driver's basic expectations that delayed driving responses and incorrect decisions may occur and result in unsafe driving. The primary inputs to the driving task are roadway geometry, traffic conditions, and roadside environment. These inputs affect the driving task and determine the workload requirement on the driver. If an inconsistency exists which violates a driver's expectation, the driver may adopt an inappropriate speed, or manoeuvre that may lead to a

collision. When design consistency is carried out through a design, abrupt changes in geometric features for highway elements are eliminated, and the safety of the road is increased.

2.5.1 Design Consistency Measures

Most of the research and work in design consistency has focused on identifying quantitative measures for design consistency evaluation and on developing models for their estimation (Ng and Sayed, 2003). These measures are classified into four categories: operating speed, driver workload, vehicle stability and alignment indices. These four categories are widely used to evaluate design consistency of a highway. For this research, the operating speed and vehicle stability will be discussed.

2.5.1.1 Operating Speed

Operating speed is one of the most commonly used measures of design consistency. The operating speed is defined as the speed selected by the drivers when not restricted by other users, under free flow conditions, and is usually represented as the 85th percentile speed, or V_{85} . Operating speed is a good indication of design consistency, as when a driver experiences any inconsistencies in the geometry of the road, the driver will usually reduce their speed. The difference between the operating and design speed ($V_{85} - V_D$) is a good indication of the level of inconsistency that exists at one element. To evaluate the level of inconsistency between two successive elements, the speed reduction between two successive elements (ΔV_{85}) must be analyzed. Lamm et al. (1999) have established consistency evaluation criteria for operating speed measures and are presented in Table 2.3.

Table 2.3: Consistency Evaluation Criteria

Evaluation	Criterion I	Criterion II
Good	$ V_{85} - V_d \leq 10 \text{ km/h}$	$\Delta V_{85} \leq 10 \text{ km/h}$
Fair	$10 < V_{85} - V_d \leq 20 \text{ km/h}$	$10 < \Delta V_{85} \leq 20 \text{ km/h}$
Poor	$ V_{85} - V_d > 20 \text{ km/h}$	$\Delta V_{85} > 20 \text{ km/h}$

Three classification cases exist for the system – Good Design, Fair Design, and Poor Design. This classification provides appropriate tuning between design speeds and the operating speed for good, fair and poor design levels. A poor design occurs when the difference between the operating speed and the design speed is greater than 20 km/h for a section of road.

A selected design speed should be consistent with the speed that the drivers are likely to expect on a given highway facility (AASHTO, 2001). Several models have been developed to estimate or predict the operating speed. Gibreel et al. (2001) developed a set of models that consider the 3D nature of highways. These models were developed from speed data collected from highways in Ontario, and predict the operating speed on horizontal curves in sag or crest vertical curve combinations. The operating speed on tangents connecting horizontal curves is also very important for design consistency. The length of a tangent also has an effect on speed reductions that occur as a vehicle enters a horizontal curve. A long tangent is defined as an independent tangent. On independent tangents, drivers are able to accelerate up to their desired operating speed, and upon entering the next horizontal curve,

must apply the brakes to reduce their speed. A reduction in speed greater than 20 km/h is a sign of inconsistency. On non-independent tangents, the tangent is not long enough for drivers to accelerate to high desired operating speeds. Thus speed reductions are less than 20 km/h as the driver enters the horizontal curve.

2.5.1.2 Vehicle Stability

Vehicle stability is an important measure of safety and design consistency. When insufficient side friction is provided at a horizontal curve, vehicles may skid out, rollover, or involve in head-on collisions (Ng and Sayed, 2003). The design of such a roadway violates the ability of the drivers to guide and control the vehicle in a safe manner. Locations that do not provide sufficient side friction or vehicle stability can be considered as geometric design inconsistencies. The difference between the assumed side friction and the side friction demanded (Δf_R) is used to represent vehicle stability. Lamm et al. (1999) developed an evaluation criterion based on Δf_R for consistency (Table 2.4).

Table 2.4: Consistency Measure for Vehicle Stability

Evaluation	Criterion III
Good	$\Delta f_R \geq 0.01$
Fair	$0.01 > \Delta f_R \geq -0.04$
Poor	$\Delta f_R < -0.04$

2.5.2 TAC and Design Consistency

TAC has taken a lead among North American transportation agencies and organizations in starting to incorporate design consistency into geometric guidelines. The more consistent road designs are over a wide geographic area, the more effective the designer's contribution will be to reducing collision occurrence (TAC, 1999). It is the responsibility of the designer to achieve "design consistency" in the design of different classes of roads and changes in topography and landscape. TAC suggests that there are three principles by which a designer can evaluate the consistency of a design. The three principles are; cross-section consistency, operating speed consistency, and driver workload consistency. The most important item needed in evaluating design consistency is accident or collision data for the highway under review.

2.6 VEHICLE STABILITY

In this section, some of the ride characteristics and basic vehicle movements will be discussed. Also, background will be provided on some of the vehicle stability models that have been developed.

2.6.1 Vehicle Dynamics and Ride Characteristics

Vehicle dynamics focuses on the movements of different vehicles (automobiles, trucks, buses, and special purpose vehicles) on a road surface. The primary forces by which a high-speed motor vehicle is controlled are developed in four locations – the contact area between the tires and the road. To understand vehicle dynamics, the forces and moments generated

by the rubber tires at the ground must be clearly defined and known. Vehicle performance and dynamics consists of several motions – acceleration, braking, and handling. The forces imposed on the vehicle from the tires, gravity, and aerodynamics determines the dynamic behaviour. To accurately describe the performance of a vehicle on a road, very complex mathematical models are used. These models are also used in the design and construction of the vehicles. A vehicle is composed of five individual parts that are connected in a movable way (including the wheels and chassis). Every movable body is considered a rigid body with six degrees of freedom – three rotational and three translatory – resulting in $5 \times 6 = 30$ degrees-of-freedom for the entire vehicle (Lamm et al., 1999).

In this section, the different movements making up vehicle dynamics will be briefly discussed. Since each movement is very complex in nature, only the basics or fundamentals of each movement will be reviewed.

2.6.1.1 Acceleration

The engine of a motor vehicle is the source of propulsive power. Engines are generally characterized by their torque and power (horsepower) curves as a function of speed (Gillespie, 1992). The ratio of engine power to vehicle weight is the first-order determinant of acceleration performance. At lower speeds, an upper limit on acceleration can be obtained by neglecting all resistance forces acting on the vehicle. From Newton's Second Law,

$$M a_x = F_x \quad (2.3)$$

Where:

M	=	Mass of the vehicle = W/g
a_x	=	Acceleration in the forward direction
F_x	=	Tractive force at the drive wheels

Since the drive power is the tractive force multiplied by the forward speed, Equation 2.3 can be written as:

$$a_x = \frac{1}{M} F_x = 550 \frac{g}{V} \frac{HP}{W} \text{ (ft/sec}^2\text{)} \quad (2.4)$$

Where:

g	=	Gravitational constant (32.2 ft/sec ²)
V	=	Forward speed (ft/sec)
HP	=	Engine horsepower
W	=	Weight of the vehicle (lb)

Since the velocity term is in the denominator, the acceleration capability of a motor vehicle decreases with increasing speed. Automobiles have much higher performance levels than heavy trucks due to the low power-to-weight ratio of trucks.

2.6.1.2 Braking Performance

The braking performance of a motor vehicle is a function of many factors. The general equation for braking performance can be obtained from Newton's Second Law:

$$M a_x = -\frac{W}{g} D_x = -F_{xf} - F_{xr} - D_A - W \sin \Theta \quad (2.5)$$

where:

D_x	=	$-a_x$ = Linear deceleration
F_{xf}	=	Front axle braking force
F_{xr}	=	Rear axle braking force
D_A	=	Aerodynamic drag
θ	=	Uphill grade

The front and rear braking force terms are attributed to the torque from the brakes as well as the rolling resistance effects, bearing friction, and driveline drags. A detailed analysis of the deceleration dynamics requires a complete knowledge of all these factors acting on the vehicle (Gillespie, 1992). Many forces act on a vehicle that may produce braking deceleration, although the brakes are the primary source. Some examples of other braking forces are the rolling resistance, aerodynamic drag, driveline drag, and grade of the road.

Automotive brakes used today come in two forms – drum, and disc brakes. Drum brakes have been used in the U.S over the years due to their high brake factor and easy incorporation of parking brake features. However, drum brakes may not be as consistent in torque performance as disc brakes. Disc brakes have a lower brake factor, and require higher actuation effort and development of integral parking brake features.

Over the past few decades, an advanced type of braking system has become more popular in automobiles. The antilock braking systems (ABS) have become more common in today's automobiles. In classical braking systems, the proportioning is adjusted directly. In an ABS system, the system senses when wheel lockup occurs, releases the brakes momentarily on locked wheels, and reapply them when the wheel spins up again. Automobiles equipped with

antilock braking systems have more control when conducting a braking manoeuvre, and can stop with shorter distances.

2.6.2 Handling – Cornering

The cornering behaviour of a motor vehicle is an important performance mode that is also referred to as the handling characteristics. Handling is defined as the responsiveness of a vehicle to driver input, or the ease of control (Gillespie, 1992). Handling can be considered an overall measure of the vehicle-driver combination. The driver and vehicle is a “closed loop” system where the driver observes the vehicle direction or position, and corrects his/her input to achieve the desired motion. When characterizing only the vehicle, an “open loop” is the desired method. An open loop refers to vehicle response to specific steering inputs, and is more precisely defined as “directional response” behaviour.

In cornering analysis, the first step is to examine the low-speed turning behaviour of a motor vehicle. At lower speeds (parking lot manoeuvres), the vehicles tires do not need to develop lateral forces. As a result, the tires roll with no slip angle. When the rear wheels have no slip angle, the centre of turn must lie on the projection of the rear axle. At higher speeds, the dynamics of turning are different because lateral acceleration is present. To counteract the lateral acceleration, the tires must develop lateral forces, and slip angles will be present at each wheel. Under cornering conditions, in which the tire must develop a lateral force, the tire will also experience lateral slip as it rolls. The angle between its direction of heading and its direction of travel is known as the slip angle.

2.6.3 Vehicle Rollover

Vehicle rollover is one of the most serious and dangerous dynamic manoeuvres that a motor vehicle can experience, and can have adverse effects on the occupants of the vehicle. Vehicle rollover is described as any manoeuvre in which the vehicle rotates 90 degrees or more about its longitudinal axis such that the body makes contact with the ground (Gillespie, 1992). Many factors may cause or influence a vehicle to rollover. It may occur on flat and level surfaces when the accelerations on a vehicle reach a level beyond that which can be compensated by lateral weight shift on the tires. The cross slope of a road surface, disturbances to the lateral forces arising from curb impacts, soft ground, or other obstructions may also cause a vehicle to roll.

As mentioned previously, a cross slope or superelevation is used in geometric design to help counterbalance the lateral acceleration. With the radius of curve known, and the design speed defined, a cross slope is chosen such that produces a lateral acceleration in the range of zero to 0.1 g. As a vehicle traverses the horizontal curve, the lateral acceleration increases and the forces on the inside wheels must be reduced. It is through this process that the vehicle acts to resist, or counterbalance, the roll moment in cornering. The limiting cornering condition occurs when the entire load has been transferred to the outside wheels, and the load on the inside wheels are equal to zero. At this critical point, rollover will begin as the vehicle can no longer maintain equilibrium in the roll plane. The lateral acceleration at which rollover begins is defined as the rollover threshold. The rollover thresholds for a number of different vehicles are presented in Table 2.5. The lateral acceleration required to reach the rollover of passenger cars and light trucks exceeds the

Table 2.5: Rollover Threshold Values for Different Vehicles (Gillespie, 1992)

Vehicle	CG Height (inches)	Tread (inches)	Rollover Threshold (g)
Sports Car	18-20	50-60	1.2-1.7
Compact Car	50-60	50-60	1.1-1.5
Luxury Car	20-24	60-65	1.2-1.6
Pickup Truck	30-35	65-70	0.9-1.1
Passenger Van	30-40	65-70	0.8-1.1
Medium Truck	45-55	65-75	0.6-0.8
Heavy Truck	60-85	70-72	0.4-0.6

cornering capabilities arising from the friction limits of the tires. Thus, it is possible for the car to spin out on a flat level section of roadway without rolling over. For heavy trucks, it is possible to reach the rollover threshold within the friction limits of the tires. As a result, heavy trucks are at risk of rollover if the driver allows the vehicle to spin out on a dry road surface.

2.6.4 Vehicle Stability Models

Vehicle stability is a good measure of design consistency and highway safety. Several models are available to simulate and quantify vehicle dynamics. Each model has its own advantages and disadvantage over the other models. In this section, the point mass model, bicycle model, and two-axle model will be discussed.

2.6.4.1 The Point Mass Model

The point mass model represents steady state turning on a two dimensional superelevated horizontal alignment with the vehicle traveling at a constant speed. The vehicle is represented as a simplified object having its mass concentrated at a single point with three normalized forces acting on the mass in the radial direction. The three forces are the centrifugal force, the component of the vehicle's weight, and the lateral frictional force. By knowing the vehicle's mass, speed (V), superelevation (e), and radius of curvature (R), the standard point mass formulation can be derived. The point-mass model is overly simplified, and has its obvious shortcomings. Since the model is 2D in nature, it does not consider any overlapping vertical alignment. Also, since the point mass model approximates the vehicles mass to a single point, there is no consideration given to the distribution of frictional forces between the inner and outer front, and rear tires.

Furthermore, the point-mass model assumes that the vehicle is following an identical path to the actual curve radius. In reality, this does not occur, as drivers will operate their vehicles differently depending on many different conditions. The final shortcoming is that the point mass model assumes a constant rate of speed (V), so any erratic or abnormal driving manoeuvres are not accurately represented (MacAdam et al., 1985).

2.6.4.2 The Bicycle Model

The bicycle model is an extension of the point mass model, and includes a front and rear tire located at a distance ' a ' and ' b ' respectively from the centre of mass. The introduction of the front and rear tire now requires two frictional factors acting in the radial direction to be

considered in the force equilibrium process. The forces acting on the mass do not coincide at a single point, and thus, it is necessary to establish both the moment and force equilibrium. The additional equation resulting from the sum of the moments facilitates establishing a solution for the additional friction force that has been introduced (MacAdam et al., 1985). By assuming the rear and front friction factors are identical, simply distributing the frictional forces to a front and rear tire result in friction factors that are identical to the values resulting from the point-mass formula.

2.6.4.3 Two-Axle Model

The two-axle model improves further on the bicycle model by adding additional width to the bicycle model in the form of left and right side tires and a mass centre elevated some distance from the ground. The addition of the two extra tires results in two additional friction factors, f_3 and f_4 , which represent the tires on the outside of a turn. By elevating the centre of mass, the model allows for the weight of the vehicle to shift laterally. This requires roll, force and moment equilibrium for the vehicle to maintain stability. The roll equilibrium equation provides the additional equation required to solve for the friction at the outer tire. Since the forces acting on the mass do not coincide at a single point, it is necessary to establish both the moment and force frictional factors introduced by accounting for the width of the vehicle (MacAdam et al., 1985). During the cornering manoeuvre, due to the effect of body roll and the transfer of vehicle weight from the outside tires, the friction factors that result from the two-axle model differ from the values obtained from the other models. Since the vertical load is reduced on the inside tires, the lateral friction factors obtained tend to be larger than provided by the point-mass formulation. The opposite occurs to the tires on the outside of

the curve, as the resulting lateral friction factor is smaller than that derived from the point-mass model.

2.6.5 Vehicle Stability Simulation Software

Many types of vehicle simulation software are available for use by engineers and designers. Car manufactures such as Volkswagen have developed systems that allows for simulated rides in a laboratory environment that can provide “real life” driving data (Lamm et al., 1999). The system allows for the possibility of modeling a landscape with roads, houses, people and vehicles all on a computer. As well, a research project in Europe called PROMETHEUS (Program for a European traffic with highest efficiency and unprecedented safety) is one of the better known projects in the field of vehicle simulation. Thirteen European car manufacturers and several industrial, private and research groups have participated in this project.

Over the past few years, the U.S. Department of Transportation’s Federal Highway Administration (FHWA) has finished developing the Interactive Highway Safety Design Model (IHSDM). Three major software programs were competing to be included within the IHSDM. The three vehicle simulation programs are: NADSdyna, VDANL, and VDM RoAD. The following sections will provide an overview and a brief description for each software program.

2.6.5.1 NADSdyna

In the later part of the 1980's, the National Highway Traffic Safety Administration (NHTSA) started researching the use and development of a new state-of-the-art driving vehicle simulator. This driving simulator was called the National Advanced Driving Simulator, or NADS. A site selection competition was conducted among several American Universities, with the winning university being the University of Iowa. The University of Iowa contributed financing, and the core of the vehicle dynamics software for the NADS. The software program was the Real Time Recursive Dynamics, (RTRD), and is a minimum coordinate set, multi-body dynamics program that breaks the vehicle chassis and suspension system into a tree topology by cutting closed kinematic loops (Tsai and Haug, 1989). The tree topology allows the multi-body system to be solved on parallel computers in "real time".

The core vehicle dynamics (RTRD) software was then supplemented with a collection of sub-modules that modeled different components of a vehicle such as the power train, tires, brakes, and steering systems. The finalized program containing the RTRD software along with the sub-modules is the NADSdyna. The Vehicle Research and Test Center in East Liberty, Ohio tested the validity of the NADSdyna simulation software. NADSdyna was compared with a 1994 Ford Taurus GL, and found that the variation in lateral acceleration was $+0.95 \text{ m/s}^2$ at the 95% confidence level (Chstos and Crygier, 1997).

2.6.5.2 Vehicle Dynamic Analysis Non-linear Model (VDANL)

The Vehicle Dynamic Analysis Non-linear Model or VDANL is a comprehensive vehicle dynamics simulation program configured to run on personal computers in a Windows

environment. The NHTSA and Systems Technology Incorporated (STI) developed VDANL in the late 1980's. The simulation software was developed for the analysis of passenger cars, light trucks, and multipurpose vehicles, and over the years, has been upgraded to handle articulated vehicles such as tractor semi trailers. The VDANL software now permits analysis of driver-induced manoeuvring up through limit performance conditions defined by tire saturation characteristics, as well as driver feedback control features. Due to the underlying complexity of the software and its detailed inputs, STI developed a simplified user interface for highway designers. The interface receives as inputs the user's choices of roadway design file, vehicle type, and desired manoeuvre and then initiates the simulation using vehicle and driver characteristics stored in hidden parameter files (Lamm et al, 1999).

VDANL was selected by the FHWA for use in the IHSDM. The driver performance module in IHSDM estimates the driver's speed and path along a section of roadway and this information is provided as input to VDANL, which estimates vehicle kinematics such as lateral acceleration, friction demand, and rolling movement. The output provided by VDANL can be used to identify conditions and scenarios that could result in a loss of vehicle control (skidding or rollover).

2.6.5.3 VDM RoAD

The Vehicle Dynamics Model (VDM) was conceived as a user-friendly means of providing design engineers with safety-related information on any alternative alignment. The VDM software allows a designer to "drive" a selected design vehicle through the alternative design alignments and develop a speed profile and data on lateral accelerations. This ability to "drive" through a design provides designers with a visual method for examining poor design

scenarios. The VDM software would be able to predict and model the speed-dependent lateral accelerations, which largely determine a vehicle's probability of staying safely on the road. Any excessive or rapid changes of lateral acceleration may indicate an incompatible or inconsistent geometric design that requires further evaluation and improvement. VDM was originally designed to be one of the modules of the IHSDM. The Vehicle Dynamic Roadway Analysis and Design model, or VDM RoAD was developed by the University of Michigan and is one of the most advanced vehicle stability models. VDM RoAD accurately simulates a vehicle traveling through a user-defined alignment, taking into account vehicle characteristics such as body roll, pitch, yaw, and lateral weight distribution.

VDM RoAD analyzes the lateral acceleration, directional control, roll stability, and stopping sight distance. VDM RoAD is an extension of two commercially available computer programs, namely TruckSim and AutoSim. The model accurately simulates vehicle dynamics behaviour of passenger cars, trucks, and two-unit combination vehicles on 3D road alignments. The model uses a fairly rudimentary wire-frame visualization program, and can run on either DOS-based personal computers or UNIX-based workstations. The software is capable of being run on Windows 95 and later versions, including Windows 98, Windows 2000, Windows ME, Windows NT, and Windows XP. On Pentium-Pro computers, the simulations run in real-time or faster. Thus, a run simulating a 2-minute test will finish in two minutes or less. VDM RoAD uses mathematical models that were developed and researched over the past few decades at the University of Michigan. A more detailed discussion and background covering the use and function of VDM RoAD will be covered in Chapter 3.

2.7 OTHER CONSIDERATIONS

In this section, other important considerations pertaining to highway safety and geometric design will be discussed.

2.7.1 Cross Section

The cross section of a highway is the portion of roadway where vehicles operate on. The cross section of a highway is composed of elements such as the traveled way, shoulders, curbs, medians, sideslopes and backslopes, clear zones, pedestrian facilities, and bicycle facilities. The road cross sections, alignments, and intersections are essential elements of the roadway, and they influence traffic safety. It is important that when designing the cross section elements of a roadway or highway, the engineer or designer should aim for consistency, aesthetics, and harmonizing all major elements.

2.7.2 Three-Dimensional (3D) Alignment

Typically transportation and highway geometric analysis has been conducted in only one or two dimensions. In the past, the design of the 3D nature of highway alignments was believed to require cumbersome and lengthy computations because of its mathematical complexity and computer intensive requirements (Easa, 2003). As a result, the horizontal and vertical alignments of a highway were designed independently (1D) or sometimes together (2D), but the 3D nature of the design was not considered (Easa et al., 2002). This practice has lead to design “inconsistencies” of highway alignments and is a major cause of driver fatigue and discomfort.

More recently, three-dimensional (3D) analysis of highway alignments is becoming a more common practice. Several researchers such as Easa and Dabbour (2004), Easa et al. (2002), Gibreel et al. (2001), Hassan et al. (1998), and Hassan (1996) have investigated and analyzed the 3D nature of geometric alignments. Hassan et al. (1998) conducted 2D analytical modeling of the geometric elements of a highway design. He determined design requirements and the locations where 2D based horizontal and vertical curves should not overlap. For the combined alignments, the 2D design was found to overestimate or underestimate the required radius of horizontal curves and the length of vertical curves.

The effects of combined alignments on the drivers perception of horizontal curve radii was examined by Hassan and Easa (2001) using computer animation. Their research revealed that the overlapping of sag curves causes horizontal curves to look flatter than they actually appear, and vice versa for overlapping crest curves. Gibreel et al. (2001) developed a 3D operating speed model for two-lane highways. The 3D model outperformed the 2D models in estimating the operating speed on horizontal curves.

Two studies conducted by Furtado et al. (2002) and Easa and Dabbour (2004) on vehicle stability for passenger cars and heavy trucks, respectively, illustrated a deficiency in the values for minimum horizontal curve radii due to 3D alignments. Furtado et al. (2002) found that the requirement for the minimum radius of a horizontal curve with vertical alignment was significantly higher compared with similar horizontal curves with no vertical alignment (level). Easa and Dabbour (2004) analyzed the effect of complex curvature and 3-dimensional alignments on the minimum required radius of horizontal curves. They found

that 3D alignments and complex curves both contribute an increase in the minimum required radius for horizontal curves.

A comprehensive vehicle dynamics model (VDM) has been developed to simulate virtually all performance conditions. This model has been incorporated into the IHSDM as a vehicle stability module. This module will help designers to simulate and analyze the vehicle stability and dynamics of a variety of different vehicles by driving them through different 3D alignments.

2.7.3 Interchange Ramps

Interchange ramps act as an interface between freeways and the surface street network and in effect should provide a smooth and safe transition from one facility to another (Liapis et al., 2001). Interchange ramps are a good application for the coordination between horizontal and vertical alignments. Interchanges provide a high degree of traffic safety and capacity. The selection of the appropriate type of grade separation and interchange is influenced by the highway classification, traffic type, design speed, and degree of access control. Typically, interchanges are characterized as having sharp horizontal curves of different types all combined with large differences in elevation, and being either rise or fall. Interchanges vary from single ramps connecting local streets to complex and comprehensive layouts that involve two or more highways (AASHTO, 2001). The design and layout of any specific ramp and type of traffic movement will reflect surrounding topography, cost, and degree of flexibility in desired traffic operations.

Interchange ramps have the same design principles as the guidelines governing horizontal alignments. Interchange ramps are typically complex curves such as compound or reverse curves. Compound or spiral transitions are generally used to help obtain the desired alignment of the ramp, to provide for a comfortable transition between the design speeds of the through and turning roadways, and to fit the natural paths of the vehicles (AASHTO, 2001).

The sight distance requirements along a ramp should be at least as great as the design stopping sight distance. A clear and unobstructed view of the entire exit terminal should be provided. The profile of a typical ramp consists of a central portion on an appreciable grade coupled with terminal vertical curves and connections to the profiles of the intersection legs. The general profile of the ramp is an “S” shape with a sag vertical curve at the lower end and a crest vertical curve at the upper end. Many of today’s interchange ramps may not fall within the scope of the original studies on which the AASHTO (2001) guidelines are based (Hunter et al., 2001).

2.7.4 The Interactive Highway Safety Design Model (IHSDM)

The Federal Highway Administration (FHWA) developed the IHSDM. The FHWA has been developing road safety evaluation software that combines the knowledge of safety into a more useful form for highway planners and designers. The latest version of the software was released in January 2003. IHSDM is a suite of software analysis tools that is used for evaluating safety and operational effects of geometric design in the highway project development process. The latest version of IHSDM is specifically designed for two-lane rural highways. Future versions of the software are being developed for use with multi-lane

highways (IHSDM, 2003). The IHSDM software is intended to be used by highway engineers and planners as an additional tool to enhance the design process. The software works by predicting the functionality of proposed or existing designs by applying chosen design guidelines and generalized data to predict the performance and non-performance of a design (IHSDM, 2003). The IHSDM software is comprised of five evaluation modules; Policy Review, Crash Prediction, Design Consistency, Traffic Analysis, and Intersection Review.

Each IHSDM module evaluates an existing or proposed geometric design from a different perspective and estimates measures describing one aspect of the expected safety and operational performance of the design. The Policy Review Module checks the design relative to the range of values for critical dimensions recommended by the guidelines (AASHTO, 2001). The Crash Prediction Module provides estimates of expected crash frequency and severity along a proposed or existing design. The Design Consistency Module estimates expected operating speeds and measures the operating-speed consistency of the design. The Intersection Review Module provides users of IHSDM with a review of intersection design elements relative to their likely safety and operational performance. The Traffic Analysis Module estimates measures of traffic operations used in highway capacity and quality of service evaluations.

The developers of the IHSDM software have mentioned three general cautions that should be considered in the use of IHSDM:

- 1) Measures of expected safety and operational performance from IHSDM are only a subset of the large number of inputs that must be considered in the decision making process.

- 2) Estimates from IHSDM are expected values, in the statistical sense. Actual performance may vary over time and among sites. The estimates from IHSDM should not substitute for, but rather supplement and complement local knowledge.
- 3) While derived from the best available data using the best available methods, both the available data and methods have limitations. The engineer's manuals for each module document limitations that should be understood to apply appropriately the resulting estimates (IHSDM, 2003).

2.7.5 Truck Transport and Safety

Trucks are a small volume, low weight, fast, and flexible mode of transportation (Figure 2.6). The extensive roadway network is the major cause of the popularity and efficiency of this mode of transport. Truck carriers are able to transport a variety of goods and provide a multitude of services such as armoured truck service, agricultural commodities, building materials, dump trucking, forest products, hazardous materials, heavy machinery, household goods, moving services, petroleum products, refrigerated goods, rental services, retail store items, and vehicles (Papacostas and Prevedouros, 1993). The major advantages of this mode of transportation are the high speed and high accessibility. The main disadvantages of this mode are the limitations in volume and weight, as well as the high rates charged compared with railroads, particularly for distances over 1000 miles. In the United States, the interstate motor carrier (trucking) industry is the largest transport industry with 592,909 operators and

Figure 2.6: Late Model 5-Axle Freightliner Tractor-Trailer

over 8 million trucks operating on US highways (USDOT, 2002). On some roads, trucks represent a large enough percentage of vehicles that they may be considered the design vehicle (Donnell et al., 2001). Due to the extremely high value of vehicle miles traveled by most trucks, the risk and chance of collision occurrence is quite high. Several factors such as geometric conditions, environmental factors, type of loading, and driver performance all influence the performance and safety of a truck. In recent years, many studies have been performed to better understand and identify factors that contribute to crashes involving trucks (Daniel et al., 2002). Their research focused on truck crashes related to interstate or freeway driving conditions, as about 40% of large-truck miles occur on interstates. Approximately 24% of fatal truck crashes occurs on interstate highways.

One of the most important issues of highway and truck safety is rollover (Figure 2.7). Truck rollover is serious, and occurs frequently on many North American highways. Truck rollover results in injury, loss of life, traffic delays, environmental damage, property damage,

Figure 2.7: Truck Rollover on Tangent Section of 2-Lane Rural Highway.

and financial loss. The rollover thresholds for trucks of varying weight are presented in Table 2.6. Trucks traveling at high speeds are often caught off guard at exit ramps and tight curves that require a speed greatly reduced from the normal traveling speed on the freeway (Baker et al., 2001). In 1998, a study revealed that approximately 10 600 commercial vehicles were involved in non-fatal rollover accidents in the United States. Another study showed that rollover accidents are the fourth most common accident type involving commercial vehicles, and account for roughly 4% of all truck accidents. To help limit or control the chance of truck rollover, several truck rollover warning systems have been proposed and implemented. Two such systems are the Speed-Based and Weight Based Rollover systems. These rollover systems use Intelligent Transportation Systems (ITS) to warn the driver of potential rollover occurrences. These systems are helpful as they provide warning and an opportunity for the driver to react and prevent an accident for a truck in danger of a potential rollover. Other methods of limiting or preventing truck rollover are by

providing signage at high rollover risk locations such as low radii horizontal curves, complex horizontal curves such as reverse curves, and through driver education.

Table 2.6: Rollover Threshold for Various Weight Ranges of Nontanker Trucks (Baker et al., 2001)

Weight Range (kg)	Rollover Threshold (g)
0 - 15 875	0.73
>15 876 - 22 680	0.60
>22 681 – 29 484	0.50
>29 485 – 36 287	0.38
>36 288 – 45 359	0.36

CHAPTER 3

VDM RoAD SOFTWARE

The Vehicle Dynamics Models Roadway Analysis and Design (VDM RoAD) simulation software is one of the most advanced vehicle stability models. VDM RoAD was developed at the University of Michigan with the intention of being included within the Interactive Highway Safety Design Model (IHSDM) that was developed by the FHWA. VDM RoAD is a user-friendly integrated set of computer tools for simulating and analyzing the braking and handling behaviour of vehicles on selected road and highway designs.

This chapter presents a brief overview of VDM RoAD and gives some detail on the capabilities and limitations of the software and the reasons for selecting the software for use in this study. A description of the data inputs and requirements to run an analysis or simulation will also be discussed.

3.1 CAPABILITIES AND LIMITATIONS OF VDM RoAD

3.1.1 Capabilities

VDM RoAD simulates a vehicle running down a roadway alignment. The road geometry is described with an IHSDM interchange file as specified by the FHWA Geometrics Laboratory. VDM RoAD can analyse lateral acceleration, directional control, roll stability, and stopping sight distance. One of the main functions of VDM RoAD is to predict the

dynamic 3D motions of heavy trucks in response to steering and braking inputs. The braking performance, directional control, and stability mainly involve dynamics in the yaw plane, while roll stability is principally related to the roll plane (Sayers, 1999). The aerodynamic effect of vehicles has a secondary influence with respect to both concerns, and is ignored in VDM RoAD.

A diagram of a four-wheeled truck viewed from the top (yaw plane) is presented in Figure 3.1(a). Three equations govern the basic behaviour in the yaw plane: the sum of the tire shear forces and the horizontal hitch force must equal the vehicle mass times its acceleration in both the vehicle X and Y directions, and the moment of those forces about the vehicle yaw moment of inertia. Yaw behaviour is also influenced by the rotary motion of the vehicle bodies in roll and pitch. Mechanical energy is transferred as the vehicle rolls and pitches, and these motions contribute to the vehicle transient response.

A free-body diagram in the roll plane for a truck body and one axle as viewed from the rear when making a left hand turn is presented in Figure 3.1(b). The external forces acting on the vehicle system are due to the tires and any hitches connecting it to other vehicle units. The truck is also subjected to roll moments (M_h) from the hitches and from the vertical component of the tire forces. The overall roll stability is characterized by the balance of moments about the vehicle mass centre. When the tires lift off the ground, the forces are zero on one side of the axle. Lifting of the tires occurs first for tires on the axle furthest to the rear of all vehicle units in the system, on the side that is on the inside of the turn. When lift-off occurs for more than a critical number of axles, roll stability cannot be maintained. This condition is considered to be the onset of rollover. The basic factors required for

predicting roll stability are the vertical tire forces, and their points of application relative to the vehicle mass centre, which change as the vehicle rolls. VDM RoAD uses detailed non-linear tire models, non-linear spring models, and includes the major kinematic and compliance effects in the suspensions and steering systems in cars and heavy trucks. The kinematical and dynamical equations are valid for full non-linear 3D motions of rigid bodies.

Figure 3.1: Analytical model for truck stability in VDM RoAD: (a) Free Body Diagram in the yaw plane (X-Y) and (b) Free-Body Diagram in the roll plane.

VDM RoAD uses closed-loop controller algorithms to steer the vehicle along a specified path and to drive at a prescribed speed. Both speed and lateral position can be specified as constants or as variable functions of roadway station number. Braking inputs are specified as functions of time. It is important to note that VDM RoAD uses mathematical models similar to those in the commercial programs TruckSim and CarSim

3.1.2 Limitations

Despite the versatility and function of the VDM RoAD software, the developers of the software list some limitations. Although the dynamics of the vehicle models in VDM RoAD are identical to those in TruckSim and CarSim, options that would not be of interest to a road engineer or designer have been removed (Sayers, 1999). As well, output variables involving suspension and tire behaviour are not included in the output files. Inputs that would be used to simulate proving ground tests (steering wheel angle as a function of time) are not supported. VDM RoAD does not simulate complex human factors such as braking as a reaction to increasing lateral acceleration when it exceeds driver comfort limit, or steering as a reaction to increasing roll angle where a rollover is likely to occur (Dabbour, 2003). Another important factor related to truck stability is the aerodynamic forces. This is not included or addressed in the VDM RoAD truck models.

3.2 REASONS FOR SELECTING VDM RoAD

In selecting a vehicle dynamics simulation software to use for this work, it was important that the simulation software had the following features:

- Ability to input any user-defined roadway alignment including both 2D and 3D combined alignments.
- Ability to select different types of vehicles to run down the specified alignments.
- Clear presentation of the output and results of the vehicle dynamics simulation, which are to be used in this work.

VDM RoAD fulfilled the requirements of the above three features. First, VDM RoAD allows a user to enter a roadway alignment using the IHSDM roadway file format. Second, VDM RoAD contains a comprehensive vehicle library, including the most common vehicles operating on North American highways. This library includes a variety of trucks, and a “real” passenger car. Finally, the output of the simulation can be viewed as an animation or a series of plots that are clearly presented and easy to read. Previous researchers such as Dabbour et al. (2004) and Easa and Dabbour (2004) have successfully used VDM RoAD in their work to simulate passenger car and truck dynamics on 2D and 3D combined alignments. It is important to note that inputting the roadway file is a tedious and time consuming task that requires the user to have some knowledge and background in the field of transportation and highway engineering.

3.3 INPUT DATA

Upon starting the VDM RoAD software, the user is prompted by the *Runs* screen (see Figure 3.2). The *Runs* screen is central to the operation of VDM RoAD, and provides users with access to vehicle data, input data, post-processing programs, and solver programs.

The *Runs* screen is divided into three main sections:

- The input area (the left hand side of the screen)
- The simulation processing area (middle or centre screen)
- The output area (right hand side of the screen)

The user works from the left to the right of the *Runs* screen to perform a simulation run and to view the results of the simulation. From the *Runs* screen, under the “Model Parameters & Inputs” section located to the left of the screen, the user can select a vehicle from the vehicle library, select the highway geometry, designate vehicle speed, and vehicle positioning relative to the centreline.

Figure 3.2: Screenshot of *Runs* Screen from VDM RoAD.

Various utility functions within the program can be accessed from the toolbar at the top of the *Runs* screen. These functions allow the user to quickly access the data libraries built into VDM RoAD. In the “Run Control” section of the *Runs* screen located at the centre of the screen, a user can designate the vehicle speed, the start and stop stations, and the computation parameters.

It is also from this section that a user can start the simulation by pressing the “Run Simulation” button. In the “Output & Post Processing” section of the *Runs* screen located to the right of the screen, a user can decide which output plots they wish to view, can select a camera angle or view, and can run a “real-time” animation of the simulation.

3.3.1 Vehicle Type

VDM RoAD contains a comprehensive vehicle library representing a number of different vehicles operating on North American highways. The type of vehicle and its characteristics (weight, number of axles, etc.) can be entered manually by the user or selected from the built-in library. To select the vehicle to be used for the simulation, the user selects the toggle button labelled “Vehicles:” under the “Model Parameters & Inputs” section of the *Runs* screen. From here, the user then selects “Go to Data Set” from the Pop-down window. The user is then prompted by the “Vehicle: *Vehicle Type*” window. From this screen the user can view or enter vehicle data concerning steering systems, front and rear suspensions, tires, inertial properties, as well as basic weights and vehicle dimensions. Figure 3.3 shows the main data screen for the WB-12 Combination truck.

Data set: **WB-12 Combination** ☐ Changed ID 38
 Category: ☐ Notes 8:01:46
☐ Locked 10/14/97

Weight Distribution: Axle Loads (kg)

Tractor axle 1	Tractor axle 2	Tractor axle 3	Trailer axle 1	Trailer axle 2
6000	7000		4000	4000

Tractor
 Tractor: 2A-Tractor
 WB-12 Tractor
 Reference Point
 Reference origin
 Options: generic
 (No data set selected)

Semitrailer
 Trailer: 2A-trailer
 WB-12 Trailer
 Reference Point
 Reference origin
 Cargo dimensions
 Big semi load
 Options:

Combination Vehicle
 Misc. (Custom) Vehicle Data
 Animator Group
 WB-12 Combo

Notes
 The field for axle 3 tractor MUST be blank or the program will crash.

Figure 3.3: Vehicle screen in VDM RoAD for WB-12 Combination truck.

From this screen (Fig. 3.3), the user can view additional data by clicking the toggle buttons for the defined parameter.

The VDM RoAD vehicle library contains a comprehensive list of vehicles that a user can use to run a simulation. The VDM RoAD software library contains the following truck components and combinations:

- AASHTO design trucks (SU, 2-axle tractor & 3-axle tractor)
- AASHTO design trailers (WB-12, WB-15, WB-19 & WB-20 trailers)
- Any user-defined combination among the listed tractors and trailers.
- AASHTO design combinations (WB-12, WB-15, WB-19 & WB-20 combinations).
- A heavy-truck combination (3a/2a) combo.

3.3.2 Roadway Geometry

The geometry and characteristics of a roadway are inputted into VDM RoAD as an IHSDM roadway file or interchange file format. The IHSDM input file is a text file that is constructed in such a way that each record or “row” in the file represents a critical point along the roadway alignment. A critical point is represented as any change in the vertical alignment, horizontal alignment, or cross section of the roadway. An IHSDM roadway file has 38 columns, each representing a variable that makes up the geometry of the roadway. VDM RoAD does not have the capability to prepare the IHSDM roadway file. Therefore, the user must manually prepare each alignment, and then input it into VDM RoAD. One of the major benefits of the IHSDM roadway file is the capability of inputting any type of 3D-roadway alignment that can then be used in VDM RoAD.

3.3.3 Driver Behaviour

Following the selection and input of the vehicle and geometric alignment data, the next step is to select the physical or tactile decisions that a driver would make while operating a vehicle. These driver decisions are mainly based on speed and steering controls (Sayers, 1999). Both the speed and steering controls can be made to vary as a function of roadway stationing, or to remain constant along the roadway throughout the length of the entire simulation. Lane change manoeuvre is another driver decision that can be inputted into VDM RoAD as a constant lateral position, a single lane change manoeuvre, or an irregular series of lane change manoeuvres (king-of-the-road manoeuvres).

VDM RoAD simulates the effect of a driver attempting to maintain a target speed through the use of a closed-loop speed controller that applies torque to the drive wheels as needed to follow a specified speed. The drive torque causes the wheel spin to increase, resulting in tire tractive force that acts to change the vehicle speed. VDM RoAD uses the speed and lateral positioning inputs and in turn produces the corresponding torque and steering wheel angles that are consistent with the “full dynamic response of the vehicle”. The controller algorithm within VDM RoAD then tries to maintain the speed and steering values that were inputted by the user. If the algorithm attempts to navigate an alignment using speeds and steering inputs that are beyond the capabilities of the roadway or the vehicle, then the vehicle crashes.

3.3.4 Cargo-Load Distribution

A major factor affecting the performance of a truck is the cargo load that the truck is transporting. Some of the important features of the cargo load are the dimensions and distributions, including the length, width, and height of the cargo. The amount or mass of cargo that can be transported is a function of the characteristics of the truck (size, number of trailers, etc.). Another important factor that Garcia et al. (2003) found is the height and lateral shift of the cargo mass centre is also a very important factor affecting truck stability. The higher and more eccentric the mass centre of the cargo load is, the lower the rollover speed of the truck is.

3.4 DATA PROCESSING AND OUTPUT

Once all the inputs have been defined and inputted into VDM RoAD by the user, the next step is the simulation run event where the simulation solver programs within VDM RoAD

are initiated to resolve the various differential equations that define vehicle motion and associated forces and reactions (Sayers, 1999). The solver program reads in the files that were predefined by the user in the input area. The data that is input is then used for the numerical integration that is required to solve the corresponding equations of motion.

After a user has conducted a simulation or run using VDM RoAD, the output and post-processing features in VDM RoAD allow the user to view the simulation run using a wire frame animator. The wire frame animator provides a “live” simulated run of the vehicle operating on the highway alignment, and allows the user to view the vehicle responses from a variety of angles (Fig. 3.4). The user can also view a variety of graphical plots such as tire

Figure 3.4: Wire frame animation in VDM RoAD

Figure 3.5: Sample of Output plots for a single run in VDM RoAD

and steering responses, vehicle motion and tracking, and select roadway geometry (Fig. 3.5).

The output plots and animations provide a useful alignment evaluation tool for highway roadway designers and engineers.

CHAPTER 4

EXPERIMENTAL DESIGN AND DATA PREPARATION

This chapter identifies the criteria and assumptions used to determine the variables used in the experimental work. As well, the test alignment configurations, design vehicles, cargo-load distribution, and other considerations are all highlighted in this chapter. For the experimental work, two main scenarios were analyzed. The two cases are:

- Analyzing the effect of placing an intermediate tangent between the two arcs of a flat reverse horizontal curve.
- Analyzing the effect of placing an intermediate tangent between the two arcs of a 3D reverse horizontal curve.

4.1 SELECTION OF VARIABLES

The main focus of this thesis is to evaluate and analyze the effect of placing an intermediate tangent between the two arcs of a reverse horizontal curve. To achieve this, VDM RoAD was used to evaluate truck stability on reverse curves. Furtado et al. (2002) and Easa and Dabbour (2003) evaluated passenger car and truck stability on other types of alignments such as simple and compound curves. The reverse curves are combined with different types of vertical alignments (upgrade, downgrade, crest curve and sag curves) as well as having an

intermediate tangent placed between the two arcs. Some of the input variables for the test alignment are the maximum superelevation rate, horizontal deflection angle, maximum vertical grades, minimum length of vertical curves, maximum rate-of-change of grade, and tangent length.

4.1.1 Minimum Radius and Superelevation

The superelevation of a roadway alignment is one of the input variables used in VDM RoAD, and is an important factor that affects vehicle stability. For this experiment, superelevation rates of $e_{\max} = 0.04$ and 0.06 were used. The justification for the use of these values lies within the fact that when traffic congestion or extensive marginal development acts to restrict top speeds, it is a common practice to utilize a low maximum rate of superelevation (AASHTO, 2001). As well, the Canadian minimum horizontal curve radius as defined by the TAC (1999) is based on a superelevation rate of 4 and 6 percent respectively. Another justification for using a maximum superelevation rate of 0.06 is that it results in better horizontal alignment in cases where minimum radii are used and improves operational characteristics for vehicles travelling at low speeds during adverse weather conditions. Since the horizontal curves are designed having a minimum radius, the curves required the superelevation rate to be at a maximum.

For the deflection angle of the horizontal curve, a value of 45° degrees was used throughout the simulation. The justification of the 45° degree deflection angle is that it results in a sufficiently long curve for maximum superelevation to be achieved along a portion of the curve.

4.1.2 Vertical Alignment

For vehicles operating on grades, the speed of a truck is affected more significantly than the speed for a passenger car. On level grades, trucks operate at speeds that are approximately the same as passenger vehicles. However, for a truck operating on a downgrade, truck speeds are about 5% higher than on level terrain. For upgrades, the grade length and the mass/power ratio of the vehicle influence truck speed. For the analysis, maximum grades of +6% and -6% are used since this grade is the controlling limit used on Canadian rural highways (TAC, 1999).

For vertical curves, the minimum length was selected as to satisfy the maximum rate-of-change of grade, r . The maximum rate-of-change of grade is constant for the length of curve as vertical curves are typically parabolic in nature. The inverse of r is the curve length corresponding to a change in grade of 1% and is denoted by K . For crest vertical curves, K was selected to satisfy the minimum stopping sight distance, while for sag curves, the distance of the headlight was the limiting factor.

4.1.3 Ratio Between Flatter and Sharper Radii for Reverse Curves

The ratio between the radius of the flatter (R_2) and sharper arc (R_1) composing a complex horizontal curve has a profound effect on truck stability. For compound curves, the AASHTO (2001) guidelines state that the ratio should not exceed 1.5. However, no design guidelines were provided for the design of reverse horizontal curves. For this analysis, the ratio was selected to range between 1.0 and 2.0. The minimum value of that ratio was

selected to be 1.0 as it produces the maximum effect of reverse curvature on vehicle stability (Dabbour, 2003).

4.1.4 Design Speed

The design speed (V_D) has one of the most profound effects on the geometric characteristics of a highway alignment. Most major geometric components such as minimum curve radii and superelevation rates take design speed into consideration. For the analysis, design speeds ranging from 40 km/h to 110 km/h were tested.

4.1.5 Selection of Design Vehicles

VDM RoAD has an extensive built-in vehicle library that contains the following truck components and combinations. For this study, the following design vehicles were selected:

- WB-15 from the AASHTO design guide as it represents a moderate sized truck that operates on North American roads.
- WB-20 from the AASHTO design guide as it represents one of the largest design trucks in the AASHTO design guides.
- The heavy truck combination (3a/2a) combo, as it is a good representation of a typical heavy truck operating on North American highways.

4.1.6 Cargo-Load Distribution

VDM RoAD allows the user to input user-defined cargo load dimensions and distribution. Some of the cargo dimensions are the length, width, and height of the cargo. Garcia et al.

(2003) found that the height and lateral shift of the mass centre are very important factors in truck stability. The higher and more eccentric the mass centre of the cargo is, the lower the speed for truck rollover to occur is.

4.1.7 Tangent Length

As previously mentioned in this chapter, the main goal of the experimental work is to analyze the effects of placing a tangent between the arcs of a reverse horizontal curve. To analyze this effect, it was first necessary to test the base case scenario of a reverse horizontal curve with no tangent (i.e. $T = 0$ m). To observe the effect of tangent length on vehicle stability, incremental values of tangent length were run until the effect on vehicle stability, and more specifically, the lateral acceleration was negligible. For the analysis, tangent lengths ranging from 0 m to 450 m were used depending on the superelevation rate and vertical alignment. This process was repeated for several types of vertical alignment.

4.2 LATERAL ACCELERATION AND SIDE FRICTION

To examine the effects of the test alignments on vehicle stability, it is necessary to examine the peak lateral acceleration values. From Figure 4.1, the rectilinear line represents the side friction obtained through manipulation of the point mass model based on driver comfort for specific values of R and V (assumed by the design guides). The scattered line represents the resulting lateral acceleration obtained based on the 3D vehicle dynamics models within VDM RoAD for the same R and V . For the Point Mass model, the radius of the curve, superelevation, and design speed are used to calculate the lateral acceleration.

Figure 4.1: Lateral acceleration calculated by VDM RoAD on 3-D alignment.

To estimate the actual lateral acceleration experienced by the vehicle, VDM RoAD uses the differential equations of motion along with the roadway geometry. The lateral acceleration values are read off the y-axis of the Ay vs. Station plot.

4.3 VDM RoAD AND HUMAN FACTORS

Human factors are not discretely addressed in VDM RoAD. However, some human factors are included to some extent. Human factors are represented in VDM RoAD as an input variable for lane change manoeuvres. The following types of manoeuvres are included in VDM RoAD: single lane change, or irregular series of lane change manoeuvres called king-of-the-road manoeuvres. By running trial simulations, it was found that the irregular series of lane change manoeuvres resulted in more lateral shift of the vehicle. As a result, this king-

of-the-road manoeuvre was used as the input variable for the analysis. Other complex human factor considerations such as braking as a reaction to increased lateral acceleration when it exceeds driver's comfort, driver fatigue, and steering reaction to increased roll angle are not included in VDM RoAD.

CHAPTER 5

DEVELOPMENT OF NEW GUIDELINES FOR REVERSE CURVES WITH INTERMEDIATE TANGENTS

This chapter examines vehicle stability and the effects of tangent length on the required minimum radius of reverse horizontal curves. As well, the testing methodology and simulation process is presented in this section. In today's highway geometric design policies (AASHTO, TAC), no clear design guidelines exist for reverse horizontal curvature. The current design guides provide limited insight into the design of compound horizontal curvature (e.g.: ratio between the radii of the flatter and sharper arcs not to exceed 1.5); however, no guidelines exist for the design of reverse curves. When designing a reverse curve, the current guidelines address each curve as an individual component, and do not take into account the effect of the other curve. As a result of this design approach, inconsistencies exist in the design of reverse horizontal curves. Work conducted by Dabbour et al. (2004) showed that the required minimum radius for a reverse curve is greater than what the AASHTO or TAC design guidelines specify.

One of the main objectives of this research is to analyze the benefits of placing an intermediate tangent between the two arcs of a reverse curve. To conduct this analysis, VDM RoAD was used throughout the experimental work. The variables that were previously described in Chapter 4 are used throughout the testing and simulation runs.

5.1 SIMULATION PROCEDURE

To investigate the effect of tangent length on reverse curvature and truck stability, VDM RoAD was used. The first analysis was conducted for reverse horizontal curves with no vertical alignment. The following procedure was used in conducting the experimental work for a given design speed, reverse curve ratio, design vehicle, and superelevation rate (Figure 5.1):

1. Input a flat alignment (horizontal curve with radius R_1) as a base case. This alignment includes a simple horizontal curve with superelevation.
2. Run simulation using design speed (V_D), and read lateral acceleration (L_1)
3. Introduce another horizontal curve in the opposite direction (having radius R_2) to construct a reverse curve composed of two horizontal curves in opposite directions having radii R_1 and R_2 .
4. Run simulation and read the lateral acceleration for simulation (L_2).
5. Increase the horizontal curve radius R_1 (R_2 increases for corresponding ratio) and re-read the lateral acceleration values (L_3). Continue to increase the radius (R_1) until the lateral acceleration L_3 is equal to the lateral acceleration obtained in (2), L_1 .
6. Record the final radius R_1 and compare to radius R_1 for simple horizontal curve in Step 1.
7. Introduce tangent of length T , between curves R_1 and R_2 , and return to Step 3.

For the testing procedure, simulations were run for several design speeds (V_D) ranging from 40 km/h to 110 km/h. Runs were made for alignments having maximum superelevation rates of $e_{max} = 0.04$ and $e_{max} = 0.06$. Three ratios (r) between the reverse curves R_1 and R_2 were also tested (1.0, 1.5, and 2.0). Three design vehicles (trucks) were used in the analysis. An intermediate tangent length ranging from 0 m to 450 m was tested. The trucks were run in the direction from the flatter to sharper arc as this was found to be the most critical case.

For the analysis of 3D reverse curves with intermediate tangents, the reverse curves were superimposed with crest and sag vertical curves, upgrades, and downgrades. A first and second grade ranging from -6% to $+6\%$ was tested. For flat (level) alignment, the values of the first and second grade are zero. For a downgrade, the signs of the first and second grade are “-“ and “-“. For an upgrade, the signs of the first and second grade are “+“ and “+“. For a crest vertical curve, the signs of the first and second grade are “+“ and “-“. For a sag vertical curve, the signs of the first and second grade are “-“ and “+“. A similar testing procedure (Figure 5.1) was carried out in VDM RoAD to develop the 3D Reverse Curve Model.

Table 5.1: Design Speeds and Minimum Design Radii used in Analysis (TAC, 199)

Design Speed, V_D (km/h)	Radius $e_{max} = 4\%$	Radius $e_{max} = 6\%$
40	60	55
50	100	90
60	150	130
70	200	190
80	280	250
90	380	340
100	490	440
110	680	600

Figure 5.1: Simulation procedure for vehicle stability

5.2 DEVELOPMENT OF NEW GUIDELINES

The design of reverse horizontal curves is not clearly addressed in the AASHTO (2001) and TAC (1999) geometric design guides. One of the main objectives of this research was to develop guidelines that could be used as a design aid by highway engineers and designers. Following the evaluation procedure outlined in Figure 5.1, the results for the required minimum radii and the required increase in curve radius R_I are presented in APPENDIX A (Tables A1 to A20) for flat and 3D alignments, for three values of ratio between R_I and R_2 , for different design speeds, for various lengths of intermediate tangent, for three design vehicles, and for $e_{max} = 0.04$ and $e_{max} = 0.06$ respectively. The values shown in Tables 5.2 and 5.3 are for flat and 3D alignments (6% downgrade), for the NHTSA 3a/2a truck, and for $e_{max} = 0.06$. The results of the analysis show that reverse curvature has a profound affect on vehicle stability.

The results show that an increase in the minimum radius is required to compensate for the effects of reverse curvature. The increase in minimum radius ranges from approximately 3% to 25% depending on the design vehicle, superelevation rate, and the ratio between R_I and R_2 . It was observed that a smaller increase in minimum radius is required when the ratio between the two curves increases. Also, the required increase in the minimum radius for $e_{max} = 0.06$ was found to be higher than the required increase for $e_{max} = 0.04$. It was also observed that a greater increase is required for higher design speeds. Furthermore, when an intermediate tangent is placed between the two curves, the required increase in minimum radius is significantly reduced. For lower design speeds, shorter lengths of tangent are needed to reduce the required increase in minimum radius caused by reverse curvature.

Table 5.2: Required increase in curve radius R_1 for flat reverse curves for different values of ratio, and intermediate tangent lengths (NHTSA 3a/2a, $e_{max} = 0.06$).

Design Speed (km/hr)	Design Minimum Radius (m)	R ₂ = 1.0 R ₁		R ₂ = 1.5 R ₁		R ₂ =2.0 R ₁	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
T = 0 m							
40	55	62	12.7	61	10.9	58	5.5
50	90	106	17.8	102	13.3	98	8.9
60	130	154	18.5	149	14.6	142	9.2
70	190	226	18.9	219	15.3	210	10.5
80	250	298	19.2	291	16.4	281	12.4
90	340	410	20.6	399	17.4	389	14.4
100	440	536	21.8	522	18.6	509	15.7
110	600	735	22.5	728	21.3	705	17.5
T = 50 m							
40	55	58	5.5	57	3.6	56	1.8
50	90	98	8.9	96	6.7	93	3.3
60	130	146	12.3	141	8.5	136	4.6
70	190	215	13.2	209	10.0	200	5.3
80	250	285	14.0	280	12.0	270	8.0
90	340	394	15.9	388	14.1	378	11.2
100	440	517	17.5	508	15.5	491	11.6
110	600	711	18.5	707	17.8	685	14.2
T = 100 m							
40	55	57	3.6	56	1.8	55	0.0
50	90	96	6.7	93	3.3	92	2.2
60	130	140	7.7	137	5.4	134	3.1
70	190	208	9.5	201	5.8	198	4.2
80	250	277	10.8	268	7.2	268	7.2
90	340	382	12.4	371	9.1	368	8.2
100	440	504	14.5	499	13.4	488	10.9
110	600	691	15.2	686	14.3	668	11.3
T = 200 m							
40	55	57	3.6	56	1.8	55	0.0
50	90	94	4.4	92	2.2	92	2.2
60	130	139	6.9	134	3.1	134	3.1
70	190	204	7.4	198	4.2	197	3.7
80	250	269	7.6	265	6.0	261	4.4
90	340	368	8.2	361	6.2	359	5.6
100	440	478	8.6	469	6.6	465	5.7
110	600	652	8.7	640	6.7	637	6.2

Table 5.3: Required increase in curve radius R_1 for 3D reverse curves for different values of ratio, and intermediate tangent lengths (NHTSA 3a/2a, $e_{max}=0.06$).

Design Speed (km/hr)	Design Minimum Radius (m)	$R_2 = 1.0 R_1$		$R_2 = 1.5 R_1$		$R_2 = 2.0 R_1$	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) $T = 0$							
40	55	67	21	65	18.2	62	12.7
50	90	109	21.1	107	18.9	103	14.4
60	130	159	22.3	159	22.3	149	14.6
70	190	234	23.2	228	20.0	221	16.3
80	250	316	26.4	304	21.6	292	16.8
90	340	435	27.9	420	23.5	402	18.2
100	440	557	26.6	547	24.3	523	18.9
110	600	761	26.8	754	25.7	723	20.5
(b) $T = 100$ m							
40	55	59	7.3	58	5.5	57	3.6
50	90	98	8.9	95	5.6	93	3.3
60	130	144	10.8	140	7.7	136	4.6
70	190	221	16.3	207	8.9	201	5.8
80	250	291	16.4	273	9.2	269	7.6
90	340	400	17.6	378	11.2	371	9.1
100	440	519	18.0	494	12.3	483	9.8
110	600	714	19.0	678	13.0	666	11.0
(c) $T = 200$ m							
40	55	58	5.5	57	3.6	56	1.8
50	90	96	6.7	94	4.4	92	2.2
60	130	141	8.5	136	4.6	133	2.3
70	190	211	11.1	201	5.8	197	3.7
80	250	280	12.0	270	8.0	264	5.6
90	340	383	12.6	370	8.8	365	7.4
100	440	498	13.2	481	9.3	472	7.3
110	600	691	15.2	659	9.8	651	8.5
(d) $T = 400$ m							
40	55	55	0.0	55	0.0	55	0.0
50	90	90	0.0	90	0.0	90	0.0
60	130	130	0.0	130	0.0	130	0.0
70	190	190	0.0	190	0.0	190	0.0
80	250	250	0.0	250	0.0	250	0.0
90	340	342	0.6	341	0.3	340	0.0
100	440	445	1.1	443	0.7	440	0.0

For higher design speeds, longer lengths of tangent are required to reduce the increase caused by reverse curvature. The tables developed in this chapter (APPENDIX A) can be used as a design aid for determining the minimum radius requirements for reverse horizontal curves with an intermediate tangent.

5.3 SIDE FRICTION VARIATION

The peak lateral accelerations for the reverse curve with intermediate tangent were recorded for each design vehicle on all the test alignments as shown in Figure 5.2. The rectilinear line represents the side friction obtained from the manipulation of the point-mass model based on driver comfort for a given radius (R), design speed (V_D), and maximum rate of superelevation (e_{max}). The scattered line represents the resulting lateral acceleration obtained based on the 3D vehicle dynamics model within VDM RoAD for the same values of R , V , and e_{max} . VDM RoAD uses the geometry of the test alignment and differential equations of motion to estimate the actual lateral acceleration experienced by the vehicle (Sayers, 1999).

Figure 5.2: Plot of Actual and Assumed Lateral Acceleration for WB15 truck on a Reverse Curve with Intermediate Tangent.

As the truck navigates through the test alignment, both the tractor and trailer experience deviations from the assumed lateral accelerations. The deviation in lateral acceleration for the trailer is generally higher than that experienced by the tractor. The trailer is connected to the tractor through a hitch mechanism, which is a single point and can be represented as a ballpoint. The forces experienced by the tractor (roll, tractive forces, etc.) are transferred to the trailer through the hitch. This may explain the increased levels of lateral acceleration experienced by the trailer.

For the simulation runs, three trucks were tested (WB20, WB15, and NHTSA 3a/2a). The actual peak lateral acceleration experienced by the WB20 design vehicle was higher than the peak values experienced by the WB15 and NHTSA 3a/2a trucks for the same highway alignments. The WB15 truck experienced lower levels of peak lateral acceleration than the WB20 and NHTSA 3a/2a trucks. This observation made sense since the WB20 truck is one of the larger design vehicles in the design guide and WB15 is considered a moderate sized design truck. Generally, the trucks experienced increases in lateral acceleration over a large portion of the first and second arc of the reverse curve with the peak values occurring upon entrance of the second arc of the reverse curve. Some deviations in lateral acceleration were observed on the intermediate tangent generally occurring at the start and end of the tangent.

5.4 EFFECT OF TANGENT LENGTH ON REVERSE CURVATURE

Reverse curvature has a profound effect on vehicle stability. Drivers operating on horizontal curves require more attention and effort in operating and guiding their vehicle along the curve as compared to operating a vehicle on a straight section of road. When reverse curves are used in highway design, the effect on driver demand is magnified even greater

(Wooldridge et al., 2000). As well, the vehicle experiences greater variations in lateral acceleration while operating on complex reverse curves than compared with simple horizontal curves. To overcome the required increase (3% to 25%) in curve radius (R_1) that is directly attributed to reverse curvature, and to provide more stability for vehicles operating on reverse curves, the effect of adding an intermediate tangent between curve radii R_1 and R_2 was evaluated. It was observed that by inserting an intermediate tangent length between the two arcs of the reverse curve, the resulting required increase in curve radii R_1 and R_2 is reduced. As the tangent length, T , increases, the required increase in curve radii R_1 and R_2 continues to decrease. Presented in Figure 5.3 is a plot representing the minimum required radius vs. the design speed for a simple horizontal curve, a reverse curve, and two reverse curves with an intermediate tangent.

Figure 5.3: Increase in Required Minimum Radius of Flat Reverse Curves for different intermediate tangent lengths (NHTSA 3a/2a, $e_{max} = 0.04$, and Ratio = 1.0).

For the simple horizontal curve, the minimum radius is the lowest value at each design speed. For the reverse curve, the required minimum radius is greater than that of the simple horizontal curve, and is the highest value at each specified design speed. This increase in the minimum radius is directly attributed to reverse curvature. For the reverse curves having an intermediate tangent, the increase in the required minimum radius is not as severe as the increase for the reverse curve with no tangent. As the tangent length is increased, the required increase in the minimum radius decreases, and eventually approaches the minimum radius of the simple horizontal curve.

An intermediate tangent placed between a reverse horizontal curve helps improve vehicle stability. The tangent length provides a section of alignment where a vehicle can recover from the higher lateral acceleration levels experienced on the horizontal curves. One of the major benefits in using an intermediate tangent between the two arcs of a reverse curve is that it can be used in situations where a highway engineer or designer may face constraints. These constraints may be environmental, economic, physical, topographical, or any other obstacle or limitation that may affect the design or construction of a highway alignment.

In situations where the minimum radius of a reverse curve is constrained or limited to a specified value, a tangent length placed between the two arcs may provide solutions to the constraint. By using a tangent, the required increase in the minimum radius is reduced for increasing lengths of tangent. Therefore, the reverse curve can be designed having smaller radii (R_1 and R_2), and the vehicle stability requirements are still addressed.

5.5 EFFECT OF VERTICAL ALIGNMENT ON REVERSE CURVES

A similar procedure was used to investigate the effects of tangent length and vertical alignment on the required radius of a reverse curve. In order to quantify the effect of vertical alignment, the original flat alignments that were used to develop the flat reverse curve guidelines were superimposed by several different types of vertical alignments (e.g.: upgrade, downgrade, crest curve, and sag curve).

To investigate the effect of tangent length on reverse curvature with vertical alignment, VDM RoAD was used. Several different combinations of 3D alignments were tested: downgrade, upgrade, sag curve and crest curve. On completion of the simulations, it was found that the most critical cases occur when the horizontal alignment is superimposed by either a downgrade or sag curve. The results for the required minimum radii and the required increase in curve radius R_I for several different lengths of tangent are presented in Appendix A (Tables A13 to Table A20) for the most critical case.

As was observed for the cases having only horizontal alignment (flat), the results illustrate that an increase in the minimum radius is required to compensate for the effect of reverse curvature being superimposed with vertical alignment. When an intermediate tangent was inserted between the two arcs (R_1 and R_2), the required increase in radius decreased for increasing lengths of tangent (T). This increase in the required radius ranged from approximately 5% to 29% for reverse curves superimposed with vertical alignment. This increase was greater than the required increase for reverse curves on flat terrain.

It was observed that vertical alignment has an effect on the required increase in minimum radius. When vertical alignment is combined with reverse curvature, there is an increase in the required minimum radius. However, the effect of reverse curvature alone on the required increase in minimum radius is more profound than the effect of vertical alignment. This observation is also supported by the work of Easa and Dabbour (2003) who found a similar result for simple, reverse, and compound horizontal curves. This is clearly presented graphically in Figure 5.4, which contrasts the required increase in minimum radius for reverse curvature alone, and reverse curvature with vertical alignment (NHTSA 3a/2a, $e_{max} = 0.06$)

Figure 5.4: Comparison of Effect of Reverse Curvature and Vertical Alignment on the Required Increase in Minimum Radius for Design Vehicle WB20, $e_{max} = 0.06$ (worst case vertical alignment).

5.6 MODEL DEVELOPMENT

In this section, the minimum radius requirements for flat and 3D reverse horizontal curves are developed. The two models were calibrated using the simulation results from VDM RoAD.

5.6.1 Development of Flat Reverse Curve Model with Intermediate Tangent

To determine the minimum radius requirements for flat reverse horizontal curves with an intermediate tangent, a mathematical model was developed. The model was calibrated using the simulation results obtained from VDM RoAD. The required minimum radius, R_{req} , was set as the dependent variable. The design speed, tangent length, superelevation rate, vehicle type, and ratio between curve radii were set as the independent variables. The final model was selected based on the following criteria:

1. The model must be highly correlated. The required coefficient of determination, R^2 , must be high.
2. Each independent variable used in the model must have a coefficient that is significantly different from zero at the 95% confidence level.
3. The algebraic signs of the coefficients of the independent variables must make logical sense.
4. All statistical multiple regression tests (e.g., linearity, homoscedasticity, normality, and collinearity, etc.) must be valid.

To develop the final Flat Reverse Curve Model with intermediate tangents, over 1200 observations were used to develop the multiple regression model. A standard multiple regression was performed between the required radius (R_l) as the dependent variable and design speed, tangent length, superelevation, ratio, and vehicle type as the independent variables. The statistical analysis was carried out using SPSS Regression and SPSS Frequencies for evaluation of the statistical assumptions. The final Flat Reverse Curve Model has six independent variables. The relevant statistical output for the model is presented in Tables B1 to B4, and Figures B1 and B2 of APPENDIX B. The final model is given by:

$$R_{req} = 0.099 (V_D - 30)^2 - 0.130T - 24.3I_e - 11.26r + 6.144WB20 - 7.484WB15 + 93.953, \quad (5.1)$$

($R^2=0.986$)

where,

R_{req} = Required Radius (m)

V_D = Design Speed (km/h)

T = Tangent Length (m)

I_e = Dummy variable for maximum superelevation
($I_e = 0$ if $e_{max} = 4\%$, and $I_e = 1$, if $e_{max} = 6\%$)

r = Ratio between Curve R_1 (flatter arc) and Curve R_2 (sharper arc)

$WB15$ = Design Vehicle is $WB15$ (1) and otherwise 0.

$WB20$ = Design Vehicle is $WB20$ (1) and otherwise 0.

(if design vehicle is $NHTSA\ 3a/2a$, $WB15 = WB20 = 0$)

One of the major requirements for the model was that the algebraic signs of the coefficients of the independent variables must make logical sense. The Flat Reverse Curve Model with intermediate tangents has logical explanation for the effect of each independent variable on

the dependent variable required radius. The positive sign for the coefficient V_D indicates that R_{req} increases with an increase in V_D . This is obvious as the radius of a horizontal curve is a function of design speed and increases as the speed is increased. The negative sign for the coefficient T indicates that R_{req} decreases for an increase in the length of intermediate tangent T . An intermediate tangent placed between the arcs of a reverse curve provides improved vehicle stability, allowing a vehicle to recover from the roll, moments, tractive forces, and increased levels of lateral acceleration experienced on the horizontal curve. The negative sign for the coefficient I_e indicates that R_{req} decreases as I_e increases from 0.04 to 0.06. The negative sign for the coefficient r indicates that R_{req} decreases with an increase in the ratio between curve radii R_1 and R_2 . The smaller the ratio, the more increase required for the minimum radius. The negative sign for the coefficient WB15 indicates that R_{req} decreases when the design vehicle is WB15 and the positive sign for the coefficient WB20 indicates that R_{req} increases when the design vehicle is WB20. This makes logical sense since the WB20 design vehicle is larger than the WB15 design vehicle and experiences higher levels of peak lateral acceleration.

1. For an increase in V_D by one km/h, there is a resulting increase in the required radius by 0.099 m when controlling for all other IVs.
2. For an increase in T by one meter, there is a resulting decrease in the required radius by -0.130 m when controlling for all other IVs.
3. For an increase in I_e from 0.04 to 0.06, there is a resulting decrease in the required radius by -24.3 m when controlling for all other IVs.

4. For an increase in the ratio (r) between R_1 and R_2 , there is a resulting decrease in the required radius by -11.26 m when controlling for all other IVs.
5. When the design vehicle is the WB15 truck, there is a resulting decrease in the required radius by -7.484 m when controlling for all other IVs.
6. When the design vehicle is the WB20 truck, there is a resulting increase in the Required Radius by 6.144 m when controlling for all other IVs.

By examining Table B4 in APPENDIX B, it is observed that all of the independent variables have significance of 0.000, which is less than the control value of 0.05. Thus, all the independent variables were significantly different from zero at the 95% confidence interval. The R^2 value of the model was found to be 0.986 (Table B2, APPENDIX B) indicating that a very strong relationship exists between the dependent variable and the independent variables. It is important to note that the Flat Reverse Curve Model is only valid for the range of variables tested. For example, using a design speed that is lower or higher than the tested range (40 km/h to 110 km/h) may not produce reasonable results.

5.6.2 Development of 3D Reverse Curve Model with Intermediate Tangent

To determine the minimum radius requirements for reverse horizontal curves with an intermediate tangent combined with vertical alignment (3D), a mathematical model was developed. The model was calibrated using the simulation results obtained from VDM RoAD. The required minimum radius, R_{req} , was set as the dependent variable. The design speed, tangent length, superelevation rate, vehicle type, first and second grades, and ratio

between curve radii were set as the independent variables. The final model was selected based on the following four criteria:

1. The model must be highly correlated. The required coefficient of determination, R^2 , must be high.
2. Each independent variable used in the model must have a coefficient that is significantly different from zero at the 95% confidence level.
3. The algebraic signs of the coefficients of the independent variables must make logical sense.
4. All statistical multiple regression tests (i.e. linearity, homoscedasticity, normality, and collinearity, etc.) must be valid.

To develop the final 3D Reverse Curve Model with intermediate tangents, over 5000 observations and approximately 300 different test alignments were used to develop the multiple regression model. A standard multiple regression was performed between the required radius (R_{req}) as the dependent variable and design speed, tangent length, maximum rate of superelevation, ratio, first and second grades, and vehicle type as the independent variables. Analysis was performed using SPSS Regression and SPSS Frequencies for evaluation of the statistical assumptions. The final 3D Reverse Curve Model with intermediate tangents has eight independent variables. The relevant statistical output for the model is presented in Tables B5 to B7, and Figures B3 and B4 of APPENDIX B. The final model is given by:

$$R_{req} = 0.0985 (V_D - 30)^2 - 0.137T - 13.306I_e - 14.514r - 0.773G_1 - 0.484G_2 - 7.283WB15 + 2.789WB20 + 154.863 \quad (5.2)$$

$(R^2 = 0.983)$

where,

R_{req} = Required Radius (m)

V_D = Design Speed (km/h)

T = Tangent Length (m)

I_e = Dummy variable for maximum superelevation

($I_e = 0$ if $e_{max} = 4\%$, and $I_e = 1$, if $e_{max} = 6\%$)

r = Ratio between Curve R_1 (flatter arc) and Curve R_2 (sharper arc)

G_1 = First Grade

G_2 = Second Grade

$WB15$ = Design Vehicle is $WB15$ (1) and otherwise 0.

$WB20$ = Design Vehicle is $WB20$ (1) and otherwise 0.

(if design vehicle is $NHTSA\ 3a/2a$, $WB15 = WB20 = 0$)

One of the major criteria for the model was that the algebraic signs of the coefficients of the independent variables must make logical sense. The 3D Reverse Curve Model with intermediate tangents has logical explanation for the effect of each independent variable on the dependent variable required radius. The positive sign for the coefficient V_D indicates that R_{req} increases with an increase in V_D . This is obvious as the radius of a horizontal curve is a function of design speed and increases as the speed is increased. The negative sign for the coefficient T indicates that R_{req} decreases for an increase in the length of intermediate tangent T . An intermediate tangent length placed between the arcs of a reverse curve provides improved vehicle stability, allowing a vehicle to recover from the roll, moments, tractive forces, and increased levels of lateral acceleration experienced on the horizontal curve. The negative sign for the coefficient I_e indicates that R_{req} decreases as I_e increases from 0.04 to 0.06. The negative sign for the coefficient r indicates that R_{req} decreases with an increase in the ratio between curve radii R_1 and R_2 . The smaller the ratio, the more increase required for

the minimum radius. The negative signs for the first and second grades (G_1 and G_2) indicate that a decrease in R_{req} occurs when the grades are increased. The most critical case occurred when reverse curves were superimposed with downgrades. By substituting a -6% downgrade into the model, an increase in R_{req} occurs. The negative sign for the coefficient WB15 indicates that R_{req} decreases when the design vehicle is WB15 and the positive sign for the coefficient WB20 indicates that R_{req} increases when the design vehicle is WB20. This makes logical sense since the WB20 design vehicle is larger than the WB15 design vehicle and experiences higher levels of lateral acceleration. In addition, we can deduce the following from interpreting the regression coefficients:

1. For an increase in V_D by one km/h, there is a resulting increase in the required radius by 0.0985 m when controlling for all other IVs.
2. For an increase in T by one meter, there is a resulting decrease in the required radius by -0.137 m when controlling for all other IVs.
3. For an increase in maximum superelevation rate (I_e) from 4% to 6%, there is a resulting decrease in the required radius by -13.306 when controlling for all other IVs.
4. For an increase in the ratio (r) between R_1 and R_2 , there is a resulting decrease in the required radius by -14.514 m when controlling for all other IVs.
5. For an increase in the first grade G_1 by 1%, there is a resulting decrease in the required radius by 0.773 m when controlling for all other IVs.
6. For an increase in the second grade G_2 by 1%, there is a resulting decrease in the required radius by -0.484 m when controlling for all other IVs.

7. When the design vehicle is the WB20 truck, there is a resulting increase in the required radius by 2.789 m when controlling for all other IVs.
8. When the design vehicle is the WB15 truck, there is a resulting decrease in the required radius by -7.283 m when controlling for all other IVs.

By examining Table B7 in APPENDIX B, it is observed that all of the independent variables have significance of 0.000, which is less than the control value of 0.05. Thus, all the independent variables were significantly different from zero at the 95% confidence interval. The coefficient of determination of the model is 0.983 (Table B6, APPENDIX B) indicating that a very strong relationship exists between the dependent and independent variables. It is important to note that the 3D Reverse Curve Model is only valid for the range of variables tested

5.7 SENSITIVITY ANALYSIS

In order to establish a model with logical explanation for all the variables, some sensitivity analysis should be conducted to determine the sensitivity of the proposed model towards the different variables.

5.7.1 Effect of Design Speed (V_D)

Design speed is one of the most critical factors affecting vehicle stability, and is a major design component of horizontal curves. From Figure 5.5, it is observed that as the design speed increases, the increase in the required minimum radius increases as well (as expected). Also, a slight elbow in the trends is observed for shorter lengths of tangent (< 25 m). This

could occur since for short tangent lengths, the vehicle does not have sufficient time to recover from the shift in load upon exiting the first curve. As previously noted, although the increase in the minimum required radius is less severe at lower design speeds, problems may still arise. This is of concern as reverse curves are generally used in highway interchanges (on and off ramps), and are normally designed having a lower design speed. It is important to note, that for lower design speeds, rollover speeds are closer to design speeds with smaller margins of safety (Dabbour and Easa, 2003). Thus a vehicle operating slightly over the design speed of a smaller radius curve (low design speed) may experience rollover, or vehicle instability.

Figure 5.5: Effect of Design Speed on Increase in Required Minimum Radius for design vehicle NHTSA 3a/2a, $e_{\max} = 0.04$, and Ratio = 1.0 for level terrain.

5.7.2 Effect of Tangent Length (T)

As can be observed in Figure 5.6 and 5.7, the required increase in minimum radius was found to decrease for increasing lengths of tangent. When the tangent length is equal to zero, the required increase in minimum radius is the greatest. As the tangent length is increased, the required increase in minimum radius continues to decrease until it approaches zero. This trend was observed for all flat and 3D reverse curves, for the three trucks, different design speeds, ratio, and maximum rates of superelevation.

For lower design speeds and $e_{max} = 0.04$, approximately 200 meters of tangent length eliminates the required increase in minimum radius caused by reverse curvature. For higher design speeds (> 70 km/h), 200 meters of tangent length or more is needed to eliminate the required increase in minimum radius caused by reverse curvature. When the maximum rate of superelevation is 0.06, longer lengths of tangent were required to eliminate the required increase in minimum radius caused by reverse curvature. For 3D reverse curves, longer lengths of tangent were required to reduce the increase in minimum radius caused by reverse curvature and vertical alignment.

The required radius of a reverse curve with various lengths of intermediate tangent (0 m to 200 m) is compared with the TAC recommended minimum radius in Figures 5.8 to 5.10 for the NHTSA 3a/2a truck, for $e_{max} = 0.04$ and for level terrain. The heavy solid black line represents the TAC recommended minimum radius. For a reverse curve with no intermediate tangent, it is observed that the required radius is the greatest for each design speed. For increasing lengths of intermediate tangent, the required radius of the reverse curve decreases, and approaches the level of the TAC recommended minimum radius.

Figure 5.6: Effect of Tangent length on the Required Radius for WB20, $e_{max} = 0.04$, $V_D = 80$ km/h, and for different values of ratio for level terrain.

Figure 5.7: Effect of Tangent Length on the Required Radius for WB15, $e_{max} = 0.04$, $V_D = 90$ km/h, and for different values of ratio for level terrain.

Figure 5.8: Minimum Radius of Flat Reverse Horizontal Curves for NHTSA 3a/2a, $e_{max} = 0.04$, and Ratio = 1.0.

Figure 5.9: Minimum Radius of Flat Reverse Horizontal Curves for NHTSA 3a/2a, $e_{max} = 0.04$, and Ratio = 1.5.

Figure 5.10: Minimum Radius of Flat Reverse Horizontal Curves for NHTSA 3a/2a, $e_{max} = 0.04$, and Ratio = 2.0.

5.7.3 Effect of Superelevation (e_{max})

The required increase for $e_{max} = 0.06$ was found to be generally higher than the required increase for $e_{max} = 0.04$ (Figure 5.11). It is also important to note that the variation in load distribution on different tires increases with the increase in superelevation, where the load increases on the inner tires and decreases on the outer tires. The variation in load distribution on the tires results in more variation in the side friction demand at various tires of the truck.

Figure 5.11: Effect of Superelevation on Increase in Required Minimum Radius for design vehicle NHTSA 3A/2A, $V_D = 60\text{km/h}$, $Ratio = 1.0$ for level terrain.

5.7.4 Effect of Design Vehicle (WB15, WB20, NHTSA 3a/2a)

The required increase in minimum curve radii for different design vehicles can be observed in Figure 5.12 for $V_D = 40\text{ km/h}$, $e_{max} = 0.04$, $ratio = 1.0$, and for flat terrain. The largest increase in the required minimum radius was found for the WB20 design vehicle. The WB20 truck is one of the largest design vehicles in the AASHTO guide, and therefore, is expected to have the largest variation in tire forces and reactions between the front and rear tires, as well as between the inside and outside tires. The required increase for the NHTSA 3a/2a truck was found to lie somewhere between the WB15 and WB20 design trucks.

Figure 5.12: Effect of Design Vehicle on Increase in Required Minimum Radius for design vehicles WB15, WB20 and NHTSA 3a/2a for $V_D = 40$ km/h, $e_{max} = 0.04$, Ratio = 1.0, and for flat terrain.

It is also important to note that the WB15 design truck is a smaller or moderate sized truck and the NHTSA 3a/2a heavy combination is measured from actual field data, so that it reflects a typical truck operating on North American highways.

5.7.5 Effect of Ratio Between Curve Radii R_1 and R_2

The effect of the ratio between the flatter and sharper arcs of the reverse curve on the required increase in minimum curve radius is presented in Figure 5.13 and Appendix A. The effect of the ratio between curve radii R_1 and R_2 was found to have an effect on the required minimum radius. The smaller the ratio between the two curves, the larger the required increase in the minimum curve radius. Since the second arc of the reverse curve is a function of the ratio (i.e. $R_2 = r R_1$), as the ratio increases, so does the radius of the second curve.

Figure 5.13: Effect of ratio between Flatter and Sharper arcs on Increase in Required Minimum Radius for design vehicle WB20, $e_{max} = 0.04$, $V_D = 40$ km/h, and for level terrain.

When a vehicle is operating on a reverse horizontal curve for a given design speed, and the curve radius is significantly larger than the minimum recommended radius, the vehicle is able to operate with more stability as a result of the lowered peak lateral acceleration.

5.7.6 Effect of First and Second Grade (G_1 and G_2)

For flat (level) alignments, the values of the first and second grade are zero. For a downgrade, the signs of the first and second grade are “-” and “-”. For an upgrade, the signs of the first and second grade are “+” and “+”. For a crest vertical curve, the signs of the first and second grade are “+” and “-”. For a sag vertical curve, the signs of the first and second grade are “-” and “+”. It was observed, that for instances when reverse curve alignments were combined with a downgrade or a sag vertical curve, the required increase in the

minimum radius is the most severe. Upgrades and crest vertical curves contributed to an increase in the minimum required radius, however, this increase was not as severe as the increase required for downgrades.

5.8 GUIDELINES FOR SPIRAL CURVES

After developing the design guidelines for reverse horizontal curves with intermediate tangents, guidelines were developed for the new minimum spiral transition curve lengths for the new reverse curves. Spiral (transition) curves are used to provide smooth transition and a natural driving path between a tangent and a circular curve. The recommended spiral transition lengths for reverse curves with an intermediate tangent and for various design speeds, V_D , *Ratio* 1.5, and design Vehicle NHTSA 3a/2a are presented in Tables 5.4 and 5.5. The transition curve length can be calculated from the following equation:

$$L = \frac{A^2}{R} \quad (5.3)$$

where L is the length of the transition curve, A is the spiral parameter in meters (TAC), and R is the curve radius. The required transition length is observed to increase with increasing design speed. From Figure 5.14 and 5.15, the thick solid line represents the TAC transition curve lengths. The thin lines beneath the TAC line are the transition curve lengths corresponding to different lengths of tangent. The new reverse curves developed in this chapter require a shorter transition curve length than the traditional AASHTO and TAC guidelines as a result of the increase in the minimum radius. A slight dip in the trend is observed at $VD = 80$ km/h. This is not a concern since three methods are used to determine transition curve lengths and the most conservative value is always selected.

Table 5.4: Minimum Lengths of Spiral Transition Curve For 3D Reverse Curves with Intermediate Tangents (NHTSA 3a/2a, Ratio = 1.5, $e_{\max} = 0.06$).

V_D (km/h)	TAC Spiral Length (m)	Tangent Length, T (m)						
		0	5	10	50	100	200	250
40	76	64	66	68	69	72	73	76
50	78	66	68	70	71	74	75	77
60	93	76	80	81	84	86	89	91
70	106	88	91	92	96	97	100	103
80	104	86	88	89	92	95	96	100
90	125	102	104	106	109	113	115	121
100	137	110	111	115	118	122	125	131
110	134	107	109	111	114	119	122	126

Table 5.5: Minimum Lengths of Spiral Transition Curve For Flat Reverse Curves with Intermediate Tangents (NHTSA 3a/2a, Ratio = 1.5, $e_{\max} = 0.06$).

V_D (km/h)	TAC Spiral Length (m)	Tangent Length, T (m)						
		0	5	10	50	100	200	250
40	76	68	69	72	73	74	74	76
50	78	69	70	72	73	76	77	77
60	93	81	82	83	85	88	90	91
70	106	92	93	94	96	100	102	102
80	104	89	90	91	93	97	98	99
90	125	107	108	109	110	115	118	119
100	137	115	116	117	118	121	128	129
110	134	111	111	112	114	118	126	128

Figure 5.14: Minimum Spiral Lengths Based on TAC Spiral Parameter for $e_{\max} = 0.06$, NHTSA 3a/2a, Ratio = 1.5 (Flat alignment) .

Figure 5.15: Minimum Spiral Lengths Based on TAC Spiral Parameter for $e_{\max} = 0.06$, NHTSA 3a/2a, Ratio = 1.5 (3D alignment).

The TAC transition curves are solely a function of the design speed and curve radius, while the new and improved reverse curves have the spiral length as a function of the tangent length.

CHAPTER 6

COST EFFECTIVENESS ANALYSIS

The results of the simulation and analysis portion of Chapter 5 demonstrate that an increase in the minimum required radius of reverse horizontal curves is needed to meet vehicle stability requirements. To examine the economic effectiveness of the new guidelines for the minimum radius of reverse horizontal curves developed in Chapter 5, it is necessary to contrast the costs and benefits of constructing curves based on the new guidelines with those of existing TAC (1999) guidelines. In this Chapter, the benefits, costs, and feasibility of implementing the new design will be evaluated. The evaluation will be conducted first for simple, followed by reverse horizontal curves.

6.1 GEOMETRY OF EXISTING AND NEW HORIZONTAL CURVE

As mentioned previously, the proposed new minimum radius is flatter (larger) than the existing TAC radius for given geometric conditions. The new flatter curves would be associated with lower collision rates and improve the level of safety of a highway alignment. Therefore, the new guidelines would improve the safety of a highway alignment. The new guidelines may be viewed as “*Curve Flattening*” with respect to the existing TAC guidelines. The new horizontal curve is less sharp, longer, and has a lower degree of curvature. Although the length of the new curve (L_n) is longer than the TAC curve (L_o), the overall length of the new alignment is reduced. This reduced alignment length produces several

benefits and cost savings. Figure 6.1 shows the effect of increasing the radius of a simple horizontal curve. From Figure 6.1, it is observed that by increasing the radius (R_1) of the sharper curve to that of the new curve having a radius (R_2), the resulting length (L_2) becomes longer than the TAC curve. However, the degree of curvature DC_2 of the new curve is less than the degree of curvature of the TAC curve, DC_1 . It is important to note that the two curves share common tangents, and that the central angle of the TAC curve is equal to the central angle of the new curve. As the degree of curvature DC is reduced, the collision rate decreases, and the level of safety of the alignment is improved (Lamm et al., 1999).

Figure 6.1: Geometry and Effect of Curve Flattening

When the radius of a sharp horizontal curve is increased, curve length increases as the following equation indicates:

$$L_2 = \frac{DC_1}{DC_2} L_1 \quad (6.1(a))$$

$$DC = \frac{5729.6}{R} \quad (6.1(b))$$

where L_1 = the length of the TAC (sharper) curve (m)

L_2 = length of the new curve (m)

DC_1 = the degree of curve of the TAC curve, °/100 m

DC_2 = the degree of curve of the new curve, °/100 m

Despite the increase in curve length, the total length of the alignment for the TAC curve decreases since a portion of the tangent section on either end of the TAC curve becomes circular. Assuming that the new horizontal curve will have the same tangents as the TAC curve, the reduced tangent length can be derived as:

$$T = \frac{5729.6}{1000} \left(\frac{1}{DC_2} - \frac{1}{DC_1} \right) \tan \left(\frac{\gamma}{2} \right) \quad (6.2)$$

where T = tangent length eliminated due to curve flattening (km), and γ = central angle (°).

Therefore, the length of the TAC alignment, L_o (km) is:

$$L_o = L_1 + 2T \quad (6.3)$$

and the length of the new alignment, L_n (km) is

$$L_n = L_2 \quad (6.4)$$

When the radius of a horizontal curve is increased, the total length of the alignment is reduced by an amount ΔL , given by:

$$\Delta L = \left[11.4592 \tan\left(\frac{\gamma}{2}\right) - 0.10\gamma \right] * \left(\frac{1}{DC_2} - \frac{1}{DC_1} \right) \quad (6.5)$$

where ΔL = the change in length (km), and γ = the central angle ($^\circ$). This decreased alignment length results in lowered construction costs, collision savings, maintenance and rehabilitation savings, travel-time savings, winter maintenance savings, and vehicle operating savings (Talarico and Morrall, 1993).

The effects and resulting benefits of increasing the radius of a sharp horizontal curve can be applied the same way to reverse horizontal curves. Since reverse curves are composed of two simple horizontal curves, the benefits of increasing the radii of a reverse horizontal curve will be two-times greater than a similar simple horizontal curve. The results of this research indicate that there is a need for an increase in the minimum required radius due to the effects of reverse curvature and vertical alignment. The reverse curve models developed in Chapter 5 that takes into consideration tangent length, vertical alignment, and vehicle stability produce larger radii reverse horizontal curves than the AASHTO and TAC designed curves. The next section will outline the various costs, benefits, savings models and concepts that were used in this analysis.

6.2 ANALYSIS METHODOLOGY

The methodology used in this chapter for evaluating the cost-effectiveness of the new guidelines is similar to that used by Talarico and Morrall (1993) who conducted a comprehensive study investigating the cost-effectiveness of curve flattening in Alberta. Their methodology and approach for investigating the cost-effectiveness of curve flattening has been applied in this analysis. Some of the parameters used in their methodology have been revised to reflect Ontario conditions.

6.2.1 Construction Cost and Savings

When the radius of a horizontal curve is increased, the total length of an alignment is decreased by an amount ΔL given by (Eq. 6.5). This decreased alignment length can be considered a construction saving, as it produces a shorter roadway alignment. Assuming the estimated costs of a 9.0 m and 11.0 m wide highway in Ontario is \$720,000/km and \$880,000/km (MTO, 2003 dollars) respectively, any reduction in alignment length can be considered a benefit. The cost of increasing the radius of an existing simple or reverse horizontal curve can be estimated from the following equations:

$$C_s = \frac{R\Delta\Pi}{180} * U \quad (6.6)$$

$$C_R = 2 * \frac{R\Delta\Pi}{180} * U \quad (6.7)$$

where C is the construction cost (\$), R is the curve radius (m), Δ is the central angle ($^\circ$), and U is the unit cost (\$/km). The construction savings attributed to constructing a larger radius horizontal curve can be calculated from:

$$C = \Delta L * U \quad (6.8)$$

where C is the cost savings (\$), ΔL is calculated from (Eq. 6.5) and U is the unit cost (\$/km). It is important to note that two major costs are associated with increasing the length of an existing curve: the acquisition of the right-of-way, and the construction costs. Talarico and Morrall (1993) state that the existing right-of-way usually provides sufficient space for minor roadway improvements; however, it does not usually provide sufficient space for curve flattening. Since the right-of-way procurement costs can vary widely from one site to the next, it will not be considered in the analysis.

6.2.2 Collision Savings

Several studies have illustrated that a relationship exists between the degree of curvature and the collision rate of a horizontal curve. The general trend outlined by Lamm et al. (1999) states that for an increasing degree of curve (decreasing radius of curve), the collision rate increases. When increasing the curve radius of a sharper horizontal curve, the degree of curvature decreases. With this decrease in DC , the collision rate on the alignment decreases, and the level of safety is improved. To quantify the benefits attributed to a reduction in collisions, consider the collision model used in the analysis by Talarico and Morrall (1993) given by:

$$\Delta A_t = AR_s(\Delta L)AADT + 0.0102(DC_1 - DC_2)AADT \quad (6.9)$$

where ΔA_t is the annual reduction in collisions, AR_s is the collision rate on a tangent section (collisions per million vehicle kilometres), ΔL is the change in length (km) calculated from Eq. 6.5, $AADT$ is the traffic volume in year t (in millions), and DC_1 and DC_2 are the degree of curvature of the TAC and new curve respectively. The annual savings due to collision reduction, K_t , can be calculated from:

$$K_t = \Delta A_t U \quad (6.10)$$

where ΔA_t is the annual reduction in collisions calculated from Eq. 6.9, and U is the average cost per collision (\$). Talarico and Morrall (1993) state that the cost of a collision varies with the severity of the collision. Also, of all collisions, generally 2% are fatal, 12.5% cause moderate injury, and 73% cause property damage. In Alberta, the costs (adjusted in 2003 dollars) are as follows (Howery, 1991): fatality, \$875,512.68; serious injury, \$586,036.07; moderate injury, \$5,729.49; and property damage, \$2,284.47. Using these collision figures, a weighted average cost per collision of \$93,148.47 was estimated and used for the analysis.

6.2.3 Vehicle Running Cost Savings

Vehicle operating costs are a function of vehicle type and speed, and also depend on the vertical profile of a highway alignment (Talarico and Morrall, 1993). The net savings per thousand vehicles in each year due to reduced vehicle running costs can be calculated from:

$$W_t = [P_{ht}(H_{ht}) + P_{sub}(H_{sub}) + P_{arv}(H_{arv})](\Delta L) \quad (6.11)$$

where W_t is the savings due to decreased running costs (2003 dollars per 1000 vehicles), P_{ht} is the percentage of heavy trucks, P_{sub} is the percentage of single unit trucks and busses, P_{arv}

is the percentage of autos and recreational vehicles, H_{ht} is the running cost of heavy trucks, H_{sub} is the running costs of single-unit trucks and buses (\$/1000 vehicle-km), and H_{arv} is the running cost of autos and recreational vehicles (\$/1000 vehicle-km). The vehicle running costs (2003 dollars) in Alberta are estimated as (Howery, 1991); autos = \$146.77 per 1000 veh-km; single unit trucks and busses = \$440.30 per 1000 veh-km; and heavy trucks = \$374.75 per 1000 veh-km. The vehicle running costs estimated by Howery (1991) have been adjusted (2003 dollars) and incorporated into this analysis.

6.2.4 Cost Associated with Speed Changes

The costs associated with speed changes vary with the size, weight, and type of vehicle. Talarico and Morrall (1993) used the models developed by Howery (1991) to estimate the speed change costs. The speed change costs can be estimated from the following models:

For Heavy Trucks,

$$Z_{ht} = (0.327V_i + 2.8)^{1.74} - (0.327V_f + 2.8)^{1.74} \quad (6.12)$$

For Single-Unit Trucks and Busses,

$$Z_{sub} = (0.1936V_i + 1.8)^{1.9} - (0.1936V_f + 1.8)^{1.9} \quad (6.13)$$

For Automobiles and Recreational vehicles,

$$Z_{arv} = (0.0588V_i + 1.4)^{1.9} - (0.0588V_f + 1.4)^{1.9} \quad (6.14)$$

where Z_{ht} is the speed change cost (adjusted 2003 dollars per 1000 heavy trucks), Z_{sub} is the speed change cost (adjusted 2003 dollars per 1000 single-unit trucks and busses), and Z_{arv} is the speed change cost (adjusted 2003 dollars per 1000 autos and recreational vehicles). The speed on the larger curve V_i and the speed on the sharper (TAC) curve V_f can be estimated from the following equation:

$$V_{85} = 104.82 - \frac{3574.51}{R} \quad (6.15)$$

where V_{85} is the 85th percentile operating speed, and R is the curve radius.

6.2.5 Net Present Value of the Annual Savings

The savings and benefits outlined in this section are annual savings and must be calculated for the life or duration of the project. The net present value of the annual operating savings and benefits can be calculated from:

$$NPV_{S\&B} = \sum_{t=1}^N \frac{1}{(1+i)^t} \quad (6.16)$$

where $NPV_{S\&B}$ is the net present value after t-years, i is the interest rate, and t is the life or duration of the project. The present value of a series of uniform end-of-period savings can be calculated by using the series present-worth factor given by:

$$P = A \frac{(1+i)^N - 1}{i(1+i)^N} \quad (6.17)$$

where P is the present value of a series of savings or benefits, A is the annual savings, N is the duration of the project, and i is the interest rate. For this analysis, a 20-year project life

and an interest rate of 3% has been assumed. The models were adjusted to 2003 dollars using an inflation rate of 1.83.

6.2.6 Other Considerations

It is important to note that many other benefits such as time savings, fuel savings, and winter maintenance savings (de-icing agents) exist. However, these benefits have not been included in the analysis as the savings are either difficult to quantify or very small. Also, the AADT of a highway alignment has a significant impact on the cost savings and benefits.

6.3 ANALYSIS OF SIMPLE HORIZONTAL CURVES

In this section, the improved simple horizontal curve design approach developed by Easa and Dabbour (2003) that proposed flatter minimum radii than those of TAC will be examined. The benefits, costs, and savings of the new design approach will be compared with the traditional TAC (1999) horizontal curve design approach. For this scenario, the benefits, costs and savings of a newly constructed simple horizontal curve will be examined. As well, it will be assumed that the two curves will share common tangents, and that the central angle of the new and TAC curve will be equal. It is important to note that the cost of land acquisition will not be considered in the analysis, as it is highly variable and dependent on several factors (Talarico and Morrall, 1993).

6.3.1 Case S1: Newly Constructed Simple Horizontal Curves

To compare the benefits, costs, and savings of the new simple horizontal curve design approach with the TAC (1999) designed horizontal curves, consider a simple horizontal

curve with design speed, $V_D = 110$ km/h, superelevation rate, $e_{max} = 0.04$, and a central angle of 45° . The TAC (1999) guidelines recommend a minimum radius of 680 m, while the new design approach (Easa and Dabbour, 2003) suggests a minimum radius of 810 m.

When the radius is increased from $R_{min} = 680$ m to $R_{min} = 810$ m, the horizontal curve radius is increased by 130 m. There are several benefits attributed to a larger radius horizontal curve. For horizontal curves of radii 680 m and 810 m, the degree of curvature (Eq. 6.1(b)) is 8.43° and 7.07° respectively. When the radius is increased, the total length of an alignment is decreased by an amount ΔL . By substituting $DC_1 = 8.43^\circ$, $DC_2 = 7.07^\circ$, and $\gamma = 45^\circ$, into Eq. 6.5, ΔL becomes 0.0429 km, or 42.9 m. Therefore, by using the radius developed by Easa and Dabbour (2003), we see that the total length of the alignment decreases by 42.9 m. This reduction in alignment length produces several benefits. The cost of the two horizontal curves can be estimated from the alignment length and the unit cost of highway construction (\$/km). When the radius of a curve is increased, the alignment length is reduced, and can be considered a construction savings, as it produces a shorter roadway alignment. Considering the average cost of a 9.0 m wide highway in Ontario is \$720,000/km (MTO, 2003), constructing the larger radius horizontal curve results in a construction savings of \$30,916.97.

By using the collision model developed by Glennon et al. (1985) where ΔA_t is the annual reduction in collisions, and by substituting $\Delta L = 0.0429$ km, $DC_1 = 8.43$, $DC_2 = 7.07$, $AR_s = 0.560$, and $AADT = 7000$ into Eq. 6.9, results in a 0.09681 reduction in the total number of annual collisions. The annual savings due to collision reduction, K_t , can be calculated from Eq. 6.10 where ΔA_t is the annual reduction in collisions calculated from

Eq. 6.9 and U is the average cost per collision (\$). Assuming the average cost of a collision is \$93,148.47 (in 2003 Dollars), the annual savings due to collision reductions in the first year is \$9,005.71. When the radius of a horizontal curve is increased, the resulting alignment length decreases resulting in savings due to reduced running costs. Motorists are able to traverse the curve at a higher rate of speed, resulting in speed change savings.

Assuming that autos make up 75% of the total vehicle traffic, single unit trucks and busses make up 7%, and heavy trucks make up 18% of the total share, the net savings per thousand vehicles in each year due to reduced vehicle running costs calculated from Eq. 6.11 is \$8.95 per 1000 vehicles. Therefore, the savings due to operating cost reductions for the first year is \$22 858.94 ($AADT = 7000$). The estimated speeds for the new curve design and TAC (1999) design calculated from Eq. 6.15 are 100.41km/h and 99.56 km/h respectively. The speed change savings achieved in each year for the three vehicle classes can be estimated from Eq. 6.12, 6.13, and 6.14. The estimated speed change savings for heavy trucks, single-unit trucks and buses, and for autos and recreational vehicles is \$12.33 per 1000 vehicles, \$8.85 per 1000 vehicles, and \$1.03 per 1000 vehicles respectively.

The net present value of the annual operating savings can be calculated by multiplying the annual savings and benefits by the present-worth factor given by Eq. 6.16. Assuming a project life of 20-years, and an interest rate of 3%, the present-worth factor is equal to 14.88. Therefore, over the 20-year life of the project, \$134,004.98 would be saved in collision reduction savings, \$137,274.04 would be saved due to speed change cost savings, \$340,141.04 would be saved due to reduced running costs, and \$30,916.97 would be saved in construction costs due to a reduced alignment length. In total, assuming a 20-year project

life, and an interest rate of 3%, \$642,337.03 in savings would be produced by constructing the simple horizontal curve ($V_D = 110$ km/h, $R_{min} = 810$ m, $e_{max} = 0.04$) developed by Easa and Dabbour (2003) compared with the TAC (1999) design curve ($V_D = 110$ km/h, $R_{min} = 680$ m, $e_{max} = 0.04$).

The simple horizontal curve developed by Easa and Dabbour (2003) is more cost effective to construct and maintain than the TAC (1999) designed horizontal curve, as it is shorter in overall length, and has a smaller degree of curvature. As well, several benefits such as time savings, vehicle operating costs, and collision reduction savings are produced when using the larger radius horizontal curve. The costs, benefits, and savings for other design speeds (60 km/h to 110 km/h) are presented in Tables 6.1 and 6.2 for $e_{max} = 0.04$.

6.3.2 Case S2: Flattening of Existing Simple Horizontal Curves

In the previous section, the benefits, costs, and savings were estimated for the construction of a new simple horizontal curve using the improved design approach developed by Easa and Dabbour (2003). In this section, the effects of increasing the radius of an existing TAC (1999) simple horizontal curve to that of the newly designed simple curves will be evaluated.

To compare the benefits, costs, and savings of the new horizontal curve design approach with the TAC (1999) designed horizontal curves, consider a similar simple horizontal curve with design speed, $V_D = 110$ km/h, superelevation rate, $e_{max} = 0.04$, and a central angle of 45° . The existing simple curve has a radius of 680 m, and will be increased to a radius of 810 m.

When the radius is increased from $R_{min} = 680$ m to $R_{min} = 810$ m, the horizontal curve radius is increased by 130 m.

The cost of flattening the TAC horizontal curve can be estimated by determining the length of the new curve. The alignment length of the larger 810 m horizontal curve is 0.636 km, and thus, the cost of constructing the new larger radius horizontal curve calculated from Eq. 6.6 is \$458,043.82. In order to determine whether or not it is feasible to increase the radius of the existing horizontal curve, the net present worth of the project must be calculated. The net present worth can be calculated from:

$$NPV_{total} = -C + NPV_S + NPV_B \quad (6.19)$$

Assuming a project life of 20-years, and an interest rate of 3%, the present-worth factor is equal to 14.88. Therefore, over the 20-year life of the project, \$134,004.98 would be saved in collision reduction savings, \$137,274.04 would be saved due to speed change savings, and \$340,141.04 would be saved due to reduced running costs. The cost of increasing the radius of the existing curve calculated from Eq. 6.6 is \$458,043.82. Therefore, the net present worth of the project can be calculated as:

$$NPV_{total} = -458,043.82 + 134,004.98 + 137,274.04 + 340,141.04 = \$153,376.24$$

Since the net present value is positive, increasing the radius of the existing horizontal curve is a cost-effective and economically feasible alternative. It is very important to note that the net present worth of any project is highly dependent on the AADT of the alignment. It was found that the net benefits increased for increasing values of AADT. For lower AADT

values (e.g., AADT < 1000), the net present worth of the project is negative, indicating that increasing the radius of the existing horizontal curve is not cost-effective.

An engineer or designer must conduct a comprehensive and detailed study focussing on the net benefits and costs of constructing the new alignment. If the net benefits and savings are greater than the cost of constructing the alignment, increasing the radius of the curve might be a feasible option. On the other hand, if the engineer or designer finds that the costs of constructing the new alignment far outweigh the net benefits, increasing the radius of the existing curve may not be a feasible alternative. Presented in Tables 6.1 and 6.2 are the construction costs, and benefits of increasing the radius of an existing simple horizontal curve.

Table 6.1 Annual Savings and Benefits of Implementing New Simple Horizontal Curves
($e_{max} = 0.04$ and AADT = 7,000)

											Speed Savings (\$)		
V_D (km/h)	$R_{min(1)}$	$R_{req(2)}$	DC_1	DC_2	V_1 (km/h)	V_2 (km/h)	ΔL (m)	Annual Collision	Annual Collision	Vehicle	Z_H	Z_{sub}	Z_{inv}
								Reductions A_i	Savings Kt (\$)	Operating Savings (\$)			
60	150	170	38.2	33.7	80.99	83.79	6.61	0.13	11789.33	1.38	35.94	25.13	2.98
70	200	230	28.65	24.91	86.95	89.28	9.91	0.11	10391.65	2.06	31.26	22.06	2.6
80	280	320	20.46	17.9	92.05	93.65	13.21	0.09	7970.2	2.75	22.17	15.76	1.85
90	380	440	15.08	13.02	95.41	96.7	19.82	0.08	7632.56	4.13	18.24	13.03	1.52
100	490	580	11.69	9.88	97.53	98.66	29.73	0.09	8366.66	6.19	16.32	11.7	1.36
110	680	810	8.43	7.07	99.56	100.41	42.94	0.1	9005.71	8.95	12.33	8.85	1.03

Table 6.2: NPV of Costs and Benefits of Implementing Larger Radius Simple Horizontal Curves in New and Existing Alignments over 20-year Project Life
($e_{max} = 4\%$ and AADT = 7,000)

V_D (km/h)	TAC $R_{min}(1)$	Improved Design $R_{req}(2)$	TAC DC_1	Improved Design DC_2	Total Annual Collision Savings K_i	Total Vehicle Operating Savings W_i	Total Speed Savings Z_i (20)	Construction Savings	Construction Cost	New Design NPW of Benefits (\$)	Existing Design NPW of Benefits (\$)
					(20) (\$)	(20) (\$)	(20) (\$)	(New Curve) (\$)	Existing Curve (\$)		
60	150	170	38.20	33.70	175425.24	52329.39	397773.30	4756.46	96132.65	630294.39	529395.28
70	200	230	28.65	24.91	154627.74	78494.09	346731.00	7134.68	130061.83	586987.51	449791.00
80	280	320	20.46	17.90	118696.59	104668.78	246308.59	9512.91	180955.58	479076.87	288608.38
90	380	440	15.08	13.02	113572.54	156988.17	202822.78	14269.37	248813.93	487652.86	224569.57
100	490	580	11.69	9.88	124495.94	235482.26	181685.01	21404.05	327982.00	563067.27	213681.22
110	680	810	8.43	7.07	134004.98	340141.04	137274.04	30916.97	458043.82	642337.03	153376.24

6.4 ANALYSIS OF REVERSE HORIZONTAL CURVES ($T = 0$)

In the previous section, costs and benefits were estimated for the construction of a new and existing simple horizontal curve using the new design approach developed by Easa and Dabbour (2003). In this section, the improved reverse horizontal curve design approach developed in Chapter 5 that considers the effects of vehicle stability and vertical alignment will be examined.

The costs and benefits of the new design approach will be compared against the traditional TAC (1999) design approach. For this scenario, costs and benefits of a newly constructed reverse horizontal curve will be examined. It is important to note that the cost of land acquisition and procurement will not be considered in the analysis, as it is highly variable and dependent on several factors (Talarico and Morrall, 1993). It is also assumed that both curves share common tangents, and that the central angle of both curves is equal.

Figure 6.2: Geometry of Reverse Curve

6.4.1 Case R1: Newly Constructed Reverse Horizontal Curves ($T = 0\text{m}$)

To compare the costs and benefits of the improved curve design approach with the TAC (1999) designed reverse curves, consider a reverse horizontal curve with design speed, $V_D = 70$ km/h, superelevation rate, $e_{max} = 0.06$, ratio $r = 1.0$, and a central angle of 45° . The TAC guidelines recommend a minimum radius of 190 m, while the new design approach suggests a minimum radius of 234 m. When the radii of the reverse curve are increased from $R_{min} = 190$ m to $R_{min} = 234$ m, the minimum curve radius is increased by 44 m. There are several benefits attributed to larger radii reverse horizontal curves. For horizontal curves of radii 190 m and 234 m, the degree of curvature (Eq.6.1 (b)) is 30.16° and 24.49° respectively. When the radius is increased, the total length of an alignment is decreased by an amount ΔL . By substituting $DC_1 = 30.16^\circ$, $DC_2 = 24.49^\circ$, and $\gamma = 45^\circ$, into Eq. 6.5, ΔL becomes 0.0145 km, or 14.5 m. Since reverse curves are composed of two simple horizontal curves, the alignment length decreases by 29 m ($14.5 * 2$). Therefore, by using the larger radius, the total length of the alignment decreases by 29 m. A reduced alignment length results in several savings and benefits.

The cost of the two reverse horizontal curves (new and TAC) can be calculated from the alignment length and the unit cost of highway construction (\$/km). When the radius of a curve is increased, the overall alignment length is reduced, and can be considered a construction savings, as it produces a shorter roadway alignment. Considering the average costs of a 9.0 m wide highway in Ontario is \$720,000/km (MTO, 2003), constructing the larger radii reverse horizontal curve results in a construction savings of \$20,928.41.

By using the collision model where ΔA_i is the annual reduction in collisions, and by substituting $\Delta L = 0.0218$ km, $DC_1 = 30.156^\circ$, $DC_2 = 24.49^\circ$, $AR_s = 0.560$, and $AADT = 7,000$ into Eq. 6.9, results in a 0.1686 reduction in the number of annual collisions. Since a reverse curve is composed of two simple horizontal curves, the annual reduction in the number of collisions is 0.3372. The annual savings due to collision reduction, K_i , can be calculated from Eq. 6.10 where ΔA_i is the annual reduction in collisions calculated from Eq. 6.9 and U is the average cost per collision (\$). Assuming the average cost of a collision is \$93,148.47 (in 2003 Dollars), the annual savings due to collision reductions in the first year is \$31,403.82.

When the radii of a reverse horizontal curve are increased, the resulting alignment length decreases resulting in savings due to reduced running costs. As well, motorists are able to traverse the reverse curve at higher rates of speed, resulting in speed change savings. Assuming that autos make up 75% of the total vehicle traffic, single unit trucks and busses make up 7%, and heavy trucks make up 18% of the total share, the net savings per thousand vehicles in each year due to reduced vehicle running costs calculated from Eq. 6.11 is \$6.06 per 1000 vehicles. Therefore, the savings due to operating cost reductions for the first year is \$15,473.74 ($AADT = 7,000$). The estimated speeds of the new reverse horizontal curve design and TAC (1999) design calculated from Eq. 6.15 are 89.54 km/h and 86.01 km/h respectively. The speed change savings achieved in each year for the three vehicle classes can be estimated from Eq. 6.12, 6.13, and 6.14. The estimated speed change savings for heavy trucks, single-unit trucks and buses, and autos and recreational vehicles is \$94.63 per 1000 vehicles, \$66.75 per 1000 vehicles, and \$7.86 per 1000 vehicles respectively.

The net present value of the annual operating and collision reduction savings can be calculated by multiplying the annual savings and benefits by the present-worth factor given by Eq. 6.16. Assuming a project life of 20-years, and an interest rate of 3%, the present-worth factor is equal to 14.88. Therefore, over the 20-year life of the project, \$467,288.85 would be saved in collision reduction savings, \$1,049,457.95 would be saved due to speed change savings, \$230,249.32 would be saved due to reduced running costs, and \$20,928.41 would be saved in construction costs due to a reduced alignment length. In total, assuming a 20-year project life, and an interest rate of 3%, \$1,767,924.53 in savings would be incurred by constructing the reverse horizontal curve ($V_D = 70$ km/h, $R_{min} = 234$ m, $e_{max} = 0.06$) developed by Easa et al. (2004) compared with the TAC (1999) design curve ($V_D = 70$ km/h, $R_{min} = 190$ m, $e_{max} = 0.06$).

The reverse horizontal curves developed in Chapter 5 are economically cheaper to construct and maintain than the TAC (1999) designed reverse curves, as they are shorter in overall alignment length, and have a smaller degree of curvature. As well, several benefits such as time savings, vehicle operating savings, and collision reductions savings occur when using larger radii reverse horizontal curves. The costs, benefits, and savings for other design speeds (60 km/h to 110 km/h) are presented in Tables 6.3 and 6.4 for $e_{max} = 0.06$.

6.4.2 Case R2: Flattening of Existing Reverse Horizontal Curves ($T=0$)

In this section, the effects of increasing the radii of an existing reverse horizontal curve (TAC, 1999) to that of the newly designed curves developed in Chapter 5 will be evaluated. To compare the benefits, costs, and savings of the new reverse curve design approach with the TAC (1999) designed reverse curves, consider a reverse horizontal curve with design

speed, $V_D = 70$ km/h, superelevation rate, $e_{max} = 0.06$, ratio, $r = 1.0$, and a central angle of 45° . The existing simple curve has a radius of 190 m, and will be increased to a radius of 234 m. When the radius is increased from $R_{min} = 190$ m to $R_{min} = 234$ m, the horizontal curve radius is increased by 44m.

The alignment length of the larger 234 m horizontal curve is 0.1838 km, and thus, the cost of constructing the new larger radius horizontal curve calculated from Eq. 6.6 is \$132,323.77. Since reverse curves are composed of two simple horizontal curves, the cost of the new alignment is \$264,647.54. In order to determine whether or not it is feasible to increase the radii of the existing reverse horizontal curve, the net present worth of the project must be calculated. The net present worth can be calculated from Eq. 6.19. Assuming a project life of 20-years, and an interest rate of 3%, the present-worth factor is equal to 14.88. Therefore, over the 20-year life of the project, \$467,288.85 would be saved in collision reduction savings, \$1,049,457.95 would be saved due to speed change savings, and \$230,249.32 would be saved due to reduced running costs. The cost of increasing the radius of the existing curve calculated from Eq. 6.6 is \$264,647.54. The net present worth of the project can be calculated as:

$$NPV_{total} = -264,647.54 + 467,288.85 + 1,049,457.95 + 264,647.54 = \$1,482,348.58$$

Since the net present value is positive, increasing the radius of the existing AASHTO designed reverse horizontal curve is a cost-effective and an economically feasible solution. It is very important to note that the net present worth of any project is highly dependent on the AADT of the alignment. It was found that the net benefits increased for increasing values of AADT. For lower AADT values (e.g., $AADT < 1000$), the net present worth of the project

would be negative, indicating that increasing the radius of the existing horizontal curve is not cost-effective. Tables 6.3 and 6.4 present the construction costs and benefits of increasing the radius of an existing reverse horizontal curve.

Table 6.3 Annual Savings and Benefits of Implementing New Reverse Horizontal Curves
($T = 0$ m, $e_{max} = 0.06$, and AADT = 7000)

V_D (km/h)	$R_{min}(1)$	$R_{req}(2)$	DC_1	DC_2	V_1 (km/h)	V_2 (km/h)	ΔL (m)	Annual Collision Reductions A_i	Annual Collision Savings K_i (\$)	Vehicle Operating Savings W_i (\$)	Speed Savings (\$)		
											Z_H	Z_{sub}	Z_{SN}
40	55	67	104.17	85.52	39.83	51.47	3.96	0.98	91643.15	1.65	203.41	131.58	16.89
50	90	109	63.66	52.56	65.1	72.03	6.28	0.6	55550.04	2.62	157.09	107.11	12.98
60	130	159	44.07	36.04	77.32	82.34	9.58	0.45	41581.45	3.99	125.88	87.63	10.43
70	190	234	30.16	24.49	86.01	89.54	14.53	0.34	31403.82	6.06	94.63	66.75	7.86
80	250	316	22.92	18.13	90.52	93.51	21.8	0.31	29051.04	9.08	82.48	58.56	6.86
90	340	435	16.85	13.17	94.31	96.6	31.38	0.28	26232.3	13.08	65.01	46.4	5.42
100	440	557	13.02	10.29	96.7	98.4	38.65	0.25	23581.31	16.1	49.03	35.1	4.09
110	600	761	9.55	7.53	98.86	100.12	53.18	0.26	23983.97	22.16	36.71	26.35	3.07

Table 6.4 Costs and Benefits of Implementing Larger Radius Reverse Horizontal Curves in New and Existing Alignments over 20-year Project Life
($T = 0$ m, $e_{max} = 0.06$, and AADT = 7,000)

V_D (km/h)	TAC $R_{min}(1)$	Improved Design $R_{req}(2)$	TAC DC_1	Improved Design DC_2	Total Annual Collision Savings K_i (20) (\$)	Total Vehicle Operating Savings W_i (20) (\$)	Total Speed Savings Z_i (20) (\$)	Construction Savings (New Curve) (\$)	Construction Cost Existing Curve (\$)	New Design NPW of Benefits (\$)	Existing Design NPW of Benefits (\$)
40	55	67	104.17	85.52	1363650.11	62795.27	2223642.64	5707.75	75775.15	3655795.77	3574312.87
50	90	109	63.66	52.56	826594.59	99425.84	1730242.40	9037.27	123275.99	2665290.10	2532976.84
60	130	159	44.07	36.04	618731.93	151755.24	1392051.85	13793.72	179824.61	2176332.75	1982714.41
70	190	234	30.16	24.49	467288.85	230249.32	1049457.95	20928.41	264647.54	1767924.53	1482348.58
80	250	316	22.92	18.13	432279.54	345373.98	915976.03	31392.61	357387.28	1725022.16	1336242.27
90	340	435	16.85	13.17	390336.57	497129.22	722849.69	45186.34	491972.99	1655501.81	1118342.48
100	440	557	13.02	10.29	350889.89	612253.88	545821.10	55650.54	629951.63	1564415.41	878813.24
110	600	761	9.55	7.53	366881.50	842503.20	408718.59	76578.95	860670.00	1684682.24	747433.29

6.5 ANALYSIS OF REVERSE HORIZONTAL CURVES ($T = 100\text{m}$)

In the previous sections, the benefits, costs, and savings were estimated for the construction of simple and reverse horizontal curves using the new design approach developed by Easa and Dabbour (2003) and in Chapter 5. In this section, the improved reverse horizontal curve design approach developed in Chapter 5 that considers vehicle stability, vertical alignment, and tangent length will be examined. The benefits, costs, and savings of the new design approach will be compared with the traditional TAC (1999) design approach. For this scenario, the benefits, costs, and savings of a newly constructed reverse horizontal curve having a 100 m intermediate tangent inserted between the two arcs will be examined. It is important to note that the cost of land acquisition and procurement will not be considered in the analysis, as it is highly variable and dependent on several factors (Talarico and Morrall, 1993).

6.5.1 Case RT1: Newly Constructed Reverse Curves ($T = 100\text{ m}$)

In this section, the effects of increasing the radii of a reverse horizontal curve (TAC design) with a 100 m tangent between the two arcs, to that of the newly designed reverse horizontal curves developed in Chapter 5 will be evaluated. To compare the benefits, costs, and savings of the improved reverse horizontal curve design approach with the TAC (1999) designed reverse horizontal curves, consider a reverse horizontal curve with design speed, $V_D = 70\text{ km/h}$, superelevation rate, $e_{max} = 0.06$, ratio $r = 1.0$, central angle of 45° , $R_{min} = 190\text{ m}$, and tangent length, $T = 100\text{ m}$. The existing reverse curve has radii of 190 m, and will be increased to a radius of 221 m. When the radii of a reverse horizontal curve are increased, the length of the new alignment is reduced resulting in many savings and benefits.

Figure 6.3: Reverse Curve with Tangent Length between two arcs.

The net present value of the annual operating and collision reduction savings can be calculated by multiplying the annual savings and benefits by the present-worth factor given by Eq. 6.16. Assuming a project life of 20-years, and an interest rate of 3%, the present-worth factor is equal to 14.88. Therefore, over the 20-year life of the project, \$346,203.45 would be saved in collision reduction savings, \$780,060.64 would be saved due to speed change savings, \$162,221.11 would be saved due to reduced running costs, and \$14,745.02 would be saved in construction costs due to a reduced alignment length. In total, assuming a 20-year project life, and an interest rate of 3%, \$1,303,230.23 in savings would be produced by constructing the reverse horizontal curve ($V_D = 70$ km/h, $R_{min} = 221$ m, $e_{max} = 0.06$) developed in this thesis (see Tables 6.5 and 6.6) compared with the TAC (1999) design curve ($V_D = 110$ km/h, $R_{min} = 190$ m, $e_{max} = 0.06$).

6.5.2 Case RT2: Flattening of an Existing Reverse Curve ($T = 100$ m)

In this section, the effects of increasing the radii of an existing reverse horizontal curve (TAC, 1999) to that of the newly designed curves developed in Chapter 5 will be evaluated.

To compare the benefits, costs, and savings of the new reverse curve design approach with the TAC (1999) designed reverse curves, consider a reverse horizontal curve with design speed, $V_D = 70$ km/h, $R_{min} = 190$ m, superelevation rate, $e_{max} = 0.06$, ratio, $r = 1.0$, $T = 100$ m and a central angle of 45° . The existing reverse has curve radii of 190 m, and will be increased to a radius of 221 m. There are several benefits attributed to a flatter reverse curve. When the radius is increased, the total length of an alignment is decreased by an amount $2\Delta L$. The cost of flattening the TAC reverse curve can be estimated by determining the length of the new curve.

The alignment length of the larger 221 m horizontal curve is 0.1736 km, and thus, the cost of constructing the new larger radius horizontal curve calculated from Eq. 6.6 is \$124,972.45. Since reverse curves are composed of two simple horizontal curves, the cost of the new alignment is \$249,944.90. In order to determine whether or not it is feasible to increase the radii of the existing reverse horizontal curve, the net present worth of the project must be calculated. The net present worth can be calculated from Eq. 6.19. Assuming a project life of 20-years, and an interest rate of 3%, the present-worth factor is equal to 14.88. Therefore, over the 20-year life of the project, \$346,203.45 would be saved in collision reduction savings, \$780,060.64 would be saved due to speed change savings, and \$162,221.11 would be saved due to reduced running costs. The cost of increasing the radius of the existing curve calculated from Eq. 6.6 is \$249,944.90. The net present worth of the project can be calculated as:

$$NPV_{total} = -249,944.90 + 346,203.45 + 780,060.64 + 162,221.11 = \$1,038,540.31$$

Since the net present value is positive, increasing the radius of the existing TAC designed reverse horizontal curve is a cost-effective and economically feasible solution. It is very important to note that the net present worth of any project is highly dependent on the AADT of the alignment. It was found that the net benefits increased for increasing values of AADT. For lower AADT values (e.g., AADT < 1000), the net present worth of the project would be negative, indicating that increasing the radius of the existing horizontal curve is not cost-effective.

Table 6.5: Annual Savings and Benefits of Implementing New Reverse Horizontal Curves
($T = 100$ m, $e_{max} = 0.06$ and AADT = 7,000)

V_D (km/h)	$R_{TAC(1)}$	$R_{TAC(2)}$	DC_1	DC_2	V_1 (km/h)	V_2 (km/h)	ΔL (m)	Annual Collision Reductions A_i	Annual Collision Savings K_i (\$)	Vehicle Operating Savings V_i (\$)	Speed Savings (\$)		
											Z_{tt}	Z_{sub}	Z_{AV}
40	55	59	104.17	97.11	39.83	44.24	1.321	0.37	34642.03	0.55	73.18	46.85	6.09
50	90	98	63.66	58.47	65.1	68.35	2.642	0.28	25905.78	1.1	72.27	49.09	5.97
60	130	144	44.07	39.79	77.32	80	4.624	0.24	22036.42	1.93	66.44	46.16	5.5
70	190	221	30.16	25.93	86.01	88.65	10.24	0.25	23266.36	4.27	70.35	49.59	5.84
80	250	291	22.92	19.69	90.52	92.54	13.543	0.21	19287.15	5.64	55.44	39.33	4.61
90	340	400	16.85	14.32	94.31	95.88	19.819	0.19	17555.21	8.26	44.54	31.77	3.71
100	440	519	13.02	11.04	96.7	97.93	26.094	0.18	16578.95	10.87	35.47	25.39	2.96
110	600	714	9.55	8.02	98.86	99.81	37.655	0.19	17439.62	15.69	27.67	19.86	2.31

Table 6.6: NPV of Costs and Benefits of Implementing Larger Radius Reverse Horizontal Curves in New and Existing Alignments over 20-year Project Life
($T = 100$ m, $e_{max} = 0.06$, and AADT = 7,000)

V_D (km/h)	TAC $R_{TAC(1)}$	Improved Design $R_{TAC(2)}$	TAC DC_1	Improved Design DC_2	Total Annual Collision Savings K_i (20) (\$)	Total Vehicle Operating Savings V_i (20) (\$)	Total Speed Savings Z_i (20) (\$)	Construction Savings (New Curve) (\$)	Construction Cost Existing Curve (\$)	New Design NPW of Benefits (\$)	Existing Design NPW of Benefits (\$)
40	55	59	104.17	97.11	515473.37	20931.76	799214.49	1902.58	66727.37	1337522.20	1269992.24
50	90	98	63.66	58.47	386924.41	41863.51	795453.65	3805.17	110835.30	1227046.73	1112406.27
60	130	144	44.07	39.79	327901.98	73261.15	734445.52	6659.04	162850.03	1142257.69	972748.63
70	190	221	30.16	25.93	346203.45	162221.11	780060.64	14745.02	249944.90	1303230.23	1039540.31
80	250	291	22.92	19.69	288992.84	214550.50	615587.21	19501.47	329112.97	1136632.02	788017.58
90	340	400	16.85	14.32	261221.49	313976.35	495152.61	28538.74	452388.96	1098889.19	617951.49
100	440	519	13.02	11.04	246694.84	413402.19	394705.56	37576.01	586974.68	1092378.60	467827.92
110	600	714	9.55	8.02	259501.56	596555.06	308110.82	54223.61	807514.29	1216391.05	356653.15

6.6 SENSITIVITY ANALYSIS

When conducting a cost-benefit analysis, it is important to conduct a sensitivity analysis that examines the effects of increases and decreases to some of the major variables. The average annual daily traffic (AADT) of an alignment is a function of many of the cost and savings models used in this analysis. Changes in the AADT will have a profound effect on the total net present worth of a project. Also, the construction costs are an important factor to consider in any new highway design.

6.6.1 Average Annual Daily Traffic

One of the most important factors affecting the cost-effectiveness of increasing the radius of a simple or reverse horizontal curve is the annual average daily traffic (AADT) of the highway alignment. Since most of the savings and benefits models used in this analysis are a function of the AADT, it was observed that AADT has a significant impact on the net present worth of a project. The laws of probability and statistics dictate that the number of collisions on a highway alignment will largely be a function of the total number of traffic on that alignment. In other words, the more traffic and vehicles operating on an alignment, the greater the chances of a traffic collision.

Presented in Figure 6.4 is the net present worth of increasing the radius of an existing curve to the new simple horizontal curves ($e_{max} = 0.04$, $V_D = 60$ km/h to 110 km/h, AADT = 1,000) developed by Easa and Dabbour (2003). It is observed that for design speeds of 80 km/h to 110 km/h, increasing the radius of the simple horizontal curve according to the new guidelines would involve some cost. This makes sense since the AADT on the alignment is

relatively low ($AADT = 1,000$), and thus, many of the benefits such as collision savings, and vehicle operating savings will be very low or negligible. However, for a higher volume of traffic, the cost-effectiveness of increasing the radius is economically feasible (Fig. 6.5). It was also observed that the benefits decreased for increases in V_D . The general trend observed is that when the AADT of an alignment increases, the net present worth of a project increases as well. Presented in Figure 6.6 is the NPW of increasing the radius of an existing simple horizontal curve. It is observed that for an increase in the AADT, the net present worth of a project increases as well. When the design speed is less than 80 km/h, the NPW of the project is positive for AADT values between 1,000 and 13,000. For higher values of design speed (>80 km/h), an increase in radius would involve some cost if low volumes of AADT are present.

Figure 6.4: Net Present Worth of Simple Horizontal Curves with $AADT = 1000$

Figure 6.5: Variation of the Net Present Worth of Simple Horizontal Curves with Design Speed

Figure 6.6: Effect of AADT on the Net Present Worth Benefits for Simple Curves for various design speeds.

6.6.2 Unit Construction Cost

The construction cost of a highway design is one of the most important factors to consider when conducting a cost-benefit analysis. When increasing the radius of an existing horizontal curve, the construction cost is a function of the alignment length. Therefore, for larger radius horizontal curves, the alignment length will be longer than a smaller (lower design speed) horizontal curve. In this analysis, an estimated construction cost of \$720,000/km (MTO, 2003 dollars) was considered. In Figure 6.7, the effects of changes in the unit cost of construction are presented. The thick solid line in Figure 6.7 is the \$720,000/km unit cost that was used for the analysis. If the unit cost is decreased by 10%, the construction cost of the new design decreases as well (dashed line). When the unit cost of construction is increased by 10% (thinner line), the construction cost of the new design increases with increasing design speed.

Figure 6.7: Construction Cost for Simple Horizontal Curve ($e_{\max} = 0.04$)

When constructing a new larger-radius reverse horizontal curve, the length of the curve increases, but the total alignment length is reduced. The construction savings attributed to a shorter highway alignment are presented in Figure 6.8 for a unit cost of \$720,000, and a +/- 10% change in unit cost. When the construction costs are reduced (-10%), the net savings due to a shorter alignment length are reduced. However, it is important to note that regardless of the unit cost, the reverse horizontal curves developed in Chapter 5 are always more cost effective to construct given the major assumption that the two curves share common tangents, and that the central angle of the new and TAC curve are the same.

Figure 6.8: Effect of a 10% increase and 10% decrease in Cost on the Construction Savings of constructing a larger radius simple horizontal curve ($e_{\max} = 0.04$).

CHAPTER 7: SUMMARY AND CONCLUSIONS

7.1 SUMMARY

Reverse horizontal curves are used in combination with vertical alignments on various classifications of highways. Traditional AASHTO (2001) and TAC (1999) geometric design guides do not adequately address the effects of vehicle stability or 3D-alignment on the required minimum radius of horizontal curves. This research utilized vehicle simulation software to investigate the effects of vehicle stability on reverse horizontal curves with intermediate tangents. A comparison examining the lateral acceleration levels for trucks on simple horizontal curves versus the lateral acceleration levels for trucks on reverse curves with 3D-alignment were carried out. The effect of an intermediate tangent placed between the two arcs of a reverse curve was also examined, along with the spiral length requirements of the new reverse horizontal curves.

An advanced vehicle simulation software called VDM RoAD (Vehicle Dynamic Models Roadway Analysis and Design) developed by the University of Michigan Transportation Research Institute (UMTRI) was used to conduct the experimental work. VDM RoAD accurately simulates a vehicle travelling through a user-defined alignment, taking into account vehicle characteristics such as body roll, moments, pitch, and yaw. The software has a comprehensive vehicle library consisting of several tractor and trailer combinations, buses, and a passenger car. The three design vehicles (trucks) that were used in the analysis are

WB-15, WB-20, and NHTSA 3a/2a. The WB-15 design vehicle represents a moderate sized truck, the WB-20 design vehicle is one of the largest design trucks in AASHTO, and the NHTSA 3a/2a combo represents a real truck operating on North American highways as its properties were measured from the field to support research at the National Highway Traffic Safety Administration (NHTSA).

Simulation runs were carried out on flat (minimum radius) simple horizontal curves with superelevation rates of 4% and 6% (base case). Vehicle dynamics (lateral accelerations) were recorded for the base case in order to facilitate a comparison with subsequent reverse horizontal curves combined with vertical alignments. The vertical alignments include crest and sag vertical curves, upgrades, and downgrades. Maximum grades of +6% and -6% were selected since this grade is the controlling limit used on Canadian rural highways. Three values of the ratio between the arcs of the reverse horizontal curve were evaluated (1.0, 1.5, 2.0). Several lengths of intermediate tangent were inserted between the two arcs of the reverse curve to examine the effects on the required minimum radius.

The cost-effectiveness of implementing a larger radius for simple and reverse horizontal curves was investigated. Several savings models developed for Alberta highways were adopted and used in this analysis. The savings models used in this analysis are construction savings, collision reduction savings, vehicle running savings, and speed change savings models. Three scenarios were evaluated; simple horizontal curves, reverse horizontal curves with no intermediate tangents, and reverse horizontal curves with an intermediate tangent.

7.2 CONCLUSIONS

The main objective of this research was to develop new design guidelines for the required minimum radius of reverse horizontal curves with intermediate tangents. The effect of reverse curvature on vehicle stability causes an increase in the minimum required radius of horizontal curves in order to maintain the same lateral accelerations that the AASHTO (2001) and TAC (1999) geometric design guides recommend. The increase in required minimum radius was found to range between 3% and 25% depending on the superelevation rate, type of design vehicle, design speed, and ratio between the radii of the shaper and flatter arcs of the reverse curve. Furthermore, when reverse curves are superimposed with various vertical alignments, an additional increase in the minimum required radius occurs. The increase was found to range between 4% and 29% to compensate for the effects of reverse curvature and vertical alignment.

An intermediate tangent length placed between the two arcs of a reverse horizontal curve reduces the required increase in the minimum radius for a reverse curve without an intermediate tangent. As the tangent length increases, the increase in the minimum radius decreases. For a 3D reverse horizontal curve without an intermediate tangent (WB20, $V_D = 90$ km/h, $e_{max} = 0.06$, $r = 1.0$, and $T = 0$ m), the required increase in minimum radius is 29.1%. This increase in minimum radius is due to the effects of reverse curvature and vertical alignment. For the same 3D reverse horizontal curve with a 100 m intermediate tangent, the required increase in minimum radius is reduced to 18.2%. For this case, the intermediate tangent produces an 11% reduction in the increase in minimum radius due to the effects of reverse curvature and vertical alignment.

Two mathematical models were developed to determine the required minimum radius for reverse horizontal curves with an intermediate tangent. The models were developed to relate the minimum radius requirement to design speed and different geometric alignment configurations. For the Flat Reverse Curve Model, the required minimum radius was set as the dependent variable, and the design speed, tangent length, superelevation rate, vehicle type, and ratio between curve radii were set as the independent variables. For the 3D Reverse Curve Model, the required minimum radius was set as the dependent variable, and the design speed, tangent length, superelevation rate, vehicle type, ratio between curve radii, and slope of the first and second grades were set as the independent variables. All variables for the Flat and 3D Reverse Curve Models were found to be significant (at the 95% confidence level) and had R^2 values of 0.986 and 0.983 respectively, indicating a strong relationship.

The results of this research indicate that an increase in the minimum radius of reverse horizontal curves is required to satisfy vehicle stability requirements. The minimum radius requirements for reverse horizontal curves with intermediate tangents developed in Chapter 5 produce larger radii horizontal curves than those of the current design guides. The new design approach, which considers vehicle stability and three-dimensional alignments, showed that the minimum radius requirements of current design guides of the Transportation Association of Canada (TAC, 1999) and the American Association of State Highway and Transportation Officials (AASHTO, 2001) are inadequate. The larger radii (flatter curves) of the new approach provide a smoother transition between the tangent and curved sections of the highway. Also, the required larger radius results in a reduction in the overall length of the alignment. In turn, this reduction in highway alignment results in savings in construction,

collision, maintenance, and vehicle operating costs. Using the larger radii reverse curves produces a 2.6 m to 75.3 m reduction in alignment length (NHTSA 3a/2a, $V_D = 40$ km/h to 110 km/h, $e_{max} = 0.06$, $r = 1.0$, and $T = 100$ m). The construction savings as a result of the reduced alignment length ranged from \$1,902.58 to \$54,233.61.

7.3 RECOMMENDATIONS

Current geometric design guides do not address the effects of reverse curvature or vertical alignment on the minimum radius of horizontal curves. As a result, highway alignments with reverse curvature are inadequately designed. A solution to this problem was researched and evaluated in this study. Based on the results of this study, the following recommendations for future research are offered:

1. An intermediate tangent length placed between the two arcs of a reverse horizontal curve reduces the required increase in the minimum radius caused by reverse curvature and vertical alignment. As the tangent length is increased, the increase in the required radius continues to decrease. An intermediate tangent length placed between a reverse curve helps improve the driving dynamics of large vehicles such as heavy tractor-trailer combinations, and increases the stability of vehicles operating on the alignment. This helps improve the level of safety on the alignment section, and reduces the risk of rollover.
2. In situations where the minimum radius of a reverse curve is constrained or limited to a specified value (physical or environmental constraints), a tangent length placed between the two arcs may provide solutions to the problem. By using a tangent, the required

increase in the minimum radius is reduced for increasing tangent lengths. Therefore, the reverse curve can be designed having smaller radii (R_1 and R_2), and the problem of vehicle stability is still addressed.

3. The results presented in this thesis were based on side friction factors for mainline highways and interstates. These side friction factors differ from those used for urban streets or interchange ramps. Future work should be conducted on the minimum radius requirements for interchange ramps and urban streets based on the procedures carried out in this study.
4. The models developed in this thesis can be used to compute the recommended speed for existing reverse horizontal curve alignments on North American highways. By knowing the radius and other geometric properties of the highway alignment, a recommended speed for the alignment of interest can be back-calculated from the models.
5. Environmental conditions such as weather and temperature should be addressed or considered. VDM RoAD does not have the capabilities to simulate icy road conditions, heavy rain, or hydroplaning. Also, complex human factors such as driver fatigue, and braking as a reaction to increased levels of lateral acceleration should be examined.
6. Additional cost and benefit models should be incorporated into the cost-effectiveness analysis. Other costs such as maintenance and rehabilitation costs, and land acquisition costs should be investigated.

7. In this thesis, the cost effectiveness analysis was based on the assumption that the flatter and sharper curves shared the same tangents, and the deflection angle of both curves is the same. A cost effectiveness analysis for the condition where the flatter (larger radius) curve passes through the PC or PT of an existing TAC simple curve should be examined (tangents and deflection angle are not the same).
8. Warning Signage such as truck rollover signs should be placed at or just before complex highway alignments such as 3D reverse horizontal curves or highway interchanges to warn a driver of the upcoming complex alignment.
9. In the future, TAC and AASHTO should consider the effects of vehicle stability and 3D alignment on the development of geometric design guidelines. Vehicle simulation software such as VDM RoAD should be used to examine the safety of highway designs prior to construction. Also, tractor-trailers should be considered as possible design vehicles on highway alignments where a high proportion of truck traffic is present.

REFERENCES

American Association of State Highway and Transportation Officials, AASHTO, 2001. *A Policy on Geometric Design of Highways and Streets*. AASHTO, Washington, D.C.

Baker, D., Bushman, R., and Berthelot, C., 2001. *Effectiveness of Truck Rollover Warning Systems*. Transportation Research Record No. 1779, TRB, National Research Council, Washington D.C., pp. 134-140.

Bonneson, J., A., 2001. *Controls for Horizontal Curve Design*. Transportation Research Record No. 1751, TRB, National Research Council, Washington D.C., pp. 82-89.

Bonneson, J., A., 2000. *Kinematic Approach to Horizontal Curve Transition Design*. Transportation Research Record No. 1737, TRB, Washington D.C., pp. 1-8.

Choueiri, E., Lamm, H., Kloeckner, J., and Mailaender, T., 1994. *Safety Aspects of Individual Design Elements and Their Interactions on Two-Lane Highways: International Perspective*. Transportation Research Record No. 1445, TRB, Washington D.C., pp.34-36.

Chrstos, J.P and Grygier, P.A., 1997. *Experimental Testing of a 1994 Ford Taurus for NADSdyna Validation*, SAE Paper No. 970563, NHTSA, 1997.

Website: http://www.nrd.nhtsa.dot.gov/vrtc/ca/capubs/taurus_test.pdf

Dabbour, E., 2003. *Truck Stability on Different Types of Horizontal Curves Combined With Vertical Alignments*. Masters Thesis, Ryerson University, Toronto, Ontario, Canada.

Dabbour, E., Easa, S.M., and Abd El Halim, A, O., 2003. *Design Guidelines for Horizontal Reverse Curves Combined with Vertical Alignments*. CSCE 31st Annual Conference, Moncton, NB.

Dabbour, E., Easa, S., and Abd El Halim, A.O.. 2003. *Radius Requirements for Reverse Horizontal Curves on 3D Alignments*. ASCE Journal of Transportation Engineering [in review].

Daniel, J., Tsai, C., and Chien, S., 2002. *Factors in Truck Crashes on Roadways with Intersections*. Transportation Research Record No. 1818, TRB, National Research Council, Washington D.C., pp. 54-59.

Donnel, E., Ni, Y., Adolini, M., and Elefteriadou, L., 2001. *Speed Prediction Models for Trucks on Two-Lane Rural Highways*. Transportation Research Record No. 1751, TRB, National Research Council, Washington D.C., pp. 44-45.

Easa, S., M., 1994. *Design Considerations for Highway Reverse Curves*. Transportation Research Record No. 1445, TRB, National Research Council, Washington D.C., pp 1-11.

Easa, S.M., 1994, *New and Improved Unsymmetrical Vertical Curve for Highways*, Transportation Research Record No. 1445, TRB, National Research Council, Washington D.C., pp. 94-100.

Easa, S., M., and Hassan, Y., 1998. *Design Requirements of Equal-Arc Unsymmetrical Vertical Curves*. Journal of Transportation Engineering, ASCE, Vol. 124, No.5, pp. 404-410.

Easa, S, M., and Hassan, Y., 2000. *Transitioned vertical curves: I. Properties*, Transportation Research, Vol. 34a, No. 6, pp. 481-496.

Easa, S., M., 2000. *Reliability Approach To Intersection Sight Distance Design*. Transportation Research Record No. 1701, TRB, Washington, D.C., pp.42-52.

Easa, S., M., Strauss, T., R., Hassan, Y., and Souleyrette, R., 2002. *Three-Dimensional Analysis: Planning and Design*. Journal of Transportation Engineering, ASCE, Vol. 128, pp. 250-258.

Easa, S., M., and Dabbour, E., 2003. *Design Radius Requirements for Simple Horizontal Curve on 3D Alignments*. Canadian Journal of Civil Engineering, 30(6), pp. 1022-1033.

Easa, S., M., 2003. *Geometric Design in The Civil Engineering Handbook, Second Edition. Chapter 63, CRC Press LLC, Canada.*

Easa, S.M., and Dabbour, E., 2004. *Radius Requirements for Compound Horizontal Curves on Three Dimensional Alignments*, ASCE, Journal of Transportation Engineering [under review].

Fambro, D., B., Fitzpatrick, K., Koppa, R., J., 2000. *New Stopping Sight Distance Model for Use In Highway Geometric Design*. Transportation Research Record No. 1701, TRB, Washington, D.C., pp.1-8.

Fitzpatrick, K., Fambro, D., B., Stoddard, M., 2000. *Safety Effects of Limited Stopping Sight Distance on Crest Vertical Curves*. Transportation Research Record No. 1701, TRB, Washington D.C., pp. 17-24.

Furtado, G., Easa, S., M., and Abd El Halim, A., O., 2002. *Vehicle Stability on Combined Horizontal and Vertical Alignments*. CSCE 4th Transportation Specialty Conference, Montreal, QC.

Furtado, G., 2002. *Vehicle Stability on Combined Horizontal and Vertical Alignments*. Thesis submitted to Department of Civil and Environmental Engineering, Carleton University, Ottawa, Canada.

Garber, N., Hoel, L., 1997. *Traffic and Highway Engineering, Second Edition*. PWS Publishing Company. Boston, MA.

Garcia, L., O., Wilson, F., R., and Innes, D., 2003. *Heavy Truck Dynamic Rollover: Effect of Load Distribution, Cargo Type, and Road Design Characteristics*. Annual Meeting of Transportation Research Boards, Washington D.C.

Gibreel, G., M., Easa, S., M., and El-Dimeery, I., A., 2001. *Prediction of Operating Speed on Three-Dimensional Highway Alignments*. Journal of Transportation Engineering, ASCE, Vol. 127, pp.21-29.

Gillespie, T, D., 1992. *Fundamentals of Vehicle Dynamics*. SAE, Inc., Warrendale, PA.

Hassan, Y., 2003. *Improved Design of Vertical Curves with Sight Distance Profiles*. Transportation Research Record No. 1851, TRB, Washington D.C., pp.13-24.

Hassan, Y., Easa, S., M., and Abd El Halim, A., O., 1998. *Highway Alignment: Three Dimensional Problem and Three Dimensional Solution*. Transportation Research Record No. 1616, TRB, National Research Council, pp. 17-25.

Hassan, Y., and Easa, S., M., 2001. *Effect of Vertical Alignment on Drivers Perception of Horizontal Curves*. Journal of Transportation Engineering, ASCE.

Hassan, Y., 1996. *Geometric Design Considerations of Combined Horizontal and Vertical Highway Alignments*. A PhD Thesis, Department of Civil and Environmental Engineering, Carleton University, Ottawa, Ontario.

Harwood, D., W., Mason, J., M., 1994. *Horizontal Curve Design for Passenger Cars and Trucks*. Transportation Research Record No. 1445, TRB, Washington, D.C., pp. 22-33.

Howery, K.E. and Applications Management Company 1991. *Benefit cost analysis*. Report prepared for Alberta Transportation and Utilities.

Hunter, M., Machemehl, R., and Tsyganov, A., 2001. *Operational Evaluation of Freeway Ramp Design*. Transportation Research Record No. 1751, TRB, Washington D.C., pp. 90-101.

Interactive Highway Safety Design Model, Engineer's Manual., 2003. McLean VA.

Kanellaidis, G., 1996. *Human Factors in Highway Geometric Design*. Journal of Transportation Engineering, ASCE, Vol. 122, pp. 59-65.

Krebs, H., G., and Kloeckner, J., H., 1977. *Investigations of the Effect of Highway and Traffic Conditions Outside Built-Up Areas on Accident Rates*. Research Road Construction and Road Traffic Technique. Minister of Transportation, Bonn, Germany, Vol. 223, pp. 1-63.

Lamm, R., Psarianos, B., Mailaender, T., 1999. *Highway Design and Traffic Safety Engineering Handbook*. McGraw-Hill Companies, Inc., New York, NY.

Lamm, R., Choueiri, M., E., 1987. *Recommendations for Evaluating Horizontal Design Consistency, Based on Investigations in the State of New York*. Transportation Research Record No. 1122, TRB, Washington D.C., pp. 68-78.

Liapis, E., D., Psarianos, B., Kasapi, E., 2001. *Speed Behaviour at Curved Ramp Sections of Minor Interchanges*. Transportation Research Record No. 1751, TRB, Washington D.C., pp. 35-43.

MacAdam, C., C., Fancher, P., S., Segel, L., 1985. *Side Friction for Superelevation on Horizontal Curves*. University of Michigan Transportation Research Institute, UMTRI – 73895, Michigan.

Messer, C., 1980. *Methodology for Evaluating Geometric Design Consistency*. Transportation Research Record No. 757, TRB, Washington D.C., pp. 7-14.

Morrall, J., F., and Talarico, R., J., 1994. *Side Friction Demanded and Margins of Safety on Horizontal Curves*. Transportation Research Record No. 1435, TRB, Washington D.C., pp. 145-152.

Ng, J., and Sayed, T., 2003. *Quantifying the Relationship Between Geometric Design Consistency and Road Safety*. Transportation Research Record, TRB, Washington D.C.

Nicholson, A., 1998. *Superelevation, Side Friction, and Roadway Consistency*. Journal of Transportation Engineering, ASCE, Vol. 124, pp. 411-418.

Ottesen, J., and Krammes R., 2000. *Speed-Profile Model for a Design-Consistency Evaluation Procedure in the United States*. Transportation Research Record No. 1701, TRB, Washington D.C., pp.76-86.

Papacostas, C., S., and Prevedouros, P., D., 1993. *Transportation Engineering and Planning, 2nd Edition*. Prentice-Hall Inc., USA.

Rumar, K., 1985. *Safety Problems and Countermeasures Effects in the Nordic Countries*. International Meeting on the Evaluation of Local Traffic Safety Measures, Paris, France.

Sayers, M., W., 1999. *Vehicle Dynamics Programs for Roadway and Roadside Studies*. University of Michigan Transportation Institute, UMTRI-98-20-1, Michigan.

Sayers, M., W., 1999. *VDM RoAD User Reference Manual*. University of Michigan Transportation Research Institute, UMTRI-98-20-2, Michigan.

Smith, B., L., and Lamm, R., 1994. *Coordination of Horizontal and Vertical Alignment with Regard to Highway Esthetics*. Transportation Research Record No. 1445, TRB, Washington D.C., pp. 73-85.

SPSS for Windows, Release 11.5.0, SPSS Inc., LEAD Technologies Inc., 2002.

Swiss Association of Road Specialists (VSS)., 1991. *Highway Design, Fundamentals, Speed as a Design Element*. Zurich, Switzerland.

Talarico, R., J., and Morrall, J., F., 1993. *The cost-effectiveness of curve flattening in Alberta*. Canadian Journal of Civil Engineering, Vol. 21, pp. 285-296.

Taignidis, I., Kanellaidis, G., 2001. *Required Stopping Sight Distance on Crest Vertical Curves*. Journal of Transportation Engineering, ASCE, pp. 275-282.

Transportation Association of Canada (TAC)., 1999. *Geometric Design Guide for Canadian Roads*. Ottawa, Ontario, Canada.

Tsai, F., F., and Haug, E., J., 1989. *Automated Methods for High Speed Simulation for Multibody Dynamic System*. Technical Report R047, Center for Computer Aided Design, The University of Iowa.

United States Department of Transportation (USDOT), 2002. *National Transportation Statistics (NTS) 2002*. U.S. Government Printing Office, Washington D.C.

Wooldridge, D., M., Fitzpatrick, K., Koppa, R., and Bauer, K., 2000. *Effects of Horizontal Curvature on Driver Visual Demand*. Transportation Research Record No. 1737, TRB, Washington D.C., pp. 71-77.

APPENDIX A

MINIMUM REQUIRED RADIUS FOR REVERSE CURVES WITH INTERMEDIATE TANGENTS FOR LEVEL AND 3D ALIGNMENTS

(For $e_{max} = 0.04$ and 0.06)

Table A1: Required Increase in curve radius R_1 corresponding to different values of opposite curve radii R_2 for different design vehicles ($e_{max} = 0.04$) for level terrain.

Tangent Length, T = 0m

Design Speed (km/h)	Design Minimum Radius (m)	R2 = 1.0 R1		R2 = 1.5 R1		R2 = 2.0 R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	60	65	8.3%	63	5.0%	62	3.3%
50	100	109	9.0%	106	6.0%	104	4.0%
60	150	168	12.0%	160	6.7%	156	4.0%
70	200	225	12.5%	215	7.5%	211	5.5%
80	280	316	12.9%	306	9.3%	297	6.1%
90	380	438	15.3%	417	9.7%	408	7.4%
100	490	574	17.1%	545	11.2%	529	8.0%
110	680	811	19.3%	764	12.4%	749	10.1%
(b) WB-20							
40	60	71	18.3%	69	15.0%	66	10.0%
50	100	117	17.0%	113	13.0%	111	11.0%
60	150	178	18.7%	174	16.0%	170	13.3%
70	200	240	20.0%	234	17.0%	228	14.0%
80	280	339	21.1%	330	17.9%	319	13.9%
90	380	464	22.1%	459	20.8%	444	16.8%
100	490	601	22.7%	593	21.0%	575	17.3%
110	680	850	25.0%	830	22.1%	815	19.9%
(c) NHTSA 3a / 2a							
40	60	68	13.3%	65	8.3%	64	6.7%
50	100	113	13.0%	109	9.0%	108	8.0%
60	150	171	14.0%	165	10.0%	163	8.7%
70	200	230	15.0%	225	12.5%	218	9.0%
80	280	325	16.1%	320	14.3%	314	12.1%
90	380	449	18.2%	439	15.5%	433	13.9%
100	490	584	19.2%	573	16.9%	568	15.9%
110	680	824	21.2%	796	17.1%	794	16.8%

Table A2: Required Increase in curve radius R_1 corresponding to different values of opposite curve radii R_2 for different design vehicles ($e_{max} = 0.04$) for level terrain.

Tangent Length, T = 50m

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 1.0 R1		R2 = 1.5 R1		R2 = 2.0 R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	60	62	3.3%	61	1.7%	61	1.7%
50	100	104	4.0%	102	2.0%	102	2.0%
60	150	156	4.0%	153	2.0%	154	2.7%
70	200	209	4.5%	205	2.5%	206	3.0%
80	280	297	6.1%	290	3.6%	289	3.2%
90	380	404	6.3%	394	3.7%	392	3.2%
100	490	533	8.8%	517	5.5%	507	3.5%
110	680	749	10.1%	719	5.7%	705	3.7%
(b) WB-20							
40	60	65	8.3%	64	6.7%	63	5.0%
50	100	109	9.0%	108	8.0%	105	5.0%
60	150	165	10.0%	162	8.0%	160	6.7%
70	200	220	10.0%	216	8.0%	214	7.0%
80	280	314	12.1%	311	11.1%	304	8.6%
90	380	434	14.2%	429	12.9%	421	10.8%
100	490	564	15.1%	554	13.1%	544	11.0%
110	680	785	15.4%	775	14.0%	770	13.2%
(c) NHTSA 3a / 2a							
40	60	63	5.0%	62	3.3%	62	3.3%
50	100	106	6.0%	105	5.0%	104	4.0%
60	150	161	7.3%	158	5.3%	157	4.7%
70	200	215	7.5%	213	6.5%	210	5.0%
80	280	314	12.1%	302	7.9%	298	6.4%
90	380	427	12.4%	411	8.2%	411	8.2%
100	490	552	12.7%	535	9.2%	532	8.6%
110	680	771	13.4%	751	10.4%	751	10.4%

Table A3: Required Increase in curve radius R_1 corresponding to different values of opposite curve radii R_2 for different design vehicles ($e_{max} = 0.04$) for level terrain.

Tangent Length, T = 100m

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 1.0 R1		R2 = 1.5 R1		R2 = 2.0 R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	60	61	1.7%	61	1.7%	60	0.0%
50	100	102	2.0%	101	1.0%	101	1.0%
60	150	153	2.0%	152	1.3%	151	0.7%
70	200	204	2.0%	204	2.0%	203	1.5%
80	280	287	2.5%	286	2.1%	285	1.8%
90	380	392	3.2%	389	2.4%	387	1.8%
100	490	507	3.5%	509	3.9%	500	2.0%
110	680	705	3.7%	708	4.1%	695	2.2%
(b) WB-20							
40	60	63	5.0%	62	3.3%	61	1.7%
50	100	105	5.0%	104	4.0%	102	2.0%
60	150	159	6.0%	156	4.0%	154	2.7%
70	200	213	6.5%	209	4.5%	206	3.0%
80	280	299	6.8%	294	5.0%	289	3.2%
90	380	407	7.1%	399	5.0%	394	3.7%
100	490	529	8.0%	521	6.3%	508	3.7%
110	680	754	10.9%	732	7.6%	707	4.0%
(c) NHTSA 3a / 2a							
40	60	61	1.7%	61	1.7%	60	0.0%
50	100	103	3.0%	102	2.0%	101	1.0%
60	150	156	4.0%	153	2.0%	153	2.0%
70	200	210	5.0%	206	3.0%	204	2.0%
80	280	295	5.4%	289	3.2%	286	2.1%
90	380	403	6.1%	393	3.4%	389	2.4%
100	490	521	6.3%	517	5.5%	504	2.9%
110	680	746	9.7%	727	6.9%	701	3.1%

Table A4: Required Increase in curve radius R_1 corresponding to different values of opposite curve radii R_2 for different design vehicles ($e_{max} = 0.04$) for level terrain.

Tangent Length, T = 200m

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 1.0 R1		R2 = 1.5 R1		R2 = 2.0 R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	60	60	0.0%	60	0.0%	60	0.0%
50	100	101	1.0%	100	0.0%	100	0.0%
60	150	152	1.3%	151	0.7%	150	0.0%
70	200	203	1.5%	202	1.0%	201	0.5%
80	280	286	2.1%	283	1.1%	282	0.7%
90	380	391	2.9%	386	1.6%	383	0.8%
100	490	506	3.3%	502	2.4%	495	1.0%
110	680	704	3.5%	702	3.2%	689	1.3%
(b) WB-20							
40	60	62	3.3%	61	1.7%	60	0.0%
50	100	104	4.0%	101	1.0%	101	1.0%
60	150	157	4.7%	152	1.3%	152	1.3%
70	200	211	5.5%	205	2.5%	203	1.5%
80	280	296	5.7%	289	3.2%	286	2.1%
90	380	405	6.6%	393	3.4%	389	2.4%
100	490	527	7.6%	518	5.7%	505	3.1%
110	680	754	10.9%	731	7.5%	705	3.7%
(c) NHTSA 3a / 2a							
40	60	61	1.7%	60	0.0%	60	0.0%
50	100	101	1.0%	100	0.0%	100	0.0%
60	150	155	3.3%	151	0.7%	150	0.0%
70	200	208	4.0%	203	1.5%	201	0.5%
80	280	293	4.6%	284	1.4%	282	0.7%
90	380	401	5.5%	391	2.9%	383	0.8%
100	490	517	5.5%	515	5.1%	501	2.2%
110	680	746	9.7%	726	6.8%	698	2.6%

Table A5: Required Increase in curve radius R_1 corresponding to different values of opposite curve radii R_2 for different design vehicles ($e_{max} = 0.04$) for level terrain.

Tangent Length, T = 250m

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 1.0 R1		R2 = 1.5 R1		R2 = 2.0 R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	60	60	0.0%	60	0.0%	60	0.0%
50	100	100	0.0%	100	0.0%	100	0.0%
60	150	150	0.0%	150	0.0%	150	0.0%
70	200	201	0.5%	200	0.0%	200	0.0%
80	280	282	0.7%	281	0.4%	280	0.0%
90	380	384	1.1%	382	0.5%	381	0.3%
100	490	496	1.2%	494	0.8%	492	0.4%
110	680	690	1.5%	687	1.0%	685	0.7%
(b) WB-20							
40	60	60	0.0%	60	0.0%	60	0.0%
50	100	101	1.0%	100	0.0%	100	0.0%
60	150	152	1.3%	151	0.7%	150	0.0%
70	200	203	1.5%	202	1.0%	201	0.5%
80	280	285	1.8%	283	1.1%	282	0.7%
90	380	388	2.1%	385	1.3%	383	0.8%
100	490	501	2.2%	497	1.4%	495	1.0%
110	680	698	2.6%	691	1.6%	688	1.2%
(c) NHTSA 3a / 2a							
40	60	60	0.0%	60	0.0%	60	0.0%
50	100	100	0.0%	100	0.0%	100	0.0%
60	150	151	0.7%	150	0.0%	150	0.0%
70	200	202	1.0%	201	0.5%	200	0.0%
80	280	283	1.1%	282	0.7%	281	0.4%
90	380	386	1.6%	383	0.8%	382	0.5%
100	490	498	1.6%	495	1.0%	493	0.6%
110	680	693	1.9%	688	1.2%	686	0.9%

Table A6: Required Increase in curve radius R_1 corresponding to different values of opposite curve radii R_2 for different design vehicles ($e_{max} = 0.06$) for level terrain.

Tangent Length, T = 0m

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 1.0 R1		R2 = 1.5 R1		R2 = 2.0 R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	55	59	7.3%	58	5.5%	57	3.6%
50	90	99	10.0%	96	6.7%	94	4.4%
60	130	145	11.5%	141	8.5%	137	5.4%
70	190	215	13.2%	209	10.0%	203	6.8%
80	250	286	14.4%	278	11.2%	270	8.0%
90	340	395	16.2%	386	13.5%	372	9.4%
100	440	525	19.3%	513	16.6%	496	12.7%
110	600	726	21.0%	715	19.2%	694	15.7%
(b) WB-20							
40	55	65	18.2%	64	16.4%	60	9.1%
50	90	107	18.9%	105	16.7%	100	11.1%
60	130	156	20.0%	155	19.2%	149	14.6%
70	190	232	22.1%	226	18.9%	219	15.3%
80	250	306	22.4%	299	19.6%	291	16.4%
90	340	422	24.1%	410	20.6%	398	17.1%
100	440	547	24.3%	534	21.4%	519	18.0%
110	600	751	25.2%	739	23.2%	725	20.8%
(c) NHTSA 3a / 2a							
40	55	62	12.7%	61	10.9%	58	5.5%
50	90	106	17.8%	102	13.3%	98	8.9%
60	130	154	18.5%	149	14.6%	142	9.2%
70	190	226	18.9%	219	15.3%	210	10.5%
80	250	298	19.2%	291	16.4%	281	12.4%
90	340	410	20.6%	399	17.4%	389	14.4%
100	440	536	21.8%	522	18.6%	509	15.7%
110	600	735	22.5%	728	21.3%	705	17.5%

Table A7: Required Increase in curve radius R_1 corresponding to different values of opposite curve radii R_2 for different design vehicles ($e_{max} = 0.06$) for level terrain.

Tangent Length, T = 50m

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 1.0 R1		R2 = 1.5 R1		R2 = 2.0 R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	55	57	3.6%	56	1.8%	56	1.8%
50	90	94	4.4%	93	3.3%	91	1.1%
60	130	137	5.4%	136	4.6%	132	1.5%
70	190	204	7.4%	200	5.3%	198	4.2%
80	250	271	8.4%	269	7.6%	261	4.4%
90	340	381	12.1%	370	8.8%	360	5.9%
100	440	508	15.5%	496	12.7%	482	9.5%
110	600	706	17.7%	696	16.0%	675	12.5%
(b) WB-20							
40	55	60	9.1%	59	7.3%	57	3.6%
50	90	101	12.2%	98	8.9%	94	4.4%
60	130	148	13.8%	145	11.5%	141	8.5%
70	190	218	14.7%	215	13.2%	209	10.0%
80	250	292	16.8%	284	13.6%	280	12.0%
90	340	402	18.2%	392	15.3%	384	12.9%
100	440	530	20.5%	510	15.9%	502	14.1%
110	600	728	21.3%	711	18.5%	702	17.0%
(c) NHTSA 3a / 2a							
40	55	58	5.5%	57	3.6%	56	1.8%
50	90	98	8.9%	96	6.7%	93	3.3%
60	130	146	12.3%	141	8.5%	136	4.6%
70	190	215	13.2%	209	10.0%	200	5.3%
80	250	285	14.0%	280	12.0%	270	8.0%
90	340	394	15.9%	388	14.1%	378	11.2%
100	440	517	17.5%	508	15.5%	491	11.6%
110	600	711	18.5%	707	17.8%	685	14.2%

Table A8: Required Increase in curve radius R_1 corresponding to different values of opposite curve radii R_2 for different design vehicles ($e_{max} = 0.06$) for level terrain.

Tangent Length, T = 100m

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 1.0 R1		R2 = 1.5 R1		R2 = 2.0 R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	55	56	1.8%	56	1.8%	55	0.0%
50	90	93	3.3%	92	2.2%	91	1.1%
60	130	135	3.8%	134	3.1%	131	0.8%
70	190	201	5.8%	198	4.2%	196	3.2%
80	250	269	7.6%	266	6.4%	259	3.6%
90	340	376	10.6%	366	7.6%	357	5.0%
100	440	498	13.2%	490	11.4%	475	8.0%
110	600	688	14.7%	681	13.5%	665	10.8%
(b) WB-20							
40	55	58	5.5%	57	3.6%	56	1.8%
50	90	97	7.8%	95	5.6%	93	3.3%
60	130	142	9.2%	141	8.5%	137	5.4%
70	190	211	11.1%	210	10.5%	204	7.4%
80	250	284	13.6%	277	10.8%	272	8.8%
90	340	392	15.3%	386	13.5%	374	10.0%
100	440	512	16.4%	502	14.1%	495	12.5%
110	600	704	17.3%	692	15.3%	684	14.0%
(c) NHTSA 3a / 2a							
40	55	57	3.6%	56	1.8%	55	0.0%
50	90	96	6.7%	93	3.3%	92	2.2%
60	130	140	7.7%	137	5.4%	134	3.1%
70	190	208	9.5%	201	5.8%	198	4.2%
80	250	277	10.8%	268	7.2%	268	7.2%
90	340	382	12.4%	371	9.1%	368	8.2%
100	440	504	14.5%	499	13.4%	488	10.9%
110	600	691	15.2%	686	14.3%	668	11.3%

Table A9: Required Increase in curve radius R_1 corresponding to different values of opposite curve radii R_2 for different design vehicles ($e_{max} = 0.06$) for level terrain.

Tangent Length, T = 200m

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 1.0 R1		R2 = 1.5 R1		R2 = 2.0 R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	55	56	1.8%	56	1.8%	55	0.0%
50	90	93	3.3%	92	2.2%	91	1.1%
60	130	135	3.8%	133	2.3%	131	0.8%
70	190	201	5.8%	198	4.2%	196	3.2%
80	250	267	6.8%	263	5.2%	259	3.6%
90	340	364	7.1%	359	5.6%	353	3.8%
100	440	473	7.5%	466	5.9%	457	3.9%
110	600	646	7.7%	636	6.0%	628	4.7%
(b) WB-20							
40	55	58	5.5%	57	3.6%	56	1.8%
50	90	95	5.6%	94	4.4%	93	3.3%
60	130	140	7.7%	136	4.6%	135	3.8%
70	190	206	8.4%	201	5.8%	199	4.7%
80	250	272	8.8%	268	7.2%	264	5.6%
90	340	371	9.1%	366	7.6%	361	6.2%
100	440	481	9.3%	474	7.7%	468	6.4%
110	600	657	9.5%	646	7.7%	639	6.5%
(c) NHTSA 3a / 2a							
40	55	57	3.6%	56	1.8%	55	0.0%
50	90	94	4.4%	92	2.2%	92	2.2%
60	130	139	6.9%	134	3.1%	134	3.1%
70	190	204	7.4%	198	4.2%	197	3.7%
80	250	269	7.6%	265	6.0%	261	4.4%
90	340	368	8.2%	361	6.2%	359	5.6%
100	440	478	8.6%	469	6.6%	465	5.7%
110	600	652	8.7%	640	6.7%	637	6.2%

Table A10: Required Increase in curve radius R_1 corresponding to different values of opposite curve radii R_2 for different design vehicles ($e_{max} = 0.06$) for level terrain.

Tangent Length, T = 250m

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 1.0 R1		R2 = 1.5 R1		R2 = 2.0 R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	55	56	1.8%	56	1.8%	55	0.0%
50	90	93	3.3%	92	2.2%	90	0.0%
60	130	135	3.8%	133	2.3%	130	0.0%
70	190	201	5.8%	198	4.2%	192	1.1%
80	250	267	6.8%	263	5.2%	254	1.6%
90	340	364	7.1%	359	5.6%	350	2.9%
100	440	473	7.5%	466	5.9%	453	3.0%
110	600	646	7.7%	636	6.0%	624	4.0%
(b) WB-20							
40	55	57	3.6%	56	1.8%	55	0.0%
50	90	94	4.4%	92	2.2%	91	1.1%
60	130	137	5.4%	134	3.1%	132	1.5%
70	190	204	7.4%	198	4.2%	194	2.1%
80	250	269	7.6%	265	6.0%	257	2.8%
90	340	367	7.9%	362	6.5%	355	4.4%
100	440	476	8.2%	469	6.6%	462	5.0%
110	600	650	8.3%	640	6.7%	631	5.2%
(c) NHTSA 3a / 2a							
40	55	56	1.8%	55	0.0%	55	0.0%
50	90	93	3.3%	92	2.2%	90	0.0%
60	130	135	3.8%	133	2.3%	130	0.0%
70	190	202	6.3%	198	4.2%	192	1.1%
80	250	268	7.2%	264	5.6%	255	2.0%
90	340	365	7.4%	360	5.9%	353	3.8%
100	440	474	7.7%	467	6.1%	459	4.3%
110	600	648	8.0%	638	6.3%	627	4.5%

Table A11: Required Increase in curve radius R_1 corresponding to different values of opposite curve radii R_2 for different design vehicles ($e_{max} = 0.06$) for level terrain.

Tangent Length, T = 300m

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 1.0 R1		R2 = 1.5 R1		R2 = 2.0 R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	55	55	0.0%	55	0.0%	55	0.0%
50	90	90	0.0%	90	0.0%	90	0.0%
60	130	131	0.8%	130	0.0%	130	0.0%
70	190	192	1.1%	191	0.5%	190	0.0%
80	250	253	1.2%	252	0.8%	251	0.4%
90	340	345	1.5%	343	0.9%	343	0.9%
100	440	448	1.8%	445	1.1%	445	1.1%
110	600	612	2.0%	607	1.2%	608	1.3%
(b) WB-20							
40	55	55	0.0%	55	0.0%	55	0.0%
50	90	91	1.1%	90	0.0%	90	0.0%
60	130	132	1.5%	131	0.8%	130	0.0%
70	190	194	2.1%	192	1.1%	191	0.5%
80	250	256	2.4%	254	1.6%	253	1.2%
90	340	349	2.6%	347	2.1%	345	1.5%
100	440	455	3.4%	451	2.5%	448	1.8%
110	600	621	3.5%	617	2.8%	612	2.0%
(c) NHTSA 3a / 2a							
40	55	55	0.0%	55	0.0%	55	0.0%
50	90	90	0.0%	90	0.0%	90	0.0%
60	130	131	0.8%	130	0.0%	130	0.0%
70	190	193	1.6%	191	0.5%	190	0.0%
80	250	254	1.6%	253	1.2%	251	0.4%
90	340	347	2.1%	345	1.5%	343	0.9%
100	440	450	2.3%	447	1.6%	446	1.4%
110	600	616	2.7%	611	1.8%	609	1.5%

Table A12: Required Increase in curve radius R_1 corresponding to different values of opposite curve radii R_2 for different design vehicles ($e_{max} = 0.06$) for level terrain.

Tangent Length, T = 350m

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 1.0 R1		R2 = 1.5 R1		R2 = 2.0 R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	55	55	0.0%	55	0.0%	55	0.0%
50	90	90	0.0%	90	0.0%	90	0.0%
60	130	130	0.0%	130	0.0%	130	0.0%
70	190	191	0.5%	190	0.0%	190	0.0%
80	250	252	0.8%	251	0.4%	250	0.0%
90	340	343	0.9%	342	0.6%	341	0.3%
100	440	445	1.1%	443	0.7%	442	0.5%
110	600	608	1.3%	605	0.8%	604	0.7%
(b) WB-20							
40	55	55	0.0%	55	0.0%	55	0.0%
50	90	90	0.0%	90	0.0%	90	0.0%
60	130	131	0.8%	130	0.0%	130	0.0%
70	190	192	1.1%	191	0.5%	190	0.0%
80	250	253	1.2%	252	0.8%	251	0.4%
90	340	345	1.5%	343	0.9%	342	0.6%
100	440	447	1.6%	445	1.1%	443	0.7%
110	600	610	1.7%	608	1.3%	606	1.0%
(c) NHTSA 3a / 2a							
40	55	55	0.0%	55	0.0%	55	0.0%
50	90	90	0.0%	90	0.0%	90	0.0%
60	130	130	0.0%	130	0.0%	130	0.0%
70	190	191	0.5%	190	0.0%	190	0.0%
80	250	252	0.8%	251	0.4%	250	0.0%
90	340	343	0.9%	342	0.6%	341	0.3%
100	440	445	1.1%	444	0.9%	442	0.5%
110	600	609	1.5%	606	1.0%	605	0.8%

Table A13: Required Increase in curve radius R_1 for reverse curves combined with vertical alignment corresponding to different values of opposite curve radii R_2 for different design vehicles ($e_{max} = 0.06$).

Tangent Length, $T = 0m$

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 1.0R1		R2 = 1.5 R1		R2 =2.0 R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	55	62	12.7%	61	10.9%	60	9.1%
50	90	104	15.6%	100	11.1%	98	8.9%
60	130	151	16.2%	147	13.1%	143	10.0%
70	190	221	16.3%	218	14.7%	210	10.5%
80	250	293	17.2%	287	14.8%	283	13.2%
90	340	402	18.2%	391	15.0%	389	14.4%
100	440	534	21.4%	526	19.5%	511	16.1%
110	600	744	24.0%	730	21.7%	712	18.7%
(b) WB-20							
40	55	69	25.5%	67	21.8%	64	16.4%
50	90	112	24.4%	110	22.2%	105	16.7%
60	130	163	25.4%	159	22.3%	155	19.2%
70	190	242	27.4%	233	22.6%	229	20.5%
80	250	319	27.6%	310	24.0%	302	20.8%
90	340	439	29.1%	422	24.1%	411	20.9%
100	440	560	27.3%	550	25.0%	538	22.3%
110	600	764	27.3%	761	26.8%	739	23.2%
(c) NHTSA 3a / 2a							
40	55	67	21.8%	65	18.2%	62	12.7%
50	90	109	21.1%	107	18.9%	103	14.4%
60	130	159	22.3%	159	22.3%	149	14.6%
70	190	234	23.2%	228	20.0%	221	16.3%
80	250	316	26.4%	304	21.6%	292	16.8%
90	340	435	27.9%	420	23.5%	402	18.2%
100	440	557	26.6%	547	24.3%	523	18.9%
110	600	761	26.8%	754	25.7%	723	20.5%

Table A14: Required Increase in curve radius R_1 for reverse curves combined with vertical alignment corresponding to different values of opposite curve radii R_2 for different design vehicles ($e_{max} = 0.06$).

Tangent Length, T = 50m

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 1.0R1		R2 = 1.5 R1		R2 =2.0 R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	55	59	7.3%	58	5.5%	56	1.8%
50	90	97	7.8%	95	5.6%	92	2.2%
60	130	141	8.5%	139	6.9%	133	2.3%
70	190	208	9.5%	204	7.4%	199	4.7%
80	250	277	10.8%	271	8.4%	264	5.6%
90	340	383	12.6%	377	10.9%	361	6.2%
100	440	504	14.5%	490	11.4%	474	7.7%
110	600	698	16.3%	671	11.8%	657	9.5%
(b) WB-20							
40	55	65	18.2%	61	10.9%	59	7.3%
50	90	108	20.0%	101	12.2%	97	7.8%
60	130	158	21.5%	149	14.6%	144	10.8%
70	190	232	22.1%	218	14.7%	211	11.1%
80	250	307	22.8%	289	15.6%	279	11.6%
90	340	425	25.0%	392	15.3%	382	12.4%
100	440	553	25.7%	515	17.0%	497	13.0%
110	600	756	26.0%	712	18.7%	683	13.8%
(c) NHTSA 3a / 2a							
40	55	61	10.9%	60	9.1%	58	5.5%
50	90	102	13.3%	99	10.0%	95	5.6%
60	130	149	14.6%	144	10.8%	139	6.9%
70	190	221	16.3%	211	11.1%	205	7.9%
80	250	302	20.8%	283	13.2%	271	8.4%
90	340	421	23.8%	390	14.7%	380	11.8%
100	440	549	24.8%	511	16.1%	494	12.3%
110	600	751	25.2%	706	17.7%	681	13.5%

Table A15: Required Increase in curve radius R_1 for reverse curves combined with vertical alignment corresponding to different values of opposite curve radii R_2 for different design vehicles ($e_{max} = 0.06$).

Tangent Length, T = 100m

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 1.0R1		R2 = 1.5 R1		R2 =2.0 R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	55	58	5.5%	57	3.6%	56	1.8%
50	90	96	6.7%	94	4.4%	92	2.2%
60	130	139	6.9%	136	4.6%	132	1.5%
70	190	204	7.4%	199	4.7%	197	3.7%
80	250	270	8.0%	262	4.8%	260	4.0%
90	340	370	8.8%	364	7.1%	355	4.4%
100	440	484	10.0%	476	8.2%	463	5.2%
110	600	673	12.2%	651	8.5%	633	5.5%
(b) WB-20							
40	55	62	12.7%	59	7.3%	58	5.5%
50	90	102	13.3%	97	7.8%	95	5.6%
60	130	149	14.6%	142	9.2%	139	6.9%
70	190	224	17.9%	211	11.1%	206	8.4%
80	250	293	17.2%	279	11.6%	272	8.8%
90	340	402	18.2%	380	11.8%	373	9.7%
100	440	522	18.6%	496	12.7%	486	10.5%
110	600	720	20.0%	679	13.2%	669	11.5%
(c) NHTSA 3a / 2a							
40	55	59	7.3%	58	5.5%	57	3.6%
50	90	98	8.9%	95	5.6%	93	3.3%
60	130	144	10.8%	140	7.7%	136	4.6%
70	190	220	15.8%	207	8.9%	201	5.8%
80	250	291	16.4%	273	9.2%	269	7.6%
90	340	400	17.6%	378	11.2%	371	9.1%
100	440	519	18.0%	494	12.3%	483	9.8%
110	600	714	19.0%	678	13.0%	666	11.0%

Table A16: Required Increase in curve radius R_1 for reverse curves combined with vertical alignment corresponding to different values of opposite curve radii R_2 for different design vehicles ($e_{max} = 0.06$).

Tangent Length, T = 200m

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 1.0R1		R2 = 1.5 R1		R2 =2.0 R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	55	57	3.6%	56	1.8%	55	0.0%
50	90	94	4.4%	92	2.2%	91	1.1%
60	130	136	4.6%	134	3.1%	132	1.5%
70	190	199	4.7%	197	3.7%	195	2.6%
80	250	262	4.8%	260	4.0%	257	2.8%
90	340	360	5.9%	359	5.6%	351	3.2%
100	440	470	6.8%	466	5.9%	456	3.6%
110	600	655	9.2%	637	6.2%	623	3.8%
(b) WB-20							
40	55	60	9.1%	58	5.5%	57	3.6%
50	90	99	10.0%	95	5.6%	93	3.3%
60	130	144	10.8%	138	6.2%	135	3.8%
70	190	212	11.6%	206	8.4%	201	5.8%
80	250	281	12.4%	272	8.8%	267	6.8%
90	340	386	13.5%	371	9.1%	366	7.6%
100	440	502	14.1%	483	9.8%	477	8.4%
110	600	691	15.2%	661	10.2%	654	9.0%
(c) NHTSA 3a / 2a							
40	55	58	5.5%	57	3.6%	56	1.8%
50	90	96	6.7%	94	4.4%	92	2.2%
60	130	141	8.5%	136	4.6%	133	2.3%
70	190	211	11.1%	201	5.8%	197	3.7%
80	250	280	12.0%	270	8.0%	264	5.6%
90	340	383	12.6%	370	8.8%	365	7.4%
100	440	498	13.2%	481	9.3%	472	7.3%
110	600	691	15.2%	659	9.8%	651	8.5%

Table A17: Required Increase in curve radius R_1 for reverse curves combined with vertical alignment corresponding to different values of opposite curve radii R_2 for different design vehicles ($e_{max} = 0.06$).

Tangent Length, T = 250m

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 1.0R1		R2 = 1.5 R1		R2 =2.0 R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	55	55	0.0%	55	0.0%	55	0.0%
50	90	91	1.1%	90	0.0%	90	0.0%
60	130	134	3.1%	131	0.8%	130	0.0%
70	190	196	3.2%	193	1.6%	192	1.1%
80	250	260	4.0%	255	2.0%	254	1.6%
90	340	354	4.1%	347	2.1%	346	1.8%
100	440	459	4.3%	451	2.5%	449	2.0%
110	600	627	4.5%	621	3.5%	612	2.0%
(b) WB-20							
40	55	57	3.6%	56	1.8%	55	0.0%
50	90	95	5.6%	93	3.3%	91	1.1%
60	130	136	4.6%	135	3.8%	133	2.3%
70	190	202	6.3%	200	5.3%	196	3.2%
80	250	267	6.8%	263	5.2%	258	3.2%
90	340	362	6.5%	357	5.0%	353	3.8%
100	440	471	7.0%	464	5.5%	458	4.1%
110	600	650	8.3%	635	5.8%	631	5.2%
(c) NHTSA 3a / 2a							
40	55	56	1.8%	55	0.0%	55	0.0%
50	90	94	4.4%	91	1.1%	90	0.0%
60	130	135	3.8%	133	2.3%	131	0.8%
70	190	200	5.3%	196	3.2%	192	1.1%
80	250	264	5.6%	260	4.0%	256	2.4%
90	340	359	5.6%	354	4.1%	349	2.6%
100	440	468	6.4%	460	4.5%	452	2.7%
110	600	642	7.0%	630	5.0%	624	4.0%

Table A18: Required Increase in curve radius R_1 for reverse curves combined with vertical alignment corresponding to different values of opposite curve radii R_2 for different design vehicles ($e_{max} = 0.06$).

Tangent Length, T = 300m

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 1.0R1		R2 = 1.5 R1		R2 =2.0 R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	55	55	0.0%	55	0.0%	55	0.0%
50	90	90	0.0%	90	0.0%	90	0.0%
60	130	131	0.8%	130	0.0%	130	0.0%
70	190	193	1.6%	191	0.5%	190	0.0%
80	250	256	2.4%	253	1.2%	251	0.4%
90	340	347	2.1%	344	1.2%	342	0.6%
100	440	450	2.3%	447	1.6%	444	0.9%
110	600	616	2.7%	610	1.7%	607	1.2%
(b) WB-20							
40	55	56	1.8%	55	0.0%	55	0.0%
50	90	92	2.2%	91	1.1%	90	0.0%
60	130	134	3.1%	133	2.3%	131	0.8%
70	190	197	3.7%	194	2.1%	192	1.1%
80	250	260	4.0%	256	2.4%	253	1.2%
90	340	355	4.4%	352	3.5%	348	2.4%
100	440	459	4.3%	456	3.6%	449	2.0%
110	600	628	4.7%	622	3.7%	614	2.3%
(c) NHTSA 3a / 2a							
40	55	55	0.0%	55	0.0%	55	0.0%
50	90	91	1.1%	90	0.0%	90	0.0%
60	130	132	1.5%	131	0.8%	130	0.0%
70	190	195	2.6%	193	1.6%	191	0.5%
80	250	257	2.8%	254	1.6%	252	0.8%
90	340	351	3.2%	348	2.4%	344	1.2%
100	440	454	3.2%	451	2.5%	446	1.4%
110	600	622	3.7%	617	2.8%	611	1.8%

Table A19: Required Increase in curve radius R_1 for reverse curves combined with vertical alignment corresponding to different values of opposite curve radii R_2 for different design vehicles ($e_{max} = 0.06$).

Tangent Length, T = 350m

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 1.0R1		R2 = 1.5 R1		R2 =2.0 R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	55	55	0.0%	55	0.0%	55	0.0%
50	90	90	0.0%	90	0.0%	90	0.0%
60	130	130	0.0%	130	0.0%	130	0.0%
70	190	190	0.0%	190	0.0%	190	0.0%
80	250	251	0.4%	250	0.0%	250	0.0%
90	340	343	0.9%	341	0.3%	340	0.0%
100	440	446	1.4%	443	0.7%	441	0.2%
110	600	610	1.7%	605	0.8%	602	0.3%
(b) WB-20							
40	55	55	0.0%	55	0.0%	55	0.0%
50	90	90	0.0%	90	0.0%	90	0.0%
60	130	131	0.8%	130	0.0%	130	0.0%
70	190	193	1.6%	191	0.5%	190	0.0%
80	250	256	2.4%	254	1.6%	252	0.8%
90	340	349	2.6%	346	1.8%	343	0.9%
100	440	451	2.5%	448	1.8%	445	1.1%
110	600	616	2.7%	611	1.8%	607	1.2%
(c) NHTSA 3a / 2a							
40	55	55	0.0%	55	0.0%	55	0.0%
50	90	90	0.0%	90	0.0%	90	0.0%
60	130	130	0.0%	130	0.0%	130	0.0%
70	190	192	1.1%	190	0.0%	190	0.0%
80	250	255	2.0%	252	0.8%	250	0.0%
90	340	347	2.1%	344	1.2%	341	0.3%
100	440	450	2.3%	446	1.4%	444	0.9%
110	600	614	2.3%	609	1.5%	606	1.0%

Table A20: Required Increase in curve radius R_1 for reverse curves combined with vertical alignment corresponding to different values of opposite curve radii R_2 for different design vehicles ($e_{max} = 0.06$).

Tangent Length, T = 400m

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 1.0R1		R2 = 1.5 R1		R2 =2.0 R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	55	55	0.0%	55	0.0%	55	0.0%
50	90	90	0.0%	90	0.0%	90	0.0%
60	130	130	0.0%	130	0.0%	130	0.0%
70	190	190	0.0%	190	0.0%	190	0.0%
80	250	250	0.0%	250	0.0%	250	0.0%
90	340	341	0.3%	340	0.0%	340	0.0%
100	440	443	0.7%	441	0.2%	440	0.0%
110	600	604	0.7%	602	0.3%	601	0.2%
(b) WB-20							
40	55	55	0.0%	55	0.0%	55	0.0%
50	90	90	0.0%	90	0.0%	90	0.0%
60	130	130	0.0%	130	0.0%	130	0.0%
70	190	190	0.0%	190	0.0%	190	0.0%
80	250	251	0.4%	250	0.0%	250	0.0%
90	340	343	0.9%	341	0.3%	340	0.0%
100	440	447	1.6%	444	0.9%	441	0.2%
110	600	610	1.7%	605	0.8%	603	0.5%
(c) NHTSA 3a / 2a							
40	55	55	0.0%	55	0.0%	55	0.0%
50	90	90	0.0%	90	0.0%	90	0.0%
60	130	130	0.0%	130	0.0%	130	0.0%
70	190	190	0.0%	190	0.0%	190	0.0%
80	250	250	0.0%	250	0.0%	250	0.0%
90	340	342	0.6%	341	0.3%	340	0.0%
100	440	445	1.1%	443	0.7%	440	0.0%
110	600	607	1.2%	604	0.7%	601	0.2%

APPENDIX B

STATISTICAL OUTPUT FOR FLAT AND 3D REVERSE CURVE MODELS WITH INTERMEDIATE TANGENTS

Table B1: SPSS Model Output – Independent Variables entered into Regression Model

Variables Entered/Removed ^b			
Model	Variables Entered	Variables Removed	Method
1	WB20, RATIO, SPEED, SUPER, TANGENT ^a , WB15		Enter

a. All requested variables entered.

b. Dependent Variable: RADIUS

Table B2: SPSS Model Output – Model Summary and Goodness of Fit

Model	R	R Square	Adjusted R Square	Std. Error of The Estimate	R Square Change	Change Statistics			
						F Change	df1	df2	Sig. F Change
1	.993 ^a	0.986	0.986	24.86447	0.986	5457.693	6	1289	0

Table B3: SPSS Model Output –ANOVA Table

ANOVA ^b					
Model		Sum of Squares	df	Mean Square	Sig.
1	Regression	57339565	6	9556594.243	.000 ^a
	Residual	796913.9	1289	618.242	
	Total	58136479	1295		

a. Predictors: (Constant), WB20, RATIO, SPEED, SUPER, TANGENT, WB15

b. Dependent Variable: RADIUS

Table B4: SPSS Model Output – Regression Coefficients

Model	Unstandardized Coefficients		Stand. Coefficients	t	Sig.
	B	Std. Error	Beta		
Const.	93.953	3.066		30.644	0
T	-0.13	0.006	-0.071	-21.144	0
I _e	-24.296	1.422	-0.057	-17.082	0
V _D	0.099	0	0.988	302.825	0
r	-11.255	1.692	-0.022	-6.652	0
WB15	- 6.144	1.692	0.014	3.631	0
WB20	7.484	1.692	-0.017	-4.424	0

Figure B1: SPSS Model Output – Histogram of Regression Standardized Residuals

Figure B2: SPSS Model Output – Normal P-P Plot of Regression

Table B5: SPSS Model Output – Independent Variables entered into Regression Model

Variables Entered/Removed^b			
Model	Variables Entered	Variables Removed	Method
1	wb20, g2, ratio, vd, t, e, g1, a, wb15		Enter

a. All requested variables entered.

b. Dependent Variable: r

Table B6: SPSS Model Output – Model Summary and Goodness of Fit

Model Summary ^b

Model	R	R Square	Adjusted R Square	Std. Error of the Estimate
1	.992 ^a	.983	.983	27.09278

a. Predictors: (Constant), wb20, g2, ratio, vd, t, e, g1, wb15

b. Dependent Variable: r

Table B7: SPSS Regression Model Output

Model	Unstand. Coefficients		Standardized Coefficients	P - Value (95% CI)
	B	Std. Error	BETA	
Const.	154.863	2.728		0
V _D	0.098	0	0.985	0
t	-0.137	0.003	-0.084	0
ratio	-14.514	0.942	-0.028	0
le	-13.306	0.399	-0.063	0
g1	-0.773	0.105	-0.014	0
g2	-0.484	0.094	-0.01	0
wb15	-7.283	0.935	-0.016	0
wb20	2.789	0.467	0.013	0

Figure B3: SPSS Model Output – Histogram of Regression Standardized Residuals

Normal P-P Plot of Regression Standardized Residual

Figure B4: Normal P-P Plot of Regression Standardized Residuals