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VEHICLE STABILITY ON COMBINED HORIZONTAL AND VERTICAL ALIGNMENTS

by

GLEN FURTADO, BEng, EIT

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

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**VEHICLE STABILITY ON COMBINED HORIZONTAL
AND VERTICAL ALIGNMENTS**

Submitted by:

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in partial fulfillment of the requirements
for the degree of
Master of Applied Science in
Civil Engineering

Thesis Co-Supervisor
Dr. Said Easa
Ryerson University

Thesis Co-Supervisor
Dr. A.O. Abd El Halim
Carleton University

Chair, Dept. of Civil and Environmental Engineering
Dr. Wayne Parker
Carleton University

ABSTRACT

This study examines vehicle stability through lateral acceleration levels using computer simulation. The objective was to compare the minimum flat horizontal curve radius used in traditional geometric design guides with the minimum radius based on vehicle dynamics on three-dimensional (3D) alignments. A vehicle dynamics simulation computer program was used to examine vehicle dynamics and stability on 3D alignments. The program uses a sophisticated two-axle dynamics model that traditionally has only been used by mechanical engineers in experimenting with vehicle characteristics.

It was found that vehicle stability is not compromised on 3D alignments compared to 2D alignments for the test cases examined. Differences in the required minimum horizontal radii were obtained due to the transient effects of a vehicle traveling along a horizontal curve. These transient effects due to changes in road geometry include driver steering, vehicle off tracking, and vehicle roll. In order to maintain the currently acceptable comfort thresholds, increases up to 16.6 % of the minimum horizontal radii suggested by the current North American geometric design guides are required.

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Finally, special thanks to my parents and family for all their love and support over the course of my education. This work is dedicated to my brother John who has never refused to lend an ear.

“If I have been able to see a little further than some others, it was because I stood on the shoulders of giants.”

Sir Isaac Newton (1642-1727)

in *Philosophiae Naturalis Principia Mathematica*

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CHAPTER 1

1.0 INTRODUCTION

There is a strong relationship between traffic safety and geometric design. Geometric design can be defined as the proportioning of various visible elements of the highway (Easa 2002). North American geometric design practices currently consider highway alignments in the design process individually. This is due to the complexity involved in evaluating and designing alignments in three-dimensions (3D). Referring to the 3D geometric alignment, Smith and Lamm (1994) state that it is "...a very complex component in the highway geometric design process...". However, neglecting the 3D nature of geometric highway alignments can result in design inconsistencies, which potentially can produce delay, increase construction and maintenance costs, and perhaps more importantly, reduce user safety.

Transportation infrastructure, or more specifically the road network, plays an important role in the Canadian economy. Almost \$1 trillion worth of goods were moved along Ontario's roads alone in 1999 (Hamilton 2001). Roads connect other major transportation hubs such as ports, airports and train stations and are vital for the movement of goods and people. In 1995, the Canadian road network contained over 901,000 two-lane equivalent kilometers of road and in 1992, there were 1.25 million vehicles involved in reportable accidents, including both casualty and property-damage collisions (TAC 1995). More specifically, over 50% of the fatalities that occur on rural two-lane highways are the result of accidents on curved sections of the highway (Lamm

1992). Given these significant figures it is conceivable why road researchers have focused their efforts on achieving safer, more consistent geometric design standards and guidelines.

This chapter provides background information on geometric alignment elements in Section 1.1 and covers some fundamental highway geometry in Section 1.2. Section 1.3 outlines the research problem to be addressed while Section 1.4 states the research objectives and organizational structure of the remaining chapters within this document.

1.1 Background

Human factors play an important role in geometric design since ultimately, roads are built for use by people. There are limitations to how far people can see, how fast they can react and how they perceive and process information. In addition to these limitations, it is also important to realize that these human traits vary from person to person depending on age, sex and other characteristics (TAC 1999).

Perception-reaction time (PRT) is perhaps the best-documented human factor measure in current design practice. This PRT is comprised of four elements: perception, identification, evaluation and reaction. By accounting for human factors in roadway design, driver performance should theoretically be error-free. This is true provided that driver expectations have not been surpassed or violated and that a consistent design has been blended into the final product (TAC 1999).

There are many different elements in geometric design such as sight distance, horizontal alignment, vertical alignment, cross section, drainage, lighting, and traffic control devices to name a few. Each of these elements can be further divided into particulars. For example, sight distance can be broken down into stopping sight distance, decision sight distance, headlight sight distance and passing sight distance. Similarly, superelevation, curve radius, spiral transition and side friction factors are all characteristics that must be considered when designing the horizontal alignment. Vertical alignment is extremely sensitive to the natural topography and as a result careful consideration must be given to maximum and minimum gradients.

The large number of elements involved in geometric design is perhaps one reason why traditional North American design practice has been to consider them individually or one at a time. Highways are three-dimensional (3D) in nature and previous research in geometric design has not accounted for this (Gibreel et al. 2001).

Each element is typically considered in two-dimensions (2D) with little consideration for how decisions made in one process can effect another. More recently however, significant advancements in 3D geometric design and consistency evaluation have been made by Hassan and Easa (2000, 2001) in the areas of sight distance and operating speed. However, research is still lacking in other areas relating to 3D geometric design.

1.2 Highway Geometry

The physical geometry of a highway directly affects the operational aspects and safety of the facility. Two main elements that define highway geometry are horizontal and vertical alignments. The horizontal alignment is a ground plan that shows a route while the vertical alignment shows elevation (longitudinal grades along the highway and curves). Highway alignments are commonly illustrated as plans and profiles with each providing their respective two-dimensional (2D) representation of the given roadway. Another main highway element is the cross section, which defines lane, shoulder and clear zone width, superelevation, the number of lanes and identifies the presence of ditches or other drainage facilities.

Both horizontal and vertical alignments contain tangents and curves. Horizontal curves refer to the circular paths a vehicle moves while vertical curves describe a parabolic path. There are four basic types of horizontal curves: simple curves, compound curves, reverse curves and transition or spiral curves. The curves are normally connected to tangents even though a curvilinear alignment, by definition, does not necessarily require the existence of tangents. Vertical curves, which can be either of the sag or crest curve variety, are also typically connected to tangents. Any roadway, regardless of its origin and destination will have a horizontal and vertical alignment and it is through the coordination of these alignments that a designer can produce a safe, operationally efficient highway.

1.2.1 Vehicle Stability

A vehicle experiences centripetal acceleration towards the centre of the horizontal curve as it navigates a circular path. Its stability along this path is dependent on the resistive forces that counteract the centripetal acceleration. These resistive forces include frictional interaction between the tires and pavement, and a component of the vehicle weight that acts perpendicular to any roadway superelevation.

The term centripetal acceleration is often interchanged with lateral acceleration when referring to horizontal curve design (AASHTO 2001). This lateral acceleration can be thought of as the friction demand acting on the vehicle. The resistive forces are analogous to friction supply. When demand exceeds supply a vehicle is unstable and either skids off the roadway or rolls over (Morrall 1994). There is a basic equation stemming from the laws of mechanics, which will be discussed in greater detail in a subsequent chapter that describes vehicle operation on curves. For now, it is sufficient to mention that the variables involved include superelevation, vehicle speed, curve radius, acceleration due to gravity and a side friction factor.

Values for the side friction factor are limited by current design guides based on research that was conducted many years ago, which suggested that maximum side friction factors should be associated with human comfort (Barnett 1936; Moyer, Berry 1940). This essentially serves to limit the roadway friction supply. However, with little consideration of the friction demand imposed by the highway, the margin of safety on

horizontal curves is questionable. In addition, the current side friction factors were established without the consideration of the potential effect of superimposed or combined alignments.

1.2.2 Combined Alignments

A combined alignment is simply the overlap of horizontal and vertical alignments. On its own a vertical curve, or more generally a vertical alignment, does not have the stability issues typically associated with horizontal curves and alignments although maintaining comfort control can be a problem. Under the vast majority of operating conditions, a vehicle will not skid or roll over when navigating a vertical curve on a horizontal tangent. The component of vehicle weight acting perpendicular to the pavement on a pure vertical alignment is large since it is parallel to the gravitational force of the Earth. However, if a vehicle travels on a combined alignment, the 3D geometry may induce different centripetal forces than would occur on 2D horizontal alignments potentially resulting in vehicle stability problems through increased friction demand and or reduced friction supply.

Morrall (1994) states “Surprisingly, very little research has been conducted into determining the maximum coefficient of side friction a vehicle is capable of obtaining on a horizontal curve.”. To date, even less research work has been completed on side friction coefficient maximums on combined horizontal and vertical highway alignments.

1.3 Problem Definition

The use and adherence to current North American geometric design guides does not guarantee the production of a consistent, adequate alignment. Perhaps the most significant problem is that current guidelines address each geometric element separately rather than collectively in a 3D analysis (Hassan 1996). Certain combined horizontal and vertical alignments can result in highway geometry that lacks consistency with driver expectations or is potentially unsafe.

It is worthwhile at this point to briefly define what is meant by consistent or consistency when referring to geometric road design. Road consistency can be defined as the degree to which the various geometric elements are integrated together to avoid critical or unexpected driving maneuvers (Al-Masaeid et al. 1995). If a designer can provide drivers with a sequential, coordinated set of geometric elements that are consistent with driver expectations then the result should be a safer, more efficient alignment. A well-designed facility must incorporate safety and mobility while respecting the associated human and environmental impacts that will result following its construction (FHWA 2001).

Superimposing a vertical alignment over a horizontal alignment without considering the consequences is not promoted in either the American Association of State Highway and Transportation Officials (AASHTO) Policy on Geometric Design of Highways and Streets (2001) or the Transportation Association of Canada's (TAC) Geometric Design

Guide for Canadian Roads (1999) design guides. However, designers with little experience may resort to ensuring that the highway element in question satisfies the published minimum or maximum criteria. Worldwide, over 500,000 deaths transpire annually as a result of vehicle accidents with over 30% of these fatalities occurring on curved sections of rural two-lane highways (Lamm et al. 1994). Given these staggering figures, and not withstanding human error as a cause, it is apparent that there are shortcomings to highway design standards, assuming that roadway alignments are designed to satisfy current guideline criteria.

Minimum horizontal curve radius is related to lateral acceleration and side friction through the point-mass formula in North American geometric design guides (AASHTO 2001, TAC 1999). Although limited research has been conducted on vehicle-road interaction on grades, a detailed extension to combined alignments has not been documented (Dunlap et al. 1978, Kontaratos et al. 1994). For a given speed, current road design theory determines a suitable minimum horizontal curve radius based on the following assumptions (Kontaratos et al. 1994):

1. Vehicle characteristics are simplified as a point-mass.
2. The road surface is considered in 2D as a plain curved surface.
3. Vehicle motion is governed by side friction values established in the design guides.

The concerns associated with the current geometric design guides as related to vehicle stability on combined alignments can be summarized as follows:

1. Current design standards recommend minimum curve radius based on a 2D projection of the highway.
2. Lateral acceleration on combined alignments may result in different friction demand/supply altering recommended/current design values.
3. Simplified 2D design standards may not guarantee a satisfactory highway design (Hassan 1996).
4. Current values of minimum horizontal curve radius may not ensure design consistency between successive alignment elements.

There are many other factors that compound the concerns listed above. These include pavement condition such as coarse or polished running surface, environmental conditions such as rain, snow or ice, and numerous human factors such as driver workload, PRT and driver expectation. The contribution of these factors to vehicle stability is very important. However, they are beyond the scope of this research and will not be considered in detail.

1.4 Research Objectives and Thesis Organization

The main objectives of this research are as follows:

1. To review previous studies and research on vehicle stability with particular emphasis on combined horizontal and vertical alignments.
2. To identify potential deficiencies in current geometric design practice as related to selection of minimum horizontal curve radius.

3. Determine through computer simulation, differences in lateral acceleration on plain 2D alignments versus 3D alignments.
4. To compare the radius of flat horizontal curves used in traditional design guides with the radius obtained based on 3D vehicle dynamics on 2D and 3D alignments.
5. To make preliminary recommendations regarding design requirements for the minimum horizontal curve radius on 3D alignments.
6. To identify areas in which future geometric design research relating to combined alignments and stability may be warranted.

The scope of this research is limited, however, to the design of rural highways and it will not account for human factors, environmental or entire vehicle fleet conditions that exist on North American highways. In addition, limiting values for the side friction factors that exist in the current design guides will not be questioned although consideration of recent research will be discussed (Morrall et al. 1994, Lamm et al. 1991, 1990).

The chapters of this thesis are organized in the following manner:

Chapter 2 contains a detailed literature review of vertical and horizontal alignments, vehicle stability and dynamic vehicle models. Consistent with the research objectives, Chapter 2 focuses on vehicle stability and the discussion includes items such as superelevation, sprung and rigid vehicle models, human comfort and side friction factors. An overview of current AASHTO and TAC design guides is also provided and

topics such as transition design and distribution of superelevation and side friction are discussed. Chapter 2 will also explain the current philosophy relating to the coordination of horizontal and vertical alignments and briefly touch on highway aesthetics.

Chapter 3 documents the data collection process and the modeling procedure used in examining vehicle stability on combined alignments. It includes a description of the software used and discusses the associated output. This chapter highlights the differences in vehicle lateral acceleration between 2D and 3D alignments with emphasis on how these differences affect the minimum horizontal curve radius. The method of model calibration is also discussed.

Chapter 4 presents the data analysis. A comparison is made between the minimum horizontal curve radii based on the dynamics output of 2D versus 3D alignments. The differences in the resulting radii are presented both graphically and in a tabular form to emphasize the potential stability concerns associated with considering horizontal and vertical alignments independently of one another.

Chapter 5 presents the main findings of this research. The main conclusions are discussed and areas pertaining to vehicle stability on combined alignments requiring further research are highlighted.

To provide an overview of the thesis, a flowchart showing the research organization is presented in Figure 1.1.

Figure 1.1: Research Overview

CHAPTER 2

2.0 LITERATURE REVIEW

2.1 Background

In this chapter a review of horizontal and vertical alignments and cross-sectional elements is conducted since these three major components are integrated together to produce a combined alignment. The chapter is divided into seven major Sections with several subsections. Section 2.2 begins by describing basic highway principles and illustrates the three-dimensional (3D) nature of highway alignments. Section 2.3 provides an explanation of the theoretical considerations for vertical alignments. Tangents, vertical curves and the various sight distance criteria are discussed. Section 2.4 reviews horizontal alignments and their characteristics.

Vehicle stability is discussed in some detail through the inclusion of topics such as superelevation, sprung and rigid vehicle models, human comfort and side friction in Section 2.5. Section 2.6 briefly covers AASHTO (2001) and TAC (1999) design guides focusing on the practical application of the guides toward designing a roadway. Section 2.7 examines vehicle dynamics in more detail by presenting three dynamic models in a more formal manner. This Section examines the bicycle and two-axle models and then describes the Interactive Highway Safety Design Model (IHSDM) – Vehicle Dynamics Model (VDM). Model capabilities as well as limitations are also described.

2.2 Highway Alignment Principles

There are three major elements involved in the geometric design of highways. These elements are:

1. Vertical alignment
2. Horizontal alignment
3. Cross section

Every highway in the world, regardless of natural topography, origin, destination, latitude or longitude will contain all three of these elements. Figure 2.1 shows how the alignment of a highway is a 3D problem (Mannering et al. 1998). While several researchers have documented the need for a 3D geometric design procedure or admitted that the complex interaction among elements is not yet fully understood, work in

Figure 2.1: A Highway Alignment in Three Dimensions (Mannering et al. 1998)

achieving a complete new procedure is still in its infancy (Hassan 1996, Cleveland 1994, Gibreel et al. 2001).

Mannering and Kilareski (1998) describe 3D design calculations as “cumbersome”. In addition, they suggest that construction of a highway facility based on 3D coordinates is another reason for the suppression of a 3D design procedure by current design guides. As a result, the 3D alignment is commonly simplified into two 2D alignments, namely a plan which represents the horizontal alignment and a profile, which corresponds to the vertical alignment. Figure 2.2 shows the same alignment configuration from Figure 2.1 represented as two 2D alignments.

Figure 2.2: Highway Alignment Plan and Profile Views (Mannering et al. 1998)

2.3 Vertical and Horizontal Sight Distance

Providing adequate sight distance is key to the safety of rural two-lane highways. A recent study showed that short sight distances in advance of sharp horizontal curves adversely affect roadway safety (Fink and Krammes 1995). Moreover, inadequate sight distances on combined alignments may lead to driver error and compromised safety due to an inaccurate perception of the road ahead (Hassan and Easa 2001).

The properties of a parabola are such that the sight distance remains unchanged over the entire length of curve providing that the curve is longer than the sight distance requirement. This important characteristic of parabolas is why they were favored as vertical design elements over other geometric shapes (TAC 1999, AASHTO 2001).

There are three different daytime scenarios where a 2D vertical alignment may obstruct sight distance whereas nighttime sight distances are typically governed by headlight coverage (Hassan 1996). These scenarios are described briefly below. Extensive research on sight distance design in three-dimensions has been documented in detail by several researchers (Hassan 1996, Hassan et al. 1996, Hassan and Easa 1998) and a review of this topic is presented in the following section.

Examples of all three scenarios can be found in Figure 2.3. Note that h_1 denotes the height of the drivers' eye while h_2 denotes the height of the obstruction or object.

a) Sight-Line Obstructed by Crest Curve

b) Sight-Hidden Dip

c) Sight-Line Obstructed by Structure or Intersecting Roadway

Figure 2.3: Three Daytime Sight-Line Obstruction Scenarios (Hassan 1996)

The first daytime scenario occurs on a crest curve where the driver sight line is obstructed by the roadway surface. The second occurs in sight-hidden dips, when a sag curve is preceded by a crest curve. Sight-hidden dips can occur regardless of whether or not a tangent exists between the two vertical curves (Hassan 1996). The third situation occurs on a sag curve where an overpass or similar type of structure is obstructing the view of the roadway ahead. Typically, sag curves present no sight distance concerns; however when other roadways intersect the main alignment, sag curves can be problematic.

The TAC (1999) guide recognizes four distinct measurements of sight distance namely: stopping sight distance, passing sight distance, decision sight distance and intersection sight distance. The TAC (1999) guide provides various tables summarizing friction coefficients for various design speeds and pavement conditions (wet or dry), perception and reaction times, object heights, braking distances for passenger vehicles and trucks and minimum passing sight distances.

Figure 2.4 shows a comparison between TAC (1999) and AASHTO (1994) on the required passing sight distance. As can be seen in Figure 2.4, the design control for the AASHTO (1994) guide is far more conservative than that used by TAC (1999).

Justification for the difference in passing sight distance requirements between design guides is based on research findings of Harwood and Glennon (1989). AASHTO has also conducted studies to confirm the validity of their passing sight distance curve

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**Figure 2.4: TAC and AASHTO Criteria for Required Passing Sight Distance
(TAC 1999)**

and have concluded that it may be conservative for modern drivers and vehicles however have decided to keep it unchanged (TAC 1999).

The AASHTO (2001) guide covers the same passing sight distance criteria as the TAC (1999) version. However, as shown by Figure 2.4, there is some discrepancy between control values that are used. For tabulated values in AASHTO (2001), metric units have taken precedent over traditional imperial units.

Occasionally due to complex clearance or other design controls, asymmetrical vertical sag curves may be required. Easa (1991) developed a sight distance modeling procedure for this type of vertical curve. Other work (Easa and Hassan 1997) has focused on headlight sight distance modeling for separate horizontal and vertical alignments. Fambro et al. (2000) criticized the reasoning behind the current AASHTO stopping sight model and presented a new model expressed as the sum of reaction distance added to braking distance. The sum of reaction and braking distance is essentially the basis for the existing model. However, it is suggested that the new model is based on reasonable parameters validated with field data and better represents “safe driver behavior” (Fambro et al. 2000).

Horizontal sight distance is another element of the horizontal alignment requiring careful consideration by roadway designers. Sight obstructions such as retaining walls, longitudinal barriers, cut slopes and buildings that are impractical to remove or relocate can inhibit lateral clearance requirements necessary to satisfy sight distance needs. Both AASHTO (2001) and TAC (1999) concur that the sight line is a chord of the horizontal curve. The required stopping sight distance (SSD) on horizontal curves is typically measured along the roadway centreline and both North American design guides facilitate calculation of SSD based on the geometric properties of the horizontal curve.

Compound horizontal curve sight distance is not discussed in detail by either AASHTO (2001) or TAC (1999). However, Easa (1993) developed a model to estimate the

minimum sight distance requirements on these types of horizontal curves. He acknowledges that these types of horizontal curves are advantageous in mountainous terrain and notes that the presented compound curve sight distance methodology should assist in maintaining safe operation on these horizontal curves.

A reverse curve sight distance model, developed by Easa (1994), can be used to determine sight distance needs for both horizontal and vertical alignments. His model suggested that the results of simpler models may "...overestimate the lateral clearance needs for reverse horizontal curves and the curve length requirements for reverse vertical curves." (NCHRP 2001). Easa and Hassan (1997) developed a headlight sight distance (HLSD) model for separate complex 2D alignments. This HLSD model can be used to evaluate nighttime sight distance needs on horizontal alignments containing any combination of spiral curves, circular curves and tangents. This model can also be used to determine nighttime sight distance on sag curves that are part of a complex vertical alignment (Easa and Hassan 1997).

Research by Gattis (1991) highlighted the importance of providing adequate head-on sight distance for two approaching vehicles. Gattis and Duncan (1995) further refined this by introducing the concept of preview sight distance (PVSD). PVSD concepts can be applied to both vertical and horizontal alignment elements and it is suggested that increased signage or pavement markings can be used to provide drivers with additional guidance in areas where the PVSD is insufficient (Gattis and Duncan 1995). Hassan

and Easa (2000) developed the 'red-zone' concept for alignment coordination that is based on PVSD. This concept is discussed further in a subsequent section.

2.3.1 3D Sight Distance

The sight distance models previously discussed, while in some cases are capable of addressing both horizontal and vertical alignments, do so independently of one another. That is, they "...are applicable to 2D separate horizontal and vertical alignments or simple elements of these separate alignments (vertical curve, horizontal curve)." (Hassan et al. 1996).

A model based on the curved parametric elements from the finite elements method was developed by Hassan et al. (1996) to represent highway features and potential sight obstructions. The flexibility of this model facilitates representation of practically all highway features including tangents, horizontal and vertical curves, side and cross slopes, superelevation and lateral obstructions. Rectangular and triangular elements are presented which allow the user to determine the particular sight distance requirements analytically instead of graphically. This difference is potentially more efficient since the requisite equations can be transcribed into computer code facilitating an automated solution (Hassan et al. 1996). The authors acknowledge that this 3D model should assist in the development of 3D geometric design standards and in quantifying the various effects of geometric elements on sight distance.

2.4 Vertical Alignment

The vertical alignment in highway design is essentially a 2D longitudinal profile or section cut vertically through the centreline of a road. A vertical alignment consists of grades or tangents and transitions between tangents known as vertical curves. The vertical curve is a parabolic function since the rate of slope change is constant (Mannering and Kilareski 1998).

Vertical curves can be classified into two main categories, crest vertical curves and sag vertical curves. For both crest and sag curves, there are three possible entry and exit tangent conditions that can exist. Entry and exit tangents can be opposite in sign, that is, one is positive and the other negative, they both can be positive or they both can be negative. Figure 2.5 shows a symmetrical sag vertical curve while Figure 2.6 shows crest and sag vertical curves for the various entry and exit grade combinations.

Figure 2.5: Symmetrical Sag Vertical Curve

Figure 2.6: Vertical Curve Combinations (Mannering 1998 from AASHTO 1994)

The vertical curves shown in Figures 2.5 and 2.6 are symmetric meaning that the distance from the beginning of vertical curve (BVC) to the point of vertical intersection (PVI) is half of the overall curve length (L). By the same token, the distance from the PVI to the end of vertical curve (EVC) is also equal to half the curve length as is illustrated in Figure 2.6. Comparing notation on Figures 2.5 and 2.6, it can be seen that BVC is analogous to PVC, or point of vertical curve, and that EVC is also known as PVT (point of vertical tangent). Depending on the literature source consulted, PVI, VPI or simply PI refers to point of vertical intersection, vertical point of intersection or point of intersection, respectively. All terms are used interchangeably and they all reference an imaginary intersection of the entry (G_1) and exit (G_2) grades. The letter A usually denotes the algebraic difference in grade between entry and exit tangents.

Asymmetrical vertical curves, although less common than symmetrical curves, are occasionally warranted in geometric design. An asymmetrical vertical curve has unequal projection of its tangents, however its component parabolic arcs are equal (Easa 1994). Easa (1994) developed a new method of asymmetrical vertical curve design that provides improved sight distance and clearance over traditional asymmetrical vertical curves. In addition, the new curve minimizes the rate of change of grade, which increases rider comfort and results in better aesthetics.

In some instances where complex vertical alignment constraints do not facilitate the use of traditional or even asymmetrical vertical curves, three-arc curves have proved to be beneficial. Easa (1998) developed the three-arc vertical curve, which consists of three separate parabolic arcs connected at the points of common curvature. He also found that the equal arc asymmetrical vertical curve was nothing more than a special case of the three-arc version. The three-arc vertical curve was compared with traditional vertical curves and it was determined that three-arc curves not only improved sight distance in some circumstances but also proved to be more flexible in satisfying challenging vertical clearance requirements (Easa 1998).

Transitioned vertical curves consist of traditional parabolic arc positioned between two vertical transitions. A transitioned vertical curve in a vertical alignment is analogous to using spiral transitions in a horizontal curve on a horizontal alignment although the mathematical formulation is different (Easa 2002). He developed formulae for the transitioned vertical curve that can be used to describe the geometry, rate of curvature

and instantaneous elevation. The transitioned vertical curve was developed based on maintaining driver comfort and is "...especially useful for sharp vertical alignments." (Easa 2002).

Most vertical curves are parabolas that are typically centered between the tangents they join about the VPI. Vertical curves can be described mathematically by the following general relationship:

$$y = ax^2 + bx + c \quad (\text{Eq. 2.1})$$

where,

- y = roadway elevation along the vertical curve
- x = distance from the BVC or PVC; meters (metric) or stations (metric/imp.)
- c = elevation at the BVC or PVC since at $x = 0$, $y = \text{PVC elevation}$

In defining the terms a and b , the first derivative of Equation 2.1 is required.

$$\begin{aligned} \frac{dy}{dx} (y = ax^2 + bx + c) \\ \frac{dy}{dx} = 2ax + b \end{aligned} \quad (\text{Eq. 2.2})$$

Applying the boundary condition at the PVC, $x = 0$, Equation 2.2 becomes,

$$\frac{dy}{dx} = b = G_1 \quad (\text{Eq. 2.3})$$

The second derivative of Equation 2.1 is the rate of change of the slope and is given by,

$$\frac{d^2y}{dx^2} = 2a \quad (\text{Eq. 2.4})$$

By examining the geometry of Figure 2.5 it can be seen that the rate of change of the slope given by Equation 2.4 above can also be written as

$$\frac{d^2y}{dx^2} = \frac{G_2 - G_1}{L} \quad (\text{Eq. 2.5})$$

Simplification of Equation 2.4 and Equation 2.5 while solving for a yields,

$$a = \frac{G_2 - G_1}{2L} \quad (\text{Eq. 2.6})$$

Substituting Equation 2.6 and 2.3 into Equation 2.1 and letting $c = y_o$ = elevation at the PVC, results in the following expression

$$y = \left(\frac{G_2 - G_1}{2L} \right) x^2 + G_1 x + y_o \quad (\text{Eq. 2.7})$$

As mentioned earlier, $G_2 - G_1$ is equal to the algebraic difference in grades, A , and the design guides define a 'K' factor, which characterizes vertical curves relative to their rate of change in slope and length as follows

$$K = \frac{L}{G_2 - G_1} = \frac{L}{A} \quad (\text{Eq. 2.8})$$

Substituting Equation 2.8 into Equation 2.7 gives

$$y = \frac{x^2}{2K} + G_1 x + y_o \quad (\text{Eq. 2.9})$$

Selecting a minimum curve length or K factor for a particular situation provides a geometric control for vertical curves. Governing situations may include sight distance, vertical acceleration comfort controls or aesthetics.

2.4.1 Current Geometric Design Philosophy

Both the AASHTO (2001) and the TAC (1999) design guides present information on vertical alignments in a similar fashion beginning with a general introduction on topography and design considerations followed by the operational effect of grades on vehicles including vertical curves.

Both guides (AASHTO 2001, TAC 1999) state "...it is generally accepted that passenger cars readily negotiate grades as steep as 4 to 5 % without appreciable loss of speed...". However, the effect of grades on trucks is recognized as having a more pronounced impact than on passenger cars. Curves based on the weight/power ratio of a typical truck are used to relate the impact of speed over distance on a particular grade. Grades play an integral role in the need and application of other design features such as passing and climbing lanes, escape lanes, turnouts, drainage, and turnouts. These design items although important, are beyond the scope of this research and will not be discussed further. Sag and crest vertical curves are then explained though there are subtle differences in the methods each guide prefers in establishing design controls.

2.4.1.1 TAC – Grades and Vertical Curves

Discussion on vertical alignments in the TAC (1999) guide sequentially covers grades, vertical curves, and synthesis of design best practices and evaluation of safety.

Maximum grades listed in Table 2.1 are tailored to the particular road classification. A three-letter designation is used to distinguish road class where the first letter is either a

Table 2.1: Maximum Gradients (TAC 1999)

Design Speed (km/h)	30/40/50		60		70		80		90		100		110		120/130	
Topography	R	M	R	M	R	M	R	M	R	M	R	M	R	M	R	M
RLU	7	11	7	11	6	9	6	8	5	7	5	7	-	-	-	-
RCU	-	-	6	10	6	9	5	8	5	7	5	7	-	-	-	-
RCD	-	-	-	-	6	9	5	8	5	7	5	7	-	-	-	-
RAU	-	-	-	-	-	-	4	7	4	6	3	6	3	6	3	5
RAD	-	-	-	-	-	-	4	7	4	6	3	6	3	5	3	5
RFD	-	-	-	-	-	-	-	-	-	-	3	5	3	5	3	5
ULU – Residential	8	15	-	-	-	-	-	-	-	-	-	-	-	-	-	-
ULU – Industrial/Commercial	6	12	-	-	-	-	-	-	-	-	-	-	-	-	-	-
UCU – Residential	8	12	7	11	7	10	-	-	-	-	-	-	-	-	-	-
UCU – Industrial/Commercial	6	12	6	11	6	9	6	8	-	-	-	-	-	-	-	-
UCD	6	10	6	9	5	8	5	7	-	-	-	-	-	-	-	-
UAU	6	10	6	9	5	8	5	7	-	-	-	-	-	-	-	-
UAD	-	-	3	6	3	6	3	6	3	6	3	5	-	-	-	-
UED	-	-	-	-	-	-	5	6	4	5	4	5	4	5	3	5
UFD	-	-	-	-	-	-	-	-	4	5	3	5	3	5	3	5

Notes:

1. Short grades less than 150 m in length, and one-way down grades may be 1% higher on urban roads, and 2% higher on low volume rural roads.
2. R refers to rolling topography.
3. M refers to mountainous topography.

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U or an *R* denoting either urban or rural. The second letter classifies *L* as local, *C* as collector, *A* as arterial, *E* as expressway and *F* as freeway. The third letter is either a *U* or *D* and denotes undivided or divided respectively. A series of design domain application heuristics are then provided so that the design engineer can better apply engineering judgment in the design process. Significant factors in the selection of maximum grade such as cost, safety, terrain, climatic condition and others are highlighted for designer consideration.

Minimum grades are then discussed for rural and urban roadways. The TAC (1999) guide indicates that level grades may be suitable for properly constructed rural roadways while suggesting that a minimum longitudinal gradient of 0.5 % be used on curbed urban roads to facilitate drainage. Several other best-practice type comments are made all focusing on the benefits of positive drainage.

In addition to the maximum gradients listed in Table 2.1, the TAC (1999) guide provides a series of performance curves for heavy trucks. These curves can be used to chart truck acceleration and deceleration on a variety of grades for three different truck mass/power ratios. The units for the mass/power ratios are grams per watt (g/W) and curves are shown for trucks boasting 120, 180 and 200 g/W ratios. It is acknowledged that the different ratios are intended to accommodate for the different acceleration and deceleration characteristics associated with shipping different commodities.

The TAC (1999) guide introduces the K value as the horizontal distance required to result in a grade change of 1 % on a vertical curve. The value of K is positive for sag curves and negative for crest curves and is as shown in Equation 2.8. Design controls are based on limiting the K value for the particular situation. Sight distance considerations were discussed in a previous section in detail. Recall that the TAC (1999) guide suggests a designer consider the following sight distances:

- stopping sight distance
- passing sight distance
- decision sight distance

- intersection sight distance.

The height of a driver's eye is taken as 1.05 m in calculating K values for the various sight distances while the object height can vary from 0.0 m to 1.3 m for pavement markings or roads prone to washouts and for passing sight distance respectively.

Minimum K values have been calculated for the different sight distance criteria for a variety of design speeds and are shown in Table 2.2.

Ranges of values are provided for stopping sight distance on both sag and crest curves while passing sight K values are provided as a finite number. Two different formulae are provided to enable manual calculation of an appropriate K value depending on whether the required sight distance is larger or smaller than the length of the proposed vertical curve. Separate pairs of equations for the above conditions are also presented based on whether or not the highway is sufficiently illuminated.

Vertical design principles are then presented, and issues such as drainage, climatic conditions and intersections are mentioned as warranting consideration when preparing a vertical alignment. Moreover, recommended vertical clearances to overhead utilities and pedestrian overpasses are also discussed. Consistent with the rest of the document, the design procedure for vertical alignments, specifically grades and vertical curves, are offered in a logical and sequential order in an attempt to prevent 'table-picking', yet still facilitate a relatively easy design process.

Table 2.2: Design Control K Values for Vertical Curves (TAC 1999)

Design Speed (Km/h)	Assumed Operating Speed (Km/hr)	Stopping Sight Distance (m)	Rate of Vertical Curvature (K)	
			Computed	Rounded
30	30	29.6	1.6	2
40	40	44.4	3.7	4
50	47-50	57.4-62.8	6.1-7.3	6-7
60	55-60	74.3-84.6	10.2-13.3	10-13
70	63-70	94.1-110.8	16.4-22.8	16-23
80	70-80	112.8-139.4	23.6-36.1	24-36
90	77-90	131.2-168.7	32.0-52.8	32-53
100	85-100	152.0-205.0	45.8-78.0	45-80
110	91-110	179.5-246.4	59.8-112.7	60-110
120	98-120	202.9-285.6	76.4-151.4	75-150
130	105-130	227.9-327.9	96.4-199.6	95-200

K Factors to Provide Stopping Sight Distance on Crest Vertical Curves

Design Speed (Km/hr)	Minimum Passing Sight Distance (m)	Rate of Vertical Curvature (K)	
		Rounded	
30	217	50	
40	285	85	
50	345	130	
60	407	175	
70	482	250	
80	541	310	
90	605	350	
100	670	480	
110	728	570	
120	792	670	
130	855	780	

K Factors to Provide Passing Sight Distance on Crest Vertical Curves

Design Speed (Km/hr)	Assumed Operating Speed (Km/hr)	Stopping Sight Distance (m)	Rate of Sag Vertical Curvature (K)			
			Headlight Control		Comfort Control	
			Calculated	Rounded	Calculated	Rounded
30	30	29.6	3.9	4	2.3	2
40	40	44.4	7.1	7	4.1	4
50	47-50	57.4-62.8	10.2-11.5	11-12	5.6-6.3	5-6
60	55-60	74.3-84.6	14.5-17.1	15-18	7.7-9.1	8-9
70	63-70	99.1-110.8	19.6-24.1	20-25	10.0-12.4	10-12
80	70-80	112.8-139.4	24.6-31.9	25-32	12.4-16.2	12-16
90	77-90	131.2-168.7	29.6-40.1	30-40	15.0-20.5	15-20
100	85-100	157.0-205.0	36.7-50.1	37-50	18.3-25.3	18-25
110	91-110	179.5-246.4	43.0-61.7	43-62	21.0-30.6	21-30
120	98-120	202.9-285.6	49.5-72.7	50-73	24.3-36.4	24-36
130	105-130	227.9-327.9	56.7-85.0	57-85	27.9-42.8	28-43

K Factors to Provide Minimum Stopping Sight Distance on Sag Vertical Curves

2.4.1.2 AASHTO – Grades and Vertical Curves

The Policy on Geometric Design of Roads and Streets by AASHTO (2001), addresses the operational importance of grades on vertical alignments in a similar fashion to the TAC (1999) guide. The most notable difference is that AASHTO (2001) presents acceleration and deceleration curves for a truck with a mass/power ratio of 120 g/W only where the TAC (1999) guide has curves for three different ratios as previously discussed. AASHTO (2001) acknowledges that this ratio is somewhat conservative. However, it also encourages designers to use a more representative ratio or alternate method in estimating critical grade length if warranted.

AASHTO (2001) suggests that a maximum grade of 5 % be used for a design speed of 110 km/hr while grades between 7 – 12 % may be appropriate for a 50 km/hr design speed. More detailed recommendations for maximum sustained gradients are presented in other AASHTO (2001) guide chapters that focus on local roads, collector roads, rural and urban arterials and freeways.

Similar to the TAC (1999) guide, the discussion in AASHTO (2001) on emergency escape ramps, passing and climbing lanes and turnouts follow guidelines on grades. Extensive details are also provided on the design of arrestor beds which are useful in suppressing out of control vehicles. Equations relating arrestor bed length to grade are provided, as are general recommendations on typical materials, anchors and access.

Crest vertical curves along with their design controls are presented first followed by sag vertical curves and design controls. The sight distance criteria as noted previously in reviewing the TAC (1999) guide accounts for sight stopping distance and passing sight distance. However, no specific procedure is outlined under vertical alignments to account for decision sight distance or intersection sight distance, rather it is dealt with in a separate guide section devoted entirely to sight distance.

Vertical alignments in 2D have little impact on vehicle stability in that a vehicle will typically not skid or roll over because the grade or vertical curve length is unfavorable. The more common adverse problem when considering vertical alignments in geometric design pertains to comfort control. Moreover, it is sag vertical curves where comfort control can be problematic because under normal operating conditions the comfort of crest curves are typically more manageable since gravitational and centripetal forces act in opposite directions (TAC 1999, AASHTO 2001).

Comfort control on sag curves is addressed through specification of a minimum curve length or K value. The comfort control K value is always smaller than the value than required for headlight control. Sag curve design controls based on comfort are not used unless the roadway facility in question is well lit. Both TAC (1999) and AASHTO (2001) guides indicate that limiting the centripetal acceleration to 0.3 m/s^2 will result in a design that ensures driver or rider comfort.

2.5 Horizontal Alignment

A horizontal alignment can be defined as "...the configuration of the roadway as seen in plan and generally consists of tangent sections, circular curves, and in some instances spiral transitions." (TAC 1999). Horizontal curves play an important role in the roadway design process since the interaction between curvature, design speed, superelevation and side friction are key to the production of a safe, efficient design that is consistent with driver expectations. Both vertical and horizontal alignment deficiencies are potentially dangerous. However, the horizontal alignment is typically regarded as more critical since horizontal shortcomings tend to introduce design inconsistencies and ultimately reduce safety (TAC 1999).

The purpose of a circular horizontal curve is to connect two tangents that extend in different directions. Spiral transitions are sometimes used to introduce the vehicle to the directional change in a more gradual manner. It should be noted that the contribution to vehicle stability provided by spiral transitions has been found to be relatively minimal (Harwood et al. 1994). NCHRP (2001) suggests that further guidance be provided by AASHTO (2001) in identifying areas where safety benefits can result through the use of spiral transition curves.

It is well known that drivers tend to steer a spiral path when entering and exiting a horizontal curve. NCHRP (2000) has determined that the length of the spiral transition curve does have a marginal effect on the operation and safety of the roadway. One of

the main benefits identified by authors in using spiral transitions was design consistency. It was noted that the use of spiral transitions was not common practice in all areas of the United States. Both AASHTO (2001) and TAC (1999) suggests the use of transition curves in horizontal alignment design, however both documents stop short of considering them a requirement. NCHRP (2000) also found that spiral transitions could improve safety on very sharp horizontal curves. The authors recommend that spirals be used where the centripetal acceleration exceeds 1.3 m/s^2 .

The relatively limited contribution to vehicle stability documented in the literature on the use of spiral transition curves precludes further investigation herein. It is acknowledged that spiral transitions do introduce a driver more gradually to the horizontal curve. It has also been documented that spiral curves assist in reducing the lateral shift experienced by a vehicle as it tracks a horizontal curve. However, the resulting effects of reduced lateral shift on stability have not been quantified (Glauz et al. 1991). All subsequent discussion on horizontal alignments assumes tangents and circular curves only as the main components involved.

Figure 2.7 shows a typical horizontal alignment. Note the *BC* denotes beginning of curve, *PI* refers to point of intersection and *EC* indicates end of curve. The *BC* and *EC* locations are where the tangent contacts the circle segment of the alignment.

Figure 2.7: 2D Horizontal Alignment

Designing a horizontal alignment requires careful consideration of many factors including safety, consistency, topography, anticipated traffic volume, climatic conditions and location of major utilities (TAC 1999). The relationships between curve radius, superelevation, side friction and vehicle speed are all extremely important in achieving a sound design and in consideration of the factors noted above. AASHTO (2001) recognizes that these relationships are tied to the laws of mechanics and dynamics but notes “the actual values for use in design depend on practical limits and factors determined more or less empirically over the range of variables involved.”

The governing horizontal relationships previously highlighted will be explored in some detail in ensuing sections but a brief explanation of vehicle steering, handling and ride characteristics may prove beneficial in appreciating the complexity of obtaining design limits and factors. A discussion of vehicle characteristics will also serve to introduce the subsequent discussion on vehicle stability and dynamics.

2.5.1 Vehicle Handling, Steering and Ride Characteristics

The following discussion of vehicle characteristics is by no means comprehensive nor is it intended to be. For a thorough review of all theory associated with ground vehicles Wong (1978) should be consulted, as should the other references noted in this section.

Vehicle handling, steering and other performance characteristics form the underlying theoretical rationale behind the limiting values and factors relating to horizontal alignment design found in current North American design guides (TAC 1999, AASHTO 2001). The theoretical considerations of vehicle dynamics and stability have been supplemented by empirical data gathered in the 1930's and 1940's, primarily relating to driver comfort which is very subjective.

Vehicle handling is essentially the response provided by the vehicle due to forces that act on it. For example, driver steering, wind, degree of horizontal curvature and vehicle suspension all affect the handling characteristic of a given vehicle. The two basic problems associated with vehicle handling are restricting the vehicle to a desired path and vehicle stabilization in that desired path (Wong 1978). The handling characteristics of a vehicle can be modeled as a rigid body with six degrees of freedom. Figure 2.8 shows a typical axis system. Motion along the y-axis is referred to as sideslip, yaw is rotation about the z-axis while roll is rotation about the x-axis, which intuitively is also known as the rotation axis (Wong 1978).

Figure 2.8: Vehicle Axis System (Wong 1978)

Wong (1978) explains that considering vehicle-cornering behavior at low speeds while neglecting centrifugal force is a more convenient introductory approach to describing the handling characteristics of a road vehicle. Centrifugal force describes the tendency of a body to continue in a straight line, tangent to the circle which it is being carried around. Newton's Laws of motion quantify the centrifugal force as mass times acceleration (Scott 1991).

An understanding of steering geometry is required for a complete appreciation of the theoretical foundation behind vehicle handling. However, details of this nature are well beyond the scope of this research and will not be considered here. Other vehicle handling characteristics include neutral steer, understeer, and oversteer coefficients, yaw velocity, lateral acceleration and curvature responses to name a few (Wong 1978).

Given the spectrum of topics that require consideration in analyzing vehicle handling it is no surprise that the geometric design guides have, at least partially, used empirical data in establishing design limits and values. Practical issues are also raised when attempting to quantify handling characteristics, such as how to average values for the national vehicle fleet. For example, large articulated trucks will handle differently than sports cars. The limiting values incorporated into design guides must not compromise safety or overall economy by being extremely conservative or too liberal.

A brief introduction to vehicle ride characteristics is warranted since both AASHTO (2001) and TAC (1999) design guides limit side friction for vehicles based in part on driver and passenger comfort tests, which were conducted over 60 years ago. “In general, passenger ride comfort (or discomfort) boundaries are difficult to determine.” (Wong 1978). This difficulty remains today despite the literature that is available on terramechanics.

Several methods have been developed for assessing human response, including subjective ride assessments, shake table tests, ride simulator experiments and ride measurements in vehicles. Human responses tend to be qualitative and as a result are influenced by the sensitivity of the individual, the diversity of the test used and the particular sensation level used by the tester (Wong 1978).

A common ride model for passenger cars is shown in Figure 2.9. As shown, the model has independent suspension for the front axle and has seven degrees of freedom (DOF).

Figure 2.9: 7 Degree-of-freedom Ride Model for Passenger Cars (Wong 1978)

The mass of the vehicle is referred to as the sprung mass while the unsprung mass is associated with the running gear and other components. Newton's second law ($F = ma$) is used to formulate equations of motion for each mass. The model shown in Figure 2.9 is relatively simple (by ride model standards) and analysis becomes increasingly difficult as more DOF's are introduced (Wong 1978).

The model shown in Figure 2.9 can be simplified to a two DOF model to represent the sprung and unsprung mass only. This simplified model can be used to examine the major motions of the vehicle (Wong 1978). Figure 2.10 shows a two DOF ride model for vehicle pitch and bounce of the sprung mass. This model is tailored to the vertical motion of the vehicle in the longitudinal direction. The z coordinates are used to describe the vertical displacements while the angular reference measures pitch.

Figure 2.10: Two degree-of-freedom ride model (Wong 1978)

Rotation of this model by 90 degrees allows the lateral movements of the vehicle to be examined. Chang (2001) has used the two DOF model to study the effect of body roll on horizontal curve design. As a vehicle travels around a curve a portion of the vehicle weight shifts to the outside tires resulting in unbalanced loads between inside and outside of curve tires. This unbalanced load due to vehicle body roll can alter the existing relationships between minimum curve radius, superelevation, side friction, and design speed (Chang 2001).

Chang (2001) found that for modern vehicles the key safety issue is rolling rather than lateral skidding. He suggested two methods in which current geometric design standards can address body roll in vehicle dynamics via modifications to the recommended values for horizontal alignment elements. He identified increasing superelevation rates on horizontal curves and increasing minimum curve radii as the most appropriate methods to account for the effect of body roll on vehicle dynamics (Chang 2001). However, since slower moving vehicles may slide inward, towards the

centre of the curve under adverse weather conditions, caution must be used when considering increased superelevation.

Finally, tire contact with the pavement and the associated friction between the two mediums also plays a role in vehicle handling, stability, and ultimately ride experience. The friction provided at the pavement-tire interface must be large enough to accommodate vehicle braking and cornering. There are at least five different variables that influence the frictional resistance available to a vehicle (Morrall et al. 1994).

They include:

1. the micro and macro texture of the pavement.
2. the surface condition of the pavement.
3. characteristics relating to the tire itself such as material, tread pattern, inflation, diameter and tractive force.
4. vehicle characteristics such as weight distribution, operating speed and driver behavior.
5. roadway vertical and horizontal alignments.

The most significant contributor to the rolling resistance, or more generally, the friction of tires on hard surfaces is the hysteresis in the tire material (Wong 1978). Hysteresis, in very broad and simplified terms, is the history dependence of a physical system or object. To visualize the meaning of hysteresis, assume a force acts on an object, such as a tire, and then is removed. If the tire does not return to its original shape then it is experiencing hysteresis.

Figure 2.11: Adhesion and Hysteresis Components of Friction

(Morrall 1994 from Kummer and Meyer 1967)

Adhesion is another component of friction that provides a notable contribution and other effects of lesser importance include "...the resistance due to air circulating inside the tire, and the fan effect of the rotating tire on the outside air..."(Wong 1978). The adhesion contribution to friction is caused by a "molecular-kinetic, thermally-activated stick-slip mechanism, which takes place essentially at the sliding interface." (Moore 1975 in Morrall 1994). Figure 2.11 shows the adhesion and hysteresis components of friction.

When side forces are applied to a tire, lateral forces are developed at the tire-pavement interface and the tire will move at an angle termed the slip angle relative to the main wheel plane (Wong 1978). The lateral forces developed at the tire-pavement interface are typically referred to as cornering forces and the relationship between these cornering forces and the slip angle are "fundamentally important to the directional control and stability of road vehicles." (Wong 1978). Figure 2.12 shows the slip angle (α), and the cornering force ($F_{y\alpha}$).

The main factor that affects the cornering behavior of tires is vertical load. Other factors include inflation pressure, tire diameter and tread pattern. When comparing the cornering characteristics of tires it is often convenient to define the cornering stiffness parameter (C_α), which is the derivative of the cornering force with respect to slip angle (Wong 1978).

Figure 2.12: Tire Behavior Subjected to a Side Force (Wong 1978)

$$C_a = \frac{\partial F_{y\alpha}}{\partial \alpha} \quad (\text{Eq. 2.10})$$

where,

$F_{y\alpha}$ = cornering force

α = slip angle

C_a = corner stiffness parameter

Significant changes in the cornering force can occur under increased normal loads on the tire. Changes are more pronounced for higher slip angles as indicated in Figure

Figure 2.13: Effect of Normal Load on the Cornering Characteristics of a Tire
(Wong 1978)

2.13. Note that the relationships are not linear therefore as load transfers from the inside to the outside tires in a turning maneuver the total cornering force that a vehicle can develop is reduced (Wong 1978). This reduction can be illustrated by considering a single axle consisting of a pair of tires where each tire has a vertical load.

Referring to Figure 2.14, F_z denotes the initial normal load on each tire while F_y represents the corresponding cornering force. As a vehicle negotiates a steady-state curve, a portion of the load shifts to the outside tire, which then experiences an increased normal load, F_{zo} and cornering force F_{yo} . Similarly, the reduced load on the inside tire and reduced cornering force can be denoted by F_{zi} and F_{yi} respectively. Consequently, the total cornering force is equal to $F_{yi} + F_{yo}$ which is less than $2F_y$ as shown in Figure 2.14.

Figure 2.14: Effect of Lateral Load Transfer the Cornering Capability of a Pair of Tires (Wong 1978)

Therefore, it can be seen that an increase in tire slip angle is required to “...develop the required amount of cornering force to balance a given centrifugal force...” (Wong 1978).

Tire behavior on a wet pavement surface is very important in considering road safety since the vast majority of commonly traveled roadways will be subject to rain and moisture for a portion their service life. Tire performance on wet surfaces is influenced by the pavement surface texture, the water depth, tread pattern and the operating mode of the tire (Wong 1978). The operating mode refers to whether or not the vehicle is cornering, braking, or accelerating.

While several studies have examined the friction requirements for braking on tangential roadway sections, few researchers have acknowledged that the horizontal alignment plays a key role in friction availability (Morrall et al. 1994). When a vehicle is

cornering, in addition to tangential friction, it must also have side friction available (Kontaratos et al. 1994). These two frictional forces, acting at 90 degrees to one another, compete for a portion of the total friction availability that the tire can supply. The proportion depends on the maneuver being performed. The coefficient of total friction available can be expressed as follows (Harwood et al. 1989, Dunlap et al. 1978, Lamm et al. 1991):

$$F_{MAX} = \sqrt{F_T^2 + F_R^2} \quad (\text{Eq. 2.11})$$

Where,

- F_{MAX} = Coefficient of total friction that is available
- F_T = Coefficient of tangential friction - for braking or acceleration
- F_R = Coefficient of side friction or radial friction

The tangential or longitudinal friction component provides vehicle stability during braking or acceleration while the side or radial friction allows the vehicle to negotiate a horizontal curve or alignment. When both tangential and radial friction forces are present, their resultant force must not exceed an upper limit if vehicle stability is to be maintained.

As shown in Figure 2.15, the resultant frictional force (F_{MAX}), must fall within the upper limit represented by the elliptical curve. If the combination of forces exceeds the maximum frictional force an accident may result (Lamm et al. 1999). For example, during a critical vehicle maneuver on a horizontal curve such as emergency braking, the

Figure 2.15: Horizontal Forces Between Road Surface and Vehicle Tire
(Lamm et al. 1991)

tangential component of friction may approach the maximum frictional force leaving little or no radial friction resulting in the vehicle sliding off the road.

There are many contributing factors that affect the friction potential at the tire-pavement interface. Figure 2.16 summarizes some of the tangible road surface and vehicle factors involved in determining the value of tangential, radial and maximum friction.

Regardless of the combination of factors at work on a given vehicle, the interaction between the tire and pavement are key in quantifying friction values. Vehicle handling and ride characteristics combined with the mechanics of pavement-tire interaction greatly influence vehicle stability.

Figure 2.16: Factors Affecting Friction Potential (Lamm et al. 1999)

Stability on horizontal alignments relies on the availability of sufficient side friction to ensure the vehicle traveling along the curve can track the desired path. Combined alignments are perhaps slightly more complex when considering frictional distribution since the superimposed grades and the vehicles' operating mode affect the component of tangential friction.

There are a great number of variables involved in determining the actual value of side friction as shown in Figure 2.16. The following section deals with vehicle stability in

more detail and will explain the role of the variables considered by current guides (AASHTO 2001, TAC 1999) in the design of horizontal alignments.

2.6 Vehicle Stability

As suggested by the numerous variables involved in vehicle behavior and cornering, the stability problem is quite complex. Highway safety is extremely dependent on whether or not a vehicle, under normal circumstances and in reasonable climatic conditions, can remain on the road surface and in its designated lane.

As a vehicle traverses a curved alignment, it experiences a centripetal acceleration that acts towards the centre of the horizontal curve (AASHTO 2001). Forces available to counteract this acceleration include a component of the vehicles weight, superelevation of the roadway, and side friction, which is developed at the tire-pavement interface (Lamm et al. 1999).

Both AASHTO (2001) and TAC (1999) design guides simplify cornering dynamics by reducing the vehicle to a point-mass traveling on a 2D horizontal alignment during analysis. In doing so, vehicle movements can be tracked by considering two directions of motion namely, the tangential and radial. Figure 2.17 illustrates the radial forces that act on a vehicle as it travels around a constant radius horizontal curve at a constant speed.

Figure 2.17: Vehicle Cornering Forces – Radial Direction (Lamm et al. 1999)

The various forces noted in the figure are broken into their components parallel and perpendicular to the pavement surface where,

α	=	angle of roadway incline (analogous to superelevation (e))
Q	=	vehicle weight force (N); $Q = \text{mass} \times \text{accel. due to gravity}$
F	=	total force opposing the centripetal acceleration
R	=	radius of the vehicles path (m)

The magnitude of both centripetal acceleration and the balancing forces are primarily dependent on the speed at which the vehicle is traveling. The radial acceleration toward the centre of the circle is equal to (Banks 1998):

$$\frac{v^2}{R} \quad (\text{Eq. 2.12})$$

Where R is as defined previously and,

v = constant vehicle speed (m/s)

The radial acceleration acting on the vehicle is opposed by a force equal to (Banks 1998, Lamm et al. 1999):

$$F = \frac{mv^2}{R} \quad (\text{Eq. 2.13})$$

Accurate estimates of the geometric variables involved are readily available to the designer. The process used to estimate the values of the resistive and tractive forces require a more detailed discussion. In addition, the integration of the above components plays a vital role in geometric design and the current North American design guides limit individual and combinations of values in the interest of safety. The margin of safety can be less than favorable under certain geometric combinations of alignment potentially eroding user safety (Dunlap et al 1978, Lamm et al. 1999, Kontaratos et al. 1994, Hassan 1996).

There are two modes of vehicle instability identified in the literature when considering vehicle stability. The first is lateral sliding or skidding and the second is vehicle rollover. Lateral sliding occurs when the radial friction falls below a level required to maintain the vehicle on course. As previously noted, the tangential frictional

component is closely related to the effect of longitudinal acceleration and deceleration while the radial component is closely associated with the cornering ability of the vehicle.

Referring to Figure 2.17, if the resisting force provided by the radial friction falls below the tractive force supplied by the centrifugal force, notwithstanding the other factors that are involved, the vehicle will skid laterally. Mathematically, radial skidding is not possible if (Lamm et al. 1999):

$$\text{tractive forces} \leq \text{resisting forces}$$

Recognizing that the radial frictional forces (F_R) that act at the tire-pavement interface are not shown in Figure 2.17, summing forces in the radial direction, parallel to the pavement surface yields:

$$\cos \alpha \left(\frac{mv^2}{R} \right) = F_R + Q \sin \alpha \quad (\text{Eq. 2.14})$$

Both AASHTO (2001) and TAC (1999) geometric design guides define a side friction factor (f), as the ratio of horizontal force to vertical force. More specifically, it is the ratio between the lateral or radial friction force and the component of vehicle weight that acts perpendicular to the pavement. The side friction factor, superelevation and other variables of vehicle stability as defined by current North American guides are discussed in more detail in subsequent sections. However, for the purposes of this section let the side friction factor (f) be defined as:

$$f = \frac{F_R}{N} \quad (\text{Eq. 2.15})$$

Where N is defined as the vertical force acting normal to the pavement surface. By summing forces in Figure 2.17, N is equal to (Banks 1998):

$$N = \sin \alpha \left(\frac{mv^2}{R} \right) + Q \cos \alpha \quad (\text{Eq. 2.16})$$

Rearranging Equation 2.15 to solve for F_{rad} and substituting Equation 2.16 and 2.15 into Equation 2.14 yields:

$$\cos \alpha \left(\frac{mv^2}{R} \right) = f \left(\sin \alpha \left(\frac{mv^2}{R} \right) + Q \cos \alpha \right) + Q \sin \alpha \quad (\text{Eq. 2.17})$$

Multiplying through the friction factor on the right hand side of the equation and dividing both sides by $Q \cos \alpha$ results in:

$$\left(\frac{v^2}{gR} \right) = f \tan \alpha \left(\frac{v^2}{gR} \right) + f + \tan \alpha \quad (\text{Eq. 2.18})$$

Recognizing that $\tan \alpha$ is simply the cross-slope of the road or superelevation (e), then Equation 2.18 reduces to:

$$\frac{v^2}{gR} = fe \left(\frac{v^2}{gR} \right) + f + e \quad (\text{Eq. 2.19})$$

Simplifying further yields:

$$\frac{v^2}{gR} (1 - fe) = f + e \quad (\text{Eq. 2.20})$$

Both AASHTO (2001) and TAC (1999) concur that the value of fe is much less than 1 and therefore can be omitted from the relationship in Equation 2.20. Therefore, Equation 2.20 can be reduced to:

$$\frac{v^2}{gR} = f + e \quad (\text{Eq. 2.21})$$

where,

v	=	vehicle speed (m/s)
f	=	side friction factor
e	=	superelevation rate on the roadway (%/100)
g	=	acceleration due to gravity (9.81 m/s ²)
R	=	horizontal curve radius (m)

A manipulation of Equation 2.21 is found in both the AASHTO (2001) and TAC (1999) geometric design guides. The formula is referred to as the point-mass formula and this basic relationship is considered by the guides to govern vehicle operation on horizontal curves.

The variables involved in the point-mass formula will be discussed in more detail in subsequent sections with particular emphasis on the limiting values used by the North American guides for the variables. Table 2.3 shows the point-mass formula as presented in TAC (1999) and AASHTO (2001).

Note that the relationships presented in Table 2.3 are based on a plain 2D horizontal alignment with no consideration for the potential effect of a superimposed vertical alignment. Moreover, the vehicle is simplified to a point-mass and considered as an unsprung body.

Table 2.3: Point-Mass Formula in Metric and Imperial Units (AASHTO 2001)

Metric	US Customary
$\frac{0.01e + f}{1 - 0.01ef} = \frac{v^2}{gR} = \frac{0.0079V^2}{R} = \frac{V^2}{127R}$	$\frac{0.01e + f}{1 - 0.01ef} = \frac{v^2}{gR} = \frac{0.067V^2}{R} = \frac{V^2}{15R}$
where: e = rate of roadway superelevation, percent; f = side friction (demand) factor; v = vehicle speed, m/s; g = gravitational constant, 9.81 m/s²; V = vehicle speed, km/h; R = radius of curve, m	where: e = rate of roadway superelevation, percent; f = side friction (demand) factor; v = vehicle speed, ft/s; g = gravitational constant, 32.2 ft/s²; V = vehicle speed, mph; R = radius of curve, ft

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Through manipulation of the point-mass formula any variable can be isolated allowing the limiting values of other variables to be quantified.

The most common manipulation of the point-mass formula is to isolate the variable R , to identify the minimum radius of horizontal curve for a given vehicle speed, superelevation and limiting value of side friction. Limiting the side friction value used in the point-mass equation is intended to prevent lateral skidding and provide comfort control. However, as noted earlier, this is based on the 2D analysis of vehicle dynamics on horizontal alignments.

The second instability mode considered by the literature is vehicle rollover. Figure 2.18 illustrates the overturning moment that is generated as a vehicle navigates a horizontal

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Figure 2.18: Overturning Moment on a Circular Curve (TAC 1999)

curve. Vehicle rollover will not occur as long as the tractive or overturning moment is smaller than the resisting moment. Referring again to Figure 2.17 and neglecting the effect of body roll and other contributing factors such as wind, this can be stated mathematically as (Lamm et al. 1999):

$$\left(\left(\frac{mv^2}{R} \right) \cos \alpha - Q \sin \alpha \right) h \leq \left(\left(\frac{mv^2}{R} \right) \sin \alpha + Q \cos \alpha \right) b \quad (\text{Eq. 2.22})$$

North American design guides do not directly account for vehicle rollover explicitly when discussing the design heuristics of horizontal curves. The simplified summation of moments in Equation 2.22 does not account for body roll. Under actual conditions a shift of vehicle weight to the tires on the outside of the curve occurs moving the vehicles' centre of gravity towards the outside of the turn. This movement of the vehicles' centre of gravity reduces the moment arm for the gravitational force that acts to resist rollover (Chang 2001).

Chang (2001) examined vehicle stability on horizontal alignments taking into account vehicle body roll and found that rollover may be more critical than lateral skidding when considering modern vehicles. He argues that the rollover threshold is actually much lower than what is currently thought correct by the AASHTO (2001) or TAC (1999) guides. He also suggests that the effect of sprung vehicles on minimum horizontal radius be incorporated into current design criteria in order to achieve greater consistency between highway and vehicle design.

Lateral skidding or sliding and vehicle rollover have been presented as two possible causes of vehicle instability on horizontal alignments. There are many vehicle design and geometric simplifications made by current design guides in addressing these potential instabilities. Perhaps the most significant is that the current design guides base their analysis on a 2D alignment. Limiting values of superelevation, centripetal acceleration, side friction and design speed are all woven together in an attempt to provide a satisfactory alignment. Each of these factors and the methods of determining their upper bound will now be discussed in more detail.

2.6.1 Side Friction Factors

The side friction factor (f), has been called the cornering ratio, lateral ratio, unbalanced centrifugal ratio, and side friction coefficient (AASHTO 2001). All terms refer to the same basic ratio, however the most popular name in the literature is side friction factor and as such, that will be the terminology favored herein.

For geometric design purposes, a vehicle is assumed to be rolling on a clean, wet pavement surface. The forces acting on the tire and pavement are examined in order to satisfy the following equation (Lamm et al. 1999).

$$F \leq F_{\max} = \mu_p Q \quad (\text{Eq. 2.23})$$

Where,

$F_{\max}$ = maximum transferable friction force (N)

μ_p = peak friction factor

Q = weight force (N)

“It is almost impossible to give general statements about the friction potential between a tire and a road surface for use as design values...” (Lamm et al. 1999). As highlighted earlier, this is due to the large number of factors and variables involved, including pavement structure and surface characteristics, climatic variability, vehicle parameters, pavement-tire interaction, vehicle speed and driver behavior.

Geometric design guides simplify the problem by considering two distinct directions of friction: longitudinal and radial. AASHTO (2001) and TAC (1999) differentiate between friction demand and supply. The side friction demand is the vehicles' need for side friction and essentially is what the side friction factor represents. The friction supply is the friction potential provided by the pavement. The interaction between friction demand and supply is key to understanding vehicle stability.

2.6.1.1 Maximum Side Friction Factors

Horizontal curves are not designed based on maximum available side friction, which is basically the point when the vehicle is about to skid (TAC 1999). Instead side friction values are limited based on driver comfort levels that were established by early researchers (Barnett 1936, Stonex and Noble 1940, Moyer and Berry 1940). Since lateral skidding occurs at a point much later than when a noticeable side pitch is first detected, it was felt that a sufficient margin of safety against skidding was being provided by limiting side friction in this manner (Barnett 1936). It was assumed that a driver would reduce his or her speed upon feeling discomfort and thus prevent any significant risk of lateral skidding. This assumed margin of safety was meant to account for adverse pavement conditions, environmental and other factors that potentially contribute to destabilizing a vehicle on a horizontal curve.

Several researchers have noted that the limiting values for side friction factors were developed over 60 years ago and that the criteria for obtaining these factors has only changed slightly over the same time period (Harwood et al. 1994, Carlson et al. 1999). The main method of obtaining design values for side friction was through the use of a ball-bank indicator. Table 2.4 lists the current permissible side friction factors for a variety of design speeds for select countries around the world. Of particular interest is the high precision factors used by Greece in contrast to the U.K. where only one value is presented for all design speeds.

Table 2.4: Maximum Permissible Side Friction Factors for Various Design Speeds
(Lamm et al. 1999)

Design Speed km/hr	Maximum Permissible Side Friction Factors									
	Australia	Austria	Belgium	Canada	France	Germany	Ireland	Italy	Japan	
50	0.35			0.16						0.10
60	0.33	0.16	0.16	0.15	0.16	0.14	0.15	0.17		0.09
70	0.31	0.15		0.15		0.12				
80	0.26	0.14	0.13	0.14	0.13	0.11	0.14	0.13		0.08
85										
90	0.18	0.13	0.11	0.13		0.10				
100	0.12	0.11		0.12	0.11	0.09	0.13	0.11		0.07
110	0.12			0.10						
120	0.11	0.10	0.10	0.09	0.10	0.07	0.12	0.10		0.06
	The		South							
	Netherlands	Portugal	Africa	Spain	Sweden	Switzerland	U.K.	U.S.	Greece	
50			0.16		0.18	0.19	0.10	0.16		0.164
60	0.17	0.16	0.15	0.16		0.16	0.10	0.15		0.150
70	0.15	0.15	0.15	0.15	0.15	0.15	0.10	0.14		0.138
80		0.14	0.14	0.14		0.14		0.14		0.127
85							0.10			0.122
90			0.13	0.14	0.12	0.13		0.13		0.117
100	0.12	0.12	0.13	0.13		0.12	0.10	0.12		0.109
110			0.12		0.10	0.11		0.11		0.102
120	0.08	0.10	0.11	0.10		0.10	0.10	0.09		0.096

2.6.1.2 Human Comfort and Ball Bank Indicators

The ball bank indicator has been widely used as “a uniform measure of lateral acceleration to set speeds on curves that avoid driver discomfort.” (AASHTO 2001). A ball bank indicator is a device containing a steel ball incased in a curved glass tube. There is a dampening liquid in the tube yet the steel ball is free to roll in response

Figure 2.19: A Typical Ball Bank Indicator

to the forces acting on it (Merritt 1998). Figure 2.19 shows a typical ball bank indicator.

Historically, the ball bank indicator has been used to set safe speeds on horizontal curves (Carlson et al. 1999). There are relationships between ball bank indicator readings, lateral acceleration and body roll rates that have evolved over the years as a result of the extensive use of ball bank indicators by highway agencies.

As documented in preceding sections, numerous forces are involved in the vehicle cornering process. The various angles involved in ball bank measurements are shown in Figure 2.20. The vehicle body roll and the centrifugal force push the steel ball outwards while the roadway superelevation angle acts to bring the steel ball back to its neutral position (Morrall et al. 1994). Ball bank indicators are typically installed with the vehicle parked on a level surface so that superelevation or body roll is not initially

Figure 2.20: Geometric Considerations of a Ball Bank Indicator (Merritt 1998)

influencing the neutral position of the steel ball. The movement of the steel ball is measured in degrees and in a neutral position, the reading should be zero degrees. The ball bank indicator shown in Figure 2.19 is in a neutral position.

In 1935, the U.S. Bureau of Public Roads issued a survey requesting that road tests be conducted to determine the safe speed on curves. Barnett (1936) analyzed the survey results and found that the side friction factors varied from 0.07 to 0.20. The average side friction factor was determined to be 0.16 for all speeds between 20 and 60 mph however there was no calibration factor used to account for differences in test vehicles or geographical locations (Merritt 1998).

Moyer and Berry (1940) found “that a ball bank indicator reading of 10 degrees was the most satisfactory indication of safe speed.” They collected data from 48 states and although the effect of body roll was neglected, they concluded that since the 10 degree reading is much lower than the maximum possible speed that the results should still be conservative. A friction factor of 0.14 to 0.15 was found to represent a ball bank indicator reading of 10 degrees.

Stonex and Noble (1940) conducted full-scale vehicle tests on the Pennsylvania Turnpike using a maximum side friction factor of 0.10. They noted that values as high as 0.39 were reached at high speeds. One of their main conclusions was that a friction factor of 0.10 should not be exceeded for design speeds above 70 mph. In addition, they recommended a factor of 0.16 be used for speeds up to 60 mph. The statistical validity of their results could be questioned since they admit only “some tests were made at 60 mph and a very few at 70 mph.” (Stonex et al. 1940).

Figure 2.21 shows the results of some of the earlier research work conducted with ball bank indicators on side friction factors. Figure 2.22 shows current lateral friction factors for various design speeds from select countries. The values plotted in Figure 2.22 are for rural high-speed facilities and the data used was taken from Table 2.4. It is interesting to note that in all cases, U.K. notwithstanding, that the trend indicates a reduced friction factor for higher design speeds. Values of lateral friction for the United

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Figure 2.21: Side Friction Factors vs. Speed (AASHTO 2001)

Kingdom are not sensitive to variation in design speed although the reasoning behind this does not seem economically justifiable for lower speed facilities.

Figure 2.23 shows how the current side friction values compare to the historical research that was conducted. Since AASHTO values are practically identical to TAC side friction limits, the figure is a good indication of how Canadian practice compare to the traditional limits as well. The heavy solid line illustrates the recommended AASHTO side friction values and consistent with previous research, the general trend is a reduction in side friction as the design speed increases.

Figure 2.22: Design Values for Lateral Friction Factors vs. Design Speed

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Figure 2.23: Side Friction Factors for Rural Highways (AASHTO 2001)

Recent research examined a number of curves with various vehicles in western Canada and found that ball bank angles of approximately 10 degrees resulted in side friction factors of 0.16 – 0.17, which correspond to AASHTO values (Morrall et al. 1994). The authors also found that curves flatter than 500 m “provide high levels of dynamic driving safety on both wet and dry pavements.”

Moreover, the value of the side friction factor is a function of length along the curve. The two points along a curve where drivers demand more side friction occur at the $2/5$ and $3/4$ points, where potentially, superelevation may not be provided at the maximum design level (Morrall et al. 1994). Figure 2.24 shows the variation in lateral acceleration as a function of distance along a curve.

Figure 2.24: Side Friction Variability Along a Curve (Morrall et al. 1994)

Carlson and Mason (1999) examined lateral acceleration and ball bank indicator readings to determine whether they were actually indicators of driver comfort. They collected data using a mid-size sedan on 5 different curves on the Pennsylvania Transportation Institute test track. Ball bank indicator readings and the unbalanced lateral acceleration were strongly correlated with a coefficient of determination (R^2) value equal to 0.924. The relationship can be expressed mathematically as follows (Carlson et al. 1999):

$$BBI = 1.115 + 52.627(ULA) \quad (\text{Eq. 2.24})$$

Where,

BBI = Ball bank indicator reading (degrees)

ULA = Unbalanced lateral acceleration or side friction factor (g's)

Harwood and Mason (1994) examined the margin of safety provided by the AASHTO (1990) design guide on minimum radius curves with respect to vehicle skidding and rollover. Providing vehicles do not exceed the design speed, they found for most situations that an adequate margin of safety does exist. However, under some conditions, skidding and rollover can occur if the vehicle exceeds the design speed even marginally (Harwood et al. 1994). The simulation performed in their analysis examined 2D horizontal alignments and no extension of the study results to combined 3D alignments was made and granted, this was not the main purpose of their study.

Moreover, concerns were raised about the time that has elapsed following the completion of research into driver comfort levels. Vehicle design has changed

significantly over the last 50 years resulting in a need to reexamine the driver comfort levels used in AASHTO design policy (Harwood et al. 1994).

With respect to friction demand on horizontal curves, Harwood and Mason (1994) recognized that the point-mass formula assumes a vehicle is tracking a curve at a constant radius. Researchers have noted that vehicles tend to oversteer at some point through a curve resulting in the vehicle traveling a radius that is smaller than the design radius (Morrall et al. 1994, Harwood et al. 1994). Consequently, the friction demanded at this point will be greater than that assumed for the initial design potentially resulting in an unsafe situation.

Vehicle rollover thresholds were also examined by Harwood and Mason (1994). They found that passenger cars have a very high threshold in the order of 1.2 g and that unstable trucks can have thresholds as low as 0.27 g. Since AASHTO (1990) allows lateral acceleration values as high as 0.17 and given the fact that vehicles typically oversteer along a portion of a curve, the margin of safety for rollover is not great (Harwood et al 1994). It was found that an unstable truck could rollover when exceeding the design speed by as little as 8 km/hr on low design speed horizontal curves and that this is a potentially realistic situation on freeway ramps (Harwood et al. 1994).

The selection criteria for the friction factors that are currently used in horizontal curve design are based on vehicle occupant comfort research that was conducted in the late 1930's and early 1940's. This criteria assumes that "drivers limit their speed on curves

to ensure comfort for the occupants of the vehicles, and discomfort is directly related to the unbalanced side friction” (Fitzpatrick 1994). There are two notable concerns that have stemmed from this assumption. The first is that side friction may not be the only cause of driver discomfort and the second is that this discomfort may be felt at speeds far slower than the limit required for vehicle safety (Fitzpatrick 1994).

Discovering all factors that contribute to driver discomfort is important so that researchers can properly evaluate the current design criteria that are based on comfort controls. In addition, if discomfort is felt at extremely slow speeds compared to the speeds that are still safe dynamically, then the highway facility will not be designed to optimal operational efficiency or economy.

2.6.1.3 Vehicle Speed and Side Friction

Vehicle speed plays an integral role in the magnitude of friction demand that is generated as a vehicle negotiates a horizontal curve. On occasion, posted speed limits may exceed the actual design speed for the facility (Fitzpatrick et al. 1997). Under adverse conditions this can result in an unsafe situation where the margin of safety is reduced relative to the instability threshold. Moreover, due to improved vehicle design, typical North American drivers tend to drive faster than the advised speed (Felipe et al. 1999). This only serves to compound the problem, by further reducing the dynamic margin of safety for the unsuspecting driver.

Recently, researchers found that drivers do not adjust their speed significantly due to wet pavement conditions (Felipe and Navin 1999). This was found to be true regardless of the gender of the driver although this contradicted the findings of others and was explained by the relatively small sample size (Bezkorovainy in Felipe and Navin 1999). In addition, their results indicated that on large radii curves drivers tend to follow the centreline while on smaller radii curves they tend to oversteer or cut the corner. This is consistent with the findings of Morrall (1994) and Harwood (1994) who also observed driver oversteer. As noted, oversteering a horizontal curve increases friction demand above what was assumed for the particular curve radius, potentially resulting in an unsafe situation especially under wet or icy pavement conditions.

The curves examined in the Felipe and Navin (1999) study were all very small in radius with the largest being only 100 m. Their findings did show that curve radius played a major role in the speed chosen by drivers. Drivers limited their speed on large radii curves based on comfortable lateral acceleration and speed environment while on small radii curves the limiting factor was strictly comfortable lateral acceleration. Speed environment refers to the speed selected by the driver as comfortable and that could be maintained through out the entire curve since on smaller radii curves the drivers tend to accelerate following the midpoint of the curve (Felipe and Navin 1999).

Bonneson (1999) developed a pair of models that together "...explain the relationship between side friction demand, curve geometry and speed from a human-behavior standpoint." One model focuses on the speed selected by drivers on curves and the

other describes the maximum side friction factors that should be used in horizontal curve design. Furthermore, there are essentially two different classes of speed models for horizontal curves. The first models speed based only on curve radius while the second considers curve radius and approach speed (Bonneson 1999).

Both models developed by Bonneson (1999) are sufficiently general to facilitate application following the appropriate calibration with observed speed distributions for existing facilities, or in the case of new projects, with estimations based on similar existing ones. The models presented are described as "...a rational basis for defining the maximum side friction factors for use in curve design." (Bonneson 1999).

The vehicle speed and side friction models described above do not explicitly address the 3D alignment issue. The models, while calibrated with actual data obtained from existing highways did not consider the effect of combined alignments. Gibreel et al. (2001) developed a model to predict operating speed on 3D alignments. Sag and crest curves combined with horizontal curves were examined and equations estimating the 85th percentile operating speed were developed for various points along the alignment (Gibreel et al. 2001).

Existing 2D operating speed models have been developed on the basis that the curve radius is "...the most significant parameter that affects the operating speed." (Gibreel et al. 2001). The model developed by the authors, due to the characteristics of combined alignments, has a larger number of required inputs including: curve radius, length of

vertical curve, superelevation rate, algebraic difference in grades, horizontal distance between horizontal and vertical points of intersection and entry and exit grades in percent.

Given the differences in input parameters between 2D and 3D operating speed models, it is difficult to compare the results graphically, however it was found that the predicted speeds strongly agreed with observed values (Gibreel et al. 2001).

Vehicle speed is important in the determination of side friction since examination of the point-mass formula (Equation 2.21) shows that vehicle speed is squared compared to the other variables involved. Moreover, an accurate prediction of operating speed allows engineers to evaluate the design consistency of an alignment and provide provision for geometric improvements to improve safety and efficiency. Design consistency will be discussed in a subsequent section but it is important to appreciate how all these different geometric elements are tied together.

2.6.2 Cross Section and Superelevation

The cross section is an important element in a geometric alignment and proper consideration of this element directly contributes to vehicle stability. Cross sectional design covers many roadway features including clearance to roadside obstructions, median width (if any), lane and shoulder width, drainage features and superelevation to name a few. All of the previously mentioned features require careful contemplation by

the designer but the focus of this discussion will be on superelevation. Superelevation, next to side friction, is perhaps the most important cross sectional characteristic when discussing vehicle stability, particularly on horizontal or combined alignments.

On tangents, the term superelevation is synonymous with cross fall or cross slope. Cross slope is essentially provided to ensure adequate drainage because even if a longitudinal slope is present on the roadway, it does not guarantee that water will migrate from the traveled portion of the lane to the shoulder and ditch or gutter. Figure 2.25 shows typical cross sections for rural divided and undivided highway alignments.

On circular curves, superelevation can be used to reduce the magnitude of the centripetal force that acts on the vehicle (Lamm et al. 1999). Recall, that as a vehicle travels along a horizontal curve it is subjected to a centripetal acceleration that is proportional to the vehicle speed. Rotation of the pavement surface towards the curve centre increases the stability of the vehicle providing the pavement surface conditions are favorable.

The selection of appropriate superelevation rates is important and all types of anticipated environmental conditions must be considered when choosing this rate. Selecting rates that are too low may not have the desired drainage effects on tangents while picking a rate that is too high or steep may hinder vehicle safety on icy curves at very low speeds. A study conducted by Lamm et al. (1999) concluded that minimum

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Figure 2.25: Rural Road Cross Sectional Elements (TAC 1999)

superelevation rates up to 2.5 % were equally safe as lower rates of 1.5 %. Moreover, a minimum superelevation rate of 2.5 % tends to be required for drainage purposes.

To complement the minimum rates, maximum superelevation must also be limited to ensure vehicle stability and safety since very high rates can cause a vehicle to slip across a highway when a vehicle is stopped or attempting to accelerate from a complete stop under icy conditions (Lamm et al. 1999). Table 2.5 shows minimum and maximum superelevation rates for various countries worldwide.

Table 2.5 Minimum and Maximum Superelevation Rates for Select Countries
(Lamm et al. 1999)

Country	Minimum Superelevation Rate, %	Maximum Superelevation Rate, %
Australia	2.0 – 3.0	Flat terrain: 6 – 7 General Maximum: 10 Mountainous terrain: 12
Austria	2.5	6 – 7
Belgium	2.5	5 – 6
Canada	1.5 – 2.0	6 – 8
Denmark	1.5 – 3.0	6
Germany	2.5	Flat terrain: 8 Flat without ice or snow: 9 Mountainous terrain: 7
Italy	2.5	7
Japan	1.5 – 2.0	10
South Africa	2.0 – 3.0	7
United States	1.5 – 2.0	With ice and snow: 8 Without ice and snow: 10 In exceptional cases: 12

An inspection of Equation 2.21 will quickly justify the variability associated with superelevation maximums depending on the nature of the terrain under consideration and anticipated prevailing environmental conditions. Higher superelevation rates offset the side friction required to maintain vehicle stability at a given speed. This has the result of decreasing the required radius of horizontal curve for a given design speed potentially reducing construction costs. Large horizontal radius curves can be unpractical and very costly to achieve in mountainous terrain due to the earthwork involved.

The preceding discussion on superelevation has focused primarily on positive superelevation where the pavement surface is rotated toward the centre of the curve. Negative superelevation is used on occasion yet it does not have the same affect on vehicle stability that positive superelevation does and will therefore only be mentioned briefly for completeness. Negative superelevation is the term used to describe the continuation of a normal crown of the pavement surface on a horizontal curve even if it is away from the centre of the curve. It is used "...as a sound solution for road surface drainage in terms of economics and environmental compatibility." (Lamm et al. 1999).

Only limited research has been conducted on the drainage aspects, dynamic driving considerations and accident potential of curves with negative superelevation, however, based on this research the following conclusions have resulted (Lamm et al. 1999):

- Negative superelevation should always be used on curves where the radii are greater than 8000 m

- For curves with $5000 \text{ m} < R < 8000 \text{ m}$ negative superelevation should be used if the arc length of the curve is less than 2 km.

Based on the recommended application of negative superelevation, it can be seen that it is only justified for very large curve radii. These curves are so flat that "...the driving dynamic investigations and accident analyses conducted so far do not allow reliable statements to be made on the friction potential and utilization..." (Lamm et al. 1999).

Providing gradual superelevation on a curve, whether positive or negative, is important so a sudden change in highway geometry does not occur. Current North American practice in the application of superelevation will be discussed in a subsequent section except it should be realized that gradual geometric cross sectional changes are used to promote safety, improve aesthetics and facilitate construction. As indicated previously, the side friction demand on a vehicle is not equal throughout the curve and this is taken into account in the application of superelevation on horizontal alignments.

2.6.3 Other Factors Contributing to Stability

The skid resistance condition of the pavement surface is stressed by both AASHTO (2001) and TAC (1999) design guides as being paramount in maintaining vehicle stability and safety. The main reason for this is that certain emergency driving maneuvers such as braking, sudden lane changes and directional adjustments within a single lane can significantly add to the frictional demands of the roadway geometry.

Often these maneuvers occur in a relatively short period of time and although high friction demands may exist, the timeframe may not be sufficiently large to facilitate a corrective response from the driver resulting in an unsafe condition (AASHTO 2001, TAC 1999).

North American guides agree, “Where practical, the maximum side friction factors used in design should be conservative for dry pavements and should provide an adequate margin of safety against skidding on pavements that are wet or covered with ice and snow.” (AASHTO 2001, TAC 1999). Other environmental conditions like high winds could potentially affect vehicle stability, however, it is expected that drivers will react appropriately and modify their driving behavior in accordance with the prevailing conditions.

2.6.4 Minimum Horizontal Curve Radius

The minimum radius of curve used in the design of a horizontal alignment is determined based on the limiting values of side friction, superelevation and design speed. Recall that based on a 2D horizontal alignment, the point-mass formula (Equation 2.21) can be derived to describe vehicle dynamics. The point-mass formula as it appears in North American design guides can be found in Table 2.3. By manipulating this equation, the variable for curve radius (R) can be isolated leading to the following mixed units formulae for minimum horizontal curve radius (AASHTO 2001, TAC 1999):

For metric units,

$$R_{\min} = \frac{V^2}{127(e_{\max} + f_{\max})} \quad (\text{Eq. 2.25})$$

where,

V	=	vehicle speed (km/hr)
$e_{\max}$	=	maximum superelevation rate (% / 100)
$f_{\max}$	=	maximum side friction factor
$R_{\min}$	=	minimum horizontal curve radius (m)

For imperial units,

$$R_{\min} = \frac{V^2}{15(e_{\max} + f_{\max})} \quad (\text{Eq. 2.26})$$

where,

V	=	vehicle speed (mi/hr)
$e_{\max}, f_{\max}$	=	as above
$R_{\min}$	=	minimum horizontal curve radius (ft.)

Although the curve radii given by Equations 2.25 and 2.26 will satisfy vehicle driving dynamics based on a 2D horizontal alignment, other factors such as sight distance and design consistency must be taken into account. For example, a minimum radius horizontal curve connecting two long tangents may appear as a kink (Lamm et al. 1999). Tables 2.6 and 2.7 show minimum radii curves for the AASHTO (2001) and TAC (1999) design guides respectively.

Table 2.6: AASHTO Minimum Horizontal Curve Radii (AASHTO 2001)

Metric						US Customary					
Design Speed (km/h)	Max. (%)	Limiting Values of f	Total (e/100-f)	Calculated Radius (m)	Rounded Radius (m)	Design Speed (mph)	Max. (%)	Limiting Values of f	Total (e/100-f)	Calculated Radius (ft)	Rounded Radius (ft)
20	4.0	0.18	0.22	14.3	15	15	4.0	0.175	0.215	70.0	70
30	4.0	0.17	0.21	33.7	35	20	4.0	0.170	0.210	127.4	125
40	4.0	0.17	0.21	60.0	60	25	4.0	0.165	0.205	203.9	205
50	4.0	0.16	0.20	98.4	100	30	4.0	0.160	0.200	301.0	300
60	4.0	0.15	0.19	149.1	150	35	4.0	0.155	0.195	420.2	420
70	4.0	0.14	0.18	214.2	215	40	4.0	0.150	0.190	563.3	565
80	4.0	0.14	0.18	279.8	280	45	4.0	0.145	0.185	732.2	730
90	4.0	0.13	0.17	375.0	375	50	4.0	0.140	0.180	929.0	930
100	4.0	0.12	0.16	491.9	490	55	4.0	0.130	0.170	1190.2	1190
						60	4.0	0.120	0.160	1505.0	1505
20	6.0	0.18	0.24	13.1	15	15	6.0	0.175	0.235	64.0	65
30	6.0	0.17	0.23	30.8	30	20	6.0	0.170	0.230	116.3	115
40	6.0	0.17	0.23	54.7	55	25	6.0	0.165	0.225	185.8	185
50	6.0	0.16	0.22	89.4	90	30	6.0	0.160	0.220	273.6	275
60	6.0	0.15	0.21	134.9	135	35	6.0	0.155	0.215	381.1	380
70	6.0	0.14	0.20	192.8	195	40	6.0	0.150	0.210	509.6	510
80	6.0	0.14	0.20	251.8	250	45	6.0	0.145	0.205	660.7	660
90	6.0	0.13	0.19	335.5	335	50	6.0	0.140	0.200	836.1	835
100	6.0	0.12	0.18	437.2	435	55	6.0	0.130	0.190	1065.0	1065
110	6.0	0.11	0.17	560.2	560	60	6.0	0.120	0.180	1337.8	1340
120	6.0	0.09	0.15	755.5	755	65	6.0	0.110	0.170	1662.4	1660
130	6.0	0.08	0.14	950.0	950	70	6.0	0.100	0.160	2048.5	2050
						75	6.0	0.090	0.150	2508.4	2510
						80	6.0	0.080	0.140	3057.8	3060
20	8.0	0.18	0.28	12.1	10	15	8.0	0.175	0.255	59.0	60
30	8.0	0.17	0.25	28.3	30	20	8.0	0.170	0.250	107.0	105
40	8.0	0.17	0.25	50.4	50	25	8.0	0.165	0.245	170.8	170
50	8.0	0.16	0.24	82.0	80	30	8.0	0.160	0.240	250.8	250
60	8.0	0.15	0.23	123.2	125	35	8.0	0.155	0.235	348.7	350
70	8.0	0.14	0.22	175.3	175	40	8.0	0.150	0.230	465.3	465
80	8.0	0.14	0.22	228.9	230	45	8.0	0.145	0.225	502.0	500
90	8.0	0.13	0.21	303.6	305	50	8.0	0.140	0.220	760.1	760
100	8.0	0.12	0.20	393.5	395	55	8.0	0.130	0.210	963.5	965
110	8.0	0.11	0.19	501.2	500	60	8.0	0.120	0.200	1204.0	1205
120	8.0	0.09	0.17	666.6	665	65	8.0	0.110	0.190	1487.4	1485
130	8.0	0.08	0.16	831.3	830	70	8.0	0.100	0.180	1820.9	1820
						75	8.0	0.090	0.170	2213.3	2215
						80	8.0	0.080	0.160	2675.6	2675
20	10.0	0.18	0.28	11.2	10	15	10.0	0.175	0.275	54.7	55
30	10.0	0.17	0.27	26.2	25	20	10.0	0.170	0.270	99.1	100
40	10.0	0.17	0.27	46.6	45	25	10.0	0.165	0.265	157.8	160
50	10.0	0.16	0.26	75.7	75	30	10.0	0.160	0.260	231.5	230
60	10.0	0.15	0.25	113.3	115	35	10.0	0.155	0.255	321.3	320
70	10.0	0.14	0.24	160.7	160	40	10.0	0.150	0.250	428.1	430
80	10.0	0.14	0.24	209.9	210	45	10.0	0.145	0.245	552.9	555
90	10.0	0.13	0.23	277.2	275	50	10.0	0.140	0.240	696.8	695
100	10.0	0.12	0.22	357.7	360	55	10.0	0.130	0.230	879.7	880
110	10.0	0.11	0.21	453.5	455	60	10.0	0.120	0.220	1094.6	1095
120	10.0	0.09	0.19	596.5	595	65	10.0	0.110	0.210	1345.8	1345
130	10.0	0.08	0.18	738.9	740	70	10.0	0.100	0.200	1838.8	1840
						75	10.0	0.090	0.190	1980.3	1980
						80	10.0	0.080	0.180	2378.3	2380
20	12.0	0.18	0.30	10.5	10	15	12.0	0.175	0.295	51.0	50
30	12.0	0.17	0.29	24.4	25	20	12.0	0.170	0.290	92.3	90
40	12.0	0.17	0.29	43.4	45	25	12.0	0.165	0.285	146.7	145
50	12.0	0.16	0.28	70.3	70	30	12.0	0.160	0.280	215.0	215
60	12.0	0.15	0.27	104.9	105	35	12.0	0.155	0.275	298.0	300
70	12.0	0.14	0.26	148.3	150	40	12.0	0.150	0.270	396.4	395
80	12.0	0.14	0.26	193.7	195	45	12.0	0.145	0.265	511.1	510
90	12.0	0.13	0.25	255.0	255	50	12.0	0.140	0.260	643.2	645
100	12.0	0.12	0.24	327.9	330	55	12.0	0.130	0.250	809.4	810
110	12.0	0.11	0.23	414.0	415	60	12.0	0.120	0.240	1003.4	1005
120	12.0	0.09	0.21	539.7	540	65	12.0	0.110	0.230	1228.7	1230
130	12.0	0.08	0.20	665.0	665	70	12.0	0.100	0.220	1489.8	1490
						75	12.0	0.090	0.210	1791.7	1790
						80	12.0	0.080	0.200	2140.5	2140

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Table 2.7: TAC Minimum Horizontal Curve Radii (TAC 1999)

Design Speed (km/h)	e_{max} (m/m)	Design Value for f	$e + f$	Minimum Radius (m) (calculated)	Minimum Radius for Design (m)
40	0.04	0.17	0.21	60	60
50	0.04	0.16	0.20	98	100
60	0.04	0.15	0.19	149	150
70	0.04	0.15	0.19	203	200
80	0.04	0.14	0.18	280	280
90	0.04	0.13	0.17	375	380
100	0.04	0.12	0.16	492	490
40	0.06	0.17	0.23	55	55
50	0.06	0.16	0.22	89	90
60	0.06	0.15	0.21	135	130
70	0.06	0.15	0.21	184	190
80	0.06	0.14	0.20	252	250
90	0.06	0.13	0.19	336	340
100	0.06	0.12	0.18	437	440
110	0.06	0.10	0.16	595	600
120	0.06	0.09	0.15	756	750
130	0.06	0.08	0.14	951	950
40	0.08	0.17	0.25	50	50
50	0.08	0.16	0.24	82	80
60	0.08	0.15	0.23	123	120
70	0.08	0.15	0.23	168	170
80	0.08	0.14	0.22	229	230
90	0.08	0.13	0.21	304	300
100	0.08	0.12	0.20	394	390
110	0.08	0.10	0.18	529	530
120	0.08	0.09	0.17	667	670
130	0.08	0.08	0.16	832	830

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Comparing Tables 2.6 and 2.7 it can be seen that the resulting radii for a given design speed and superelevation are similar as would be expected given the consistency of limiting values for side friction between design guides. The point-mass derivation of vehicle dynamics does not account for the interaction between a vehicles suspension and body. Chang (2001) suggested that revisions to current horizontal design criteria are warranted based on modeling a vehicle as a sprung body. Using a typical height, roll rate, and other characteristics of a passenger vehicle and truck, he presented the following formulae for minimum horizontal curve radius:

$$R_i = \frac{V^2}{121(0.5e + f)} \quad \text{for passenger cars} \quad (\text{Eq. 2.27})$$

$$R_i = \frac{V^2}{122.5(0.75e + f)} \quad \text{for trucks} \quad (\text{Eq. 2.28})$$

where,

V = vehicle speed (km/hr)

e = superelevation rate (% / 100)

f = side friction factor

R_i = minimum horizontal curve radius based on a sprung vehicle (m)

By inspection of Equations 2.27 and 2.28 it can be seen that the passenger cars scenario will always govern since the denominator will always be smaller for a given superelevation and side friction thereby resulting in a larger radius. These equations suggested by Chang (2001) imply that a vehicle body roll approach in selecting the minimum horizontal curve radius may be more appropriate than the current design approach of simply balancing lateral forces. His results are counter-intuitive and against the findings of Harwood and Mason (1994) who clearly demonstrated that skidding, and not rollover, was the critical instability mode for passenger vehicles.

To illustrate the differences between the AASHTO (2001) approach (Equation 2.25) and the approach presented by Chang (2001) (Equation 2.27), curve radii values were calculated and plotted in Figure 2.26. A superelevation rate of 6 % was used and values of side friction were taken for design speeds of 20 – 130 km/hr from Table 2.4.

Curve Radius vs. Design Speed

Figure 2.26: AASHTO and Chang Minimum Curve Radius vs. Design Speed

As shown by Figure 2.26, the difference between the AASHTO (2001) and Chang (2001) formulation is fairly significant at the higher design speeds. The results of the Chang (2001) formulation imply that the current design guides are not providing an adequate margin of dynamic safety.

In order to achieve the same geometric radius using Equation 2.27, the superelevation rate, side friction factor or combination of the two variables must be increased.

Alternately, the design speed could be reduced but no further comparisons between the two different methodologies could then be made since the facilities would not be fundamentally the same.

Increasing superelevation may have adverse effects at lower speeds under poor climatic conditions while allowing a high value of side friction to be developed would theoretically increase driver discomfort and reduce the dynamic factor of safety that has traditionally been provided. In light of the difference between the minimum horizontal curve radii shown in Figure 2.26, Chang (2001) suggests increasing superelevation rates or implementing speed reduction methods as potential solutions to maintaining the desired margin of safety.

2.7 Other Important Related Aspects

All of the major factors involved in vehicle stability have been discussed in some detail and the general methodology for the establishment of a minimum horizontal curve radius has been presented. There are other pertinent details presented in the AASHTO (2001) and TAC (1999) guides with respect to the integration of these factors into a consistent, safe, and economical alignment that necessitate further discussion.

Other horizontal geometric design elements such as spiral transitions or clothoids, compound curves, reverse curves while all important and commonly used in practice, are beyond the scope of this research. This section will examine the current guideline consideration given to superelevation, side friction factors, design consistency, and alignment aesthetics. In addition, a short discussion on the guideline recommendations regarding the coordination of horizontal and vertical alignments will be provided.

2.7.1 Distribution of e and f

The maximum side friction factors and superelevation rates were discussed in Section 2.5. This section complements that discussion by examining the current methodology that is employed in the application of side friction and superelevation. Roadway lengths, referred to as development lengths, are required to “change the cross slope from one of a normal roadway (i.e.: for a tangent section) to that of a fully superelevated cross section on a circular curve.” (TAC 1999).

The length required to facilitate this transition is referred to as the overall length of superelevation development, and it consists of two components (TAC 1999).

1. Superelevation runoff: length required to change the cross section from an adverse crown to a fully superelevated section.
2. Tangent runout: length required to change the cross section from a normal crown to where the adverse crown is removed.

These components are illustrated in Figure 2.27. In addition, there are three common methods used in current practice for implementing superelevation. First, the pavement surface can be rotated around the centreline. This is shown as Method 1 in Figure 2.27. Methods 2 and 3 rotate the pavement about the inside and outside edges respectively. All three methods can be used on rural two lane roadways although Method 1 is the most common since it causes less pavement distortion than the other two resulting in improved aesthetics (TAC 1999).

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Figure 2.27: Three Methods of Superelevation Development (TAC 1999)

Although spiral transitions to horizontal curves are beyond the scope of this research, Figure 2.27 also illustrates superelevation development methods for spiral transitions and for simple curves. Considering simple curves as the horizontal component of combined alignments in this research, Figure 2.27 shows that 60 % of the superelevation runoff occurs on the tangent before the curve begins and the remaining 40 % occurs along the curve itself.

Applying only a portion of the design superelevation prior to the horizontal curve means that the design rate is not yet established when a vehicle enters the curve. This could potentially result in an unsafe situation especially if side friction is approaching the upper limit and the vehicle is steering the minimum radius path. Other research has found that vehicles tend to steer a spiral path at the beginning and end of a horizontal curve and therefore the superelevation distribution method discussed above does not adversely affect safety (Morrall et al. 1994).

AASHTO (2001) describes five different distribution methods for “sustaining centripetal acceleration on curves by use of e or f or both.” The methods are discussed individually below but can briefly be listed as follows:

1. Method 1: Superelevation and side friction are directly proportional to the inverse of the curve radius.
2. Method 2: Side friction is used alone until f_{max} is reached at which point superelevation is introduced up to e_{max} in order to sustain the lateral acceleration. In

this method side friction followed by superelevation are increased in inverse proportion to the radius of curvature.

3. Method 3: Opposite of Method 2 in that first superelevation is applied to the maximum rate and then side friction is used to counteract the balance of the lateral acceleration. Superelevation followed by side friction is increased in inverse proportion to the radius of curvature.
4. Method 4: Identical to Method 3 except the running speed is used instead of design speed for determining superelevation and side friction.
5. Method 5: Superelevation and side friction are applied in a curvilinear relationship with the inverse of curve radius. Values tend to fall between Methods 1 and 3.

By providing numerous acceptable methods, the AASHTO (2001) guide in particular, provides ample opportunity for the design engineer to use engineering judgment. The TAC (1999) guide only presents the AASHTO (2001) Method 5 for use on rural two lane roadways although Method 2 is suggested as being appropriate for use in low speed urban situations.

Vehicles traveling on any given highway are negotiating tangents and curves. These curves can potentially vary in radius from very large, down to the minimum radius for the given design speed. Method 1 uses a linear relationship to vary the side friction and superelevation based on the inverse of the curve radius. On a tangent, the values of side friction and superelevation are taken as zero, while for a minimum horizontal radius curve both side friction and superelevation would be at the maximum for the given

design speed. Any intermediate curve is given side friction and superelevation proportional to the inverse of its radius. This method, although very simple, has its disadvantages since traffic speeds vary (AASHTO 2001).

Method 2 uses side friction, up to the maximum side friction factor, to counteract all the lateral acceleration that occurs on a curve. Once the maximum side friction factor has been reached superelevation is introduced rapidly to assist with vehicle stabilization. On very flat curves, superelevation may not be used at all if the maximum side friction factor has not been reached. This method is advantageous in urban low speed locations where due to physical constraints, superelevation cannot always be used (AASHTO 2001, TAC 1999).

Method 3 operates in reverse of Method 2 in that superelevation is used to counteract all the lateral acceleration on a curve up to the maximum design superelevation rate. On sharper curves, the maximum superelevation rate is sustained and then side friction is used to balance any surplus lateral acceleration up to the maximum side friction factor for the minimum horizontal radius curve. On very flat curves, positive side friction may not be used at all. However, negative side friction may develop, especially at lower speeds requiring the driver to steer against the curve. This can potentially result in erratic driving behavior. This method was used in the early days of highway construction and although it does not appear to be favored by current design guides, it is still a documented design alternative in the literature (AASHTO 2001).

Method 4 was developed to eliminate the shortcomings of Method 3 through the use of speeds lower than the design speed. Superelevation is introduced to sustain lateral acceleration up to the maximum rate and then lateral friction is used to counteract the balance. In theory, this method reduces the negative side friction that develops because the speeds used in calculations are the average running speed and not the design speed. Average running speeds are typically taken as low as 80 % of the design speed. The disadvantages of method 4 are identical to those of Method 3 however tend to be less pronounced due to the reduced speeds that are considered (AASHTO 2001).

Method 5 draws on the advantages of Method 1 and Method 4 to provide a “practical distribution of the range of radii.” (AASHTO 2001, TAC 1999). Method 4 accommodates overdriving on flatter intermediate curves while method 1 avoids the maximum superelevation rate for a large part of the range of curve radii. Both these features are desirable for safety reasons and as such are both incorporated into Method 5. Figure 2.28 shows a graphical representation of the five different methods.

Figure 2.28: Methods 1-5 for Distributing f and e (AASHTO 2001)

The inherent advantages of Method 5 make it the preferred or recommended method by both AASHTO (2001) and TAC (1999) design guides. The superelevation and side friction distribution of Method 5 is represented by an unsymmetrical parabolic curve that can be created for a range of curve radii for a given horizontal alignment.

AASHTO (2001) presents an extensive explanation on the application of Method 5 that will not be repeated here. Figure 2.29 shows the results of the required mathematical computations. The range of radii under consideration in the figure, vary from 80 m to approximately 450 m for a design speed of 50 km/hr with a maximum superelevation of 8 %. These values are arbitrary and have been selected only to facilitate production of the set of curves in the figure.

Briefly, the $e + f$ curve is created by adding together the maximum superelevation and side friction factor and applying that value to the minimum horizontal curve radius for the given design speed. A series of construction curves and mathematical manipulations of the point-mass formula are then used to produce the parabolic f curve. Side friction values (f) are then subtracted from the corresponding $e + f$ value resulting in the e distribution curve.

It should be noted that a distinction is made between design speed and running speed for the Method 5 distribution. Maximum superelevation and side friction values used for the $e + f$ curve are based on the design speed and the maximum side friction value used for the f curve is based on the running speed.

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Figure 2.29: Method 5 - Superelevation and Side Friction Distribution (TAC 1999)

The AASHTO (2001) guide should be consulted for the exact procedure and methodology used for the construction of Method 5 curves, or alternately, both TAC (1999) and AASHTO (2001) have a series of tables that can also be used. These tables contain values for the superelevation design elements for curves ranging from approximately 7000 m (23,000 ft.) down to the minimum horizontal radius based on design speed.

Current methodologies for the establishment of tangent runoff and superelevation runoff vary slightly between the AASHTO (2001) and TAC (1999) guides. Recall that tangent runoff is the transition distance required to eliminate the pavement cross slope while superelevation runoff is the transition distance needed to achieve design superelevation from a neutral (zero) cross slope.

Current North American guides determine minimum superelevation runoff lengths based on both comfort and appearance. The TAC (1999) guide recommends the use of spiral segments for tangent to curve transitions and subsequently correlate the superelevation runoff length to the spiral length. For simple curves where a spiral segment is not desired, the same superelevation runoff length is selected as if a spiral were being used, however the distance is proportionally distributed onto the tangent and circular curve. As can be seen in Figure 2.27, 60 % of the runoff is accommodated on the tangent while 40 % of the required transition distance occurs on the curve itself.

AASHTO (2001) recommends that the superelevation runoff distance be established based on comfort control and aesthetics by limiting the maximum relative gradient between the longitudinal grade of the road and the pavement edge. A maximum value of 0.5 % (1:200 slope) has been documented as being appropriate for speeds equal or greater than 80 km/hr (MacAdam et al. 1985). This value has been adopted by AASHTO (2001) although they have extrapolated greater values for slower design speeds and lesser values for higher speeds. Table 2.8 shows the maximum relative gradients adopted by AASHTO (2001) for superelevation runoff.

Using these gradients, the required runoff distance is then computed. Distribution of the runoff length varies, however it is suggested that most U.S. agencies proportion 67

Table 2.8: Superelevation Runoff Maximum Relative Gradients (AASHTO 2001)

Metric			US Customary		
Design Speed (km/hr)	Maximum relative gradient (%)	Equivalent maximum relative slope	Design Speed (mi/hr)	Maximum relative gradient (%)	Equivalent maximum relative slope
20	0.80	1:125	15	0.78	1:128
30	0.75	1:133	20	0.74	1:135
40	0.70	1:143	25	0.70	1:143
50	0.65	1:150	30	0.66	1:152
60	0.60	1:167	35	0.62	1:161
70	0.55	1:182	40	0.58	1:172
80	0.50	1:200	45	0.54	1:185
90	0.47	1:213	50	0.50	1:200
100	0.44	1:227	55	0.47	1:213
110	0.41	1:244	60	0.45	1:222
120	0.38	1:263	65	0.43	1:233
130	0.35	1:286	70	0.40	1:250
			75	0.38	1:263
			80	0.35	1:286

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% of the computed distance to the tangent while the remaining 33 % is reserved for the curve (AASHTO 2001). It is recognized that proportioning the entire runout length before the curve on the tangent or completely within the curve each raise different operational issues (AASHTO 2001). Providing superelevation where it is not needed for vehicle stability can be just as detrimental as not providing it where it is needed.

In determining a suitable length for tangent runout, the amount of cross slope to be removed and the rate of removal must be considered (AASHTO 2001). The objective is to provide a smooth transition that is free of visible distortion to the driver and to gradually introduce the changing cross section. AASHTO (2001) presents the following equation for determining the minimum runout length:

$$L_t = \frac{e_{NC}}{e_d} L_r \quad (\text{Eq. 2.29})$$

where,

L_t	=	minimum tangent runout length (m)
e_{NC}	=	normal cross slope rate (%)
e_d	=	design superelevation rate (%)
L_r	=	minimum length of superelevation runoff (m)

Assuming a typical cross slope rate of 2 %, the tangent runout length can vary between 16 and 50 % of the superelevation runoff length. This applies for design superelevation rates of 12 to 4 % respectively.

The TAC (1999) guide takes a slightly simpler approach to the tangent runoff problem. Instead of varying the length as a function of the superelevation runoff, it is suggested that a relative slope of 1:400 be used in all high speed rural situations. It is noted that for low speed urban situations, the slope can be increased in order to match the slope of the superelevation runoff (TAC 1999).

2.7.2 Design Consistency and Aesthetics

Design consistency and aesthetics are relatively new areas that are currently gaining notoriety in North American geometric design guides. Design consistency is essentially the process of selecting consecutive design elements so that a drivers' expectation is not violated. Al-Masaeid et al. (1995) define roadway consistency as a design system structured to eliminate erratic driving maneuvers that can be potentially dangerous. Aesthetics deal with the looks or beauty of the highway facility. Some researchers note that safest highways tend to be the most aesthetically pleasing as well (Lamm et al. 1999).

The TAC (1999) design guide suggest three principles that can be used by road designers in evaluating the consistency of a road design as follows:

- Cross section consistency
- Operating speed consistency
- Driver workload consistency

These criteria are all inter-related and contribute to the overall consistency of a highway alignment. Collision history for the roadway in question is even more important in evaluating consistency than the criteria previously noted. Collision history is documented as the “most valuable tool for evaluating design consistency.” (TAC 1999).

Cross section consistency refers to the selection of cross sectional elements that complement the horizontal and vertical alignments. This includes items such as lane and shoulder widths and lateral clearances from the road edge. Once selected, these features should remain constant throughout the alignment and if changes are warranted, they should occur gradually with plenty of advance warning given to the driver (TAC 1999).

North American design standards are based on the design speed concept. It should be noted that the application of this concept does not eliminate design inconsistencies (TAC 1999, Hassan et al. 2000). The promotion of a uniform operating speed throughout a geometric alignment enhances the safety of the roadway (TAC 1999). The 85th percentile speed is often measured or estimated to evaluate the consistency of the alignment. The 85th percentile speed is the speed not exceeded by 85 % of the drivers on a given roadway.

The importance of accurately predicting the operating speed is well documented in the literature (TAC 1999, Hassan et al. 2000, Harwood et al. 2000). Drivers traveling on

long tangents that precede horizontal curves may be traveling well above the design speed for the alignment resulting in unsafe situations. Many mathematical relationships have been developed to estimate the 85th percentile speed based on geometric elements such as curve radius, lane width and length of tangent between curves to name a few. Most of these relationships are based on a 2D consideration of the horizontal alignment. TAC (1999) presents the following formulation for predicting the 85th percentile speed based on the characteristics of a horizontal curve:

$$V_{85} = 102.45 + 0.012L - \frac{(2741.75 + 5.73L)}{R} \quad (\text{Eq. 2.30})$$

where,

V_{85} = 85th percentile speed on curves (km/hr)

L = length of curve (m)

R = radius of horizontal curve (m)

The 85th percentile speed predicted by Equation 2.30 has been found to consistently exceed the design speed on curves by 10 km/hr for design speeds up to 100 km/hr. For design speeds over 100 km/hr the 85th percentile speed is typically lower than the design speed (TAC 1999).

Driver workload is another factor requiring consideration when evaluating the consistency of a particular highway segment. Workload can be defined as the “demands imposed on the driver by the road.” (TAC 1999). It is recognized that

although various methods have been developed to assess driver workload that none have been validated to the point where they can be used in a quantitative manner (TAC 1999, Wooldridge et al. 2000).

The roadway must provide enough stimulation and interaction so that the driver remains alert and attentive while at the same time not requiring so much effort from drivers that they are unable to concentrate on the task at hand. Figure 2.30 shows the perceived relationship between workload and accident rates (Wooldridge 1994, Lamm et al. 1999).

Although no chapter in the AASHTO (2001) guide explicitly deals with the broad topic of 'consistency', all of the same criteria found in the TAC (1999) guide are discussed.

Figure 2.30: Driver Workload and Accident Rate Relationship (Lamm et al. 1999)

The effects of operating speed, driver workload and cross sectional elements on design consistency are duly noted in AASHTO (2001) and recommended design heuristics are presented.

Of the three consistency measures noted above, operating speed consistency and its effect on safety is perhaps documented most extensively in the literature. Lamm et al. (1999) suggest that differences between design speed and operating speed, usually the 85th percentile speed, can be used to effectively evaluate the safety of an alignment. Equation 2.30 is presented in the TAC (1999) guide as a reasonable estimate of the 85th percentile speed on 2D horizontal curves.

Morrall and Talarico (1994) developed a model that related the 85th percentile speed to the degree of curve. Their model was developed for simple horizontal curves with paved surfaces and constant shoulder widths that had overlapping vertical grades less than 5%. Gibreel et al. (2001) developed a method to estimate operating speed on 3D alignments. The authors acknowledge that the input requirements are fairly extensive, however, they also note that the 3D model provides a more representative estimate of actual operating speeds on combined alignments.

Speed profiles for a given alignment can be developed based on the anticipated operating speed on tangent and curved roadway elements. Speed differences between successive alignment elements less than or equal to 10 km/hr indicate a good design, while a fair and poor design are denoted by speed differences between 10-20 km/hr and

more than 20 km/hr respectively (Lamm et al. 1999). By identifying alignment elements that produce large speed differentials, engineers can modify their designs so a more consistent operating speed, and safer alignment, is achieved.

Two consistency measures, which are documented in the literature but are not discussed in great detail in the North American design guides, are operating speed consistency, alignment indices, and vehicle stability. Alignment indices are used to measure the general characteristics of a roadway section (Hassan et al. 2001). Their use in evaluating consistency is valuable since alignment inconsistencies occur concurrently with significant alignment changes. Vehicle stability as a consistency measure is based on the difference between side friction supply and demand although more research work is required to accurately model these parameters (Hassan et al. 2001).

Good roadway aesthetics are regarded as an important characteristic of the geometric design process resulting in more relaxed drivers and subsequently a safer facility (TAC 1999). Good aesthetics are the result of proper consideration and coordination of vertical and horizontal alignments. Simply designing a roadway to meet the current North American geometric design guides will not ensure a satisfactory alignment. Consistent alignments should always be the goal of geometric design for safety, efficiency, and aesthetic reasons (AASHTO 2001).

Finally, it should be noted that the brief discussion on design consistency and aesthetics above is rudimentary in scope when considering all of the available literature and

current research on the topic. The material has only been presented to highlight sources for the potential vehicle stability problems that can arise in designing a geometric alignment. In particular, operating speed as a consistency measure indicates that variation from the design speed on a facility is not an uncommon occurrence. In light of this observation, the objectives of this research as documented earlier are warranted and relevant to current geometric alignment design practices.

2.7.3 Coordination of Horizontal and Vertical Alignments

Assuming that the location of the proposed highway is generally fixed, the designer is then faced with the challenge of selecting horizontal and vertical alignment elements that complement one another (AASHTO 2001, TAC 1999). The design speed selected for the facility under consideration becomes increasingly important as the process moves from the preliminary to the detailed design stage. Sight distances, maximum gradients, minimum horizontal curve radii and other critical design elements are all determined based on the design speed (TAC 1999).

Many factors must be considered when coordinating the design elements for a geometric alignment. With respect to vehicle stability, the overlapping of horizontal and vertical curvature is a key consideration. While it is recognized that the superposition of horizontal and vertical design elements results in an aesthetically improved facility, both curvature and longitudinal grades should be compatible with one another (AASHTO 2001). Interestingly, in contradiction to the previous statement,

current design guides also suggest that minimum horizontal curves not overlap crest or sag vertical curves for safety and stability reasons (AASHTO 2001, TAC 1999).

Proper 3D visualization of the proposed alignment is recommended to identify any potential design flaws or issues early in the process since these features are permanent and costly to adjust after the fact (AASHTO 2001, TAC 1999). Computer programs and drafting packages have recently evolved to the point where they are valuable in assisting designers with this task.

Other factors such as lighting, erosion control, and utility placement, while very important to the overall quality of the proposed facility, are generally less significant when considering vehicle stability. Proper drainage can be a main stability concern since excess water on the pavement surface reduces friction supply as discussed earlier and can potentially cause hydroplaning.

Figure 2.31 shows examples of good and poor coordination of alignments in a 3D perspective. As illustrated in Figure 2.31, significant differences in appearance can result through the selection of horizontal and vertical alignment elements. The results are permanent, so care should be taken to ensure that the selected elements are compatible prior to the detailed design process and construction. Poorly coordinated alignments can reflect poorly on the designer and are perceived negatively by roadway users.

Figure 2.31: Good and Poor Alignment Coordination and Aesthetics (TAC 1999)

In order to minimize the adverse effects of poor alignment coordination and aesthetics, combinations that are potentially troublesome must be identified. When examining the available literature on alignment coordination, one particular combination is well documented as being problematic with respect to vehicle stability. The troublesome combination occurs when a sag curve coincides with a sharp horizontal curve. This combination can cause an illusion where the horizontal curve appears to be flatter than it actually is (Smith et al. 1994).

It has been suggested that in order to achieve a satisfactory 3D design solution for sag curves that the ratio between horizontal curve radii and vertical curve radii be as small as possible (Smith et al. 1994). Smith et al. (1994) recommend this ratio to lie between 1/5 and 1/10. The authors indicate it is not always feasible or practical to achieve these ratios in design. In addition, if the vertices of the horizontal and vertical curves coincide then the alignment will appear less distorted from a distance (Lamm et al. 1999).

When referring to alignment coordination there is no quantifiable indicator or measure found in the current North American geometric design guides but general statements and best practices are presented. Speed is emphasized as an important consideration because any discrepancies between operating and design speed can lead to reduced safety either through increased collisions or loss of vehicle stability (AASHTO 2001, TAC 1999). The complexities associated with the 3D geometric highway alignment “still represent the weakest link in the overall design of highways.” (Smith et al. 1994).

Recently, 'red-zone' analysis has shown promise in identifying locations on vertical curves in which horizontal curvature should not be introduced or combined into the alignment (Hassan and Easa 1998). The 'red-zone' concept, which is based on sight distance needs, is typically related to identifying deficiencies associated with preview sight distance and stopping sight distance. Hassan and Easa (1998) indicate that 'red-zone' analysis is a tool that can be used by design engineers to eliminate inadequate alignment coordination in new designs and identify potential coordination problems with existing facilities.

2.8 Point-Mass and Dynamic Models

Vehicle dynamics were briefly introduced in previous sections to justify the discussion on side friction, superelevation, minimum horizontal curve radius, and vehicle stability. The derivation of one model, the point-mass model, was presented to highlight the interdependence between the various elements just mentioned. This section will briefly re-examine the point-mass model and present three other models that can be used to assess or quantify vehicle dynamics.

All dynamic models have their inherent advantages and disadvantages. Whether exchanging accuracy for simplicity or computer simulation for full-scale tests, every effort should be made in understanding the limitations and data requirements of the particular model so that the results are as meaningful as possible.

2.8.1 Point-Mass Model

The point-mass formula was shown in Table 2.3 and the radial forces that act on a vehicle on a 2D alignment were illustrated in Figure 2.17. The point-mass model represents steady state turning on a 2D superelevated horizontal alignment with the vehicle traveling at a constant speed. Representing the vehicle as an object with its mass concentrated at a single point, the point-mass model can be illustrated as shown in Figure 2.32.

Figure 2.32: Representation of the Point-Mass Model (MacAdam et al. 1985)

As can be seen in the plan view, there are three normalized forces acting on the mass in the radial direction. These are the centrifugal force, the weight component of the vehicle and the lateral friction force. Since the vehicle mass (m), speed (v), superelevation (e) and curve radius (R) are all known quantities the standard point-mass formulation can be derived as previously shown in Table 2.3 by substituting in the acceleration due to gravity.

The point-mass model, although relatively simple, is not without its shortcomings. The effect of an overlapping vertical alignment is totally ignored. Since the model approximates a vehicle as a point-mass there is no consideration given to the distribution of frictional forces between the inner and outer or front and rear tires. Moreover, the point-mass formulation assumes the vehicle is following a path identical to the actual curve radius, which does not actually occur (Harwood et al. 1994). Finally, only the driving mode of the vehicle is considered and no provision is made for the reduction of available lateral friction due to other driving modes such as braking or an erratic emergency maneuver (Kontaratos et al. 1994).

2.8.2 Bicycle Model

An extension of the point-mass model can be made by inclusion of a front and rear tire located at a distance a and b respectively from the mass centre (MacAdam et al. 1985). The model resulting from this point-mass model extension, shown in Figure 2.33, is known as the bicycle model. As illustrated in the figure, there are now two frictional

factors acting in the radial direction that must be considered in the force equilibrium process. Recall that the friction factor is the ratio of lateral friction force to vertical friction force. More specifically, it is the ratio between the lateral or radial friction force and the component of vehicle weight that acts perpendicular to the pavement. The introduction of a front and rear tire now require that two individual friction factors be determined (MacAdam et al. 1985).

Since the forces acting on the mass in Figure 2.33 do not coincide at one point, it is necessary to establish both moment and force equilibrium. The additional equation resulting from the summation of moments facilitates establishing a solution for the

Figure 2.33: Representation of the Bicycle Model (MacAdam et al. 1985)

additional friction force that has been introduced. Notwithstanding the mathematical details it can be shown that the front and rear frictional factors are equal as follows (MacAdam et al. 1985):

$$f_1 = f_2 = \frac{V^2}{gR} - e \quad (\text{Eq. 2.31})$$

All the variables remain as they have been defined in previous equations. Therefore, simply distributing the frictional forces to a front and rear tire result in friction factors that are identical to the values resulting from the use of the point-mass formula (MacAdam et al. 1985).

2.8.3 Two-Axle Model

The two-axle model shown in Figure 2.34 “represents the next step toward realism by adding width to the bicycle model in the form of left and right side tires and a mass centre elevated some distance above the ground.” (MacAdam et al. 1985). The addition of width results in two additional friction factors, f_3 and f_4 that correspond to the tires on the outside of the turn as shown in Figure 2.34.

The elevated mass centre assumed for this model allows the weight of the vehicle to shift laterally. This requires roll equilibrium for the vehicle to maintain stability in addition to force and moment equilibrium. Roll equilibrium provides the additional

equation needed to solve for the outer tire frictional factors introduced by accounting for vehicle width (MacAdam et al. 1985).

Comparing Figures 2.33 and 2.34 it can be seen that the bicycle and two-axle models are quite similar with the exception of the different vehicle loads that occur along each axle (MacAdam et al. 1985). As a two-axle vehicle negotiates a curve at a reasonable speed, the vehicle weight shifts towards the outside tires due to body roll (Wong 1978).

Due to the effect of body roll and ultimately the transfer of vehicle weight to the outside tires during cornering, the friction factors that result from the two-axle model differ

Figure 2.34: Two-axle Model (MacAdam et al. 1985)

from the values obtained from the point-mass formula (Wong 1978, MacAdam et al. 1985). 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 holds true for the tires on the outside of the curve, the resulting lateral friction factor will be smaller than that given by point-mass estimation.

2.8.4 Interactive Highway Safety Design Model – VDM

The United States Federal Highway Administration (FHWA) is currently developing the Interactive Highway Safety Design Model (IHSDM). The IHSDM consists of a group of modules that allows designers to evaluate the safety of a geometric design (Lamm et al. 1999). One module within the IHSDM is the Vehicle Dynamics Module (VDM).

The VDM is capable of accurately simulating a vehicle traveling through a user-defined alignment. Estimates of the lateral acceleration can be obtained allowing the designer to evaluate geometric alternatives against one another. Figure 2.35 shows the intended role for VDM within the IHSDM.

The FHWA engaged the University of Michigan Transportation Research Institute (UMTRI) and Systems Technology Inc., (STI) to produce alternative prototypes of the VDM for integration with the IHSDM. Since this research is based on the use of the

Figure 2.35: VDM Role in IHSDM (Lamm et al. 1999)

UMTRI product called VDM RoAD (Vehicle Dynamic Models Roadway Analysis and Design), only the UMTRI modeling procedure will be discussed herein.

When compared to the bicycle or even the two-axle model, the VDM is significantly more sophisticated. It takes into account vehicle characteristics such as body roll, pitch, yaw, and lateral weight distribution (MacAdam et al. 1985). Consideration must be given in reaching a balance between model simplicity and utility. The more complex a model is made, the more difficult it becomes to use. Furthermore, increased complexity does not necessarily make a model more accurate (Sayers 1999).

Figure 2.36 illustrates the VDM for a single unit vehicle. In addition to the movements shown, the model is capable of accounting for vehicle bounce and pitch in the longitudinal direction similar to that shown in Figure 2.10. The flexibility of this

model allows for one body element or unsprung mass and up to six axles or sprung masses. Input parameters for the model include vehicle characteristics such as weight, and geometric information as well as highway geometry and speed. The model can be used to calculate the tire forces and required steer angles for a given highway geometry and operating speed (MacAdam et al. 1985).

The VDM model has been created with the understanding that it would be eventually integrated into a larger computer program so efforts were taken to ensure and maintain its flexibility. Sparing the reader of the parametric inputs required, the single unit VDM model can be extended to consider a tractor semitrailer configuration by including another sprung mass element (semitrailer) and the required articulation input between the masses (MacAdam et al. 1985). More details on the specifics of the VDM RoAD dynamic model are presented in the next chapter.

The evolution of vehicle dynamics models allows designers to better appreciate the relationships between the vehicle and roadway. The point-mass model is still regarded by AASHTO (2001) and TAC (1999) design guides as a reasonable estimation of vehicle dynamics on a 2D alignment despite its well-documented shortcomings. The reluctance to utilize a more sophisticated dynamics model is perhaps due to the increased complexity. However, with the increased role of computers and software packages more efforts should be made in using more accurate models if it can be confirmed that the results they produce are indeed warranted or superior to the simple point-mass estimation.

The National Cooperative Highway Research Program (NCHRP) has recently completed publication of Synthesis 299 (2001) titled 'Recent Geometric Design Research for Improved Safety and Operations'. This synthesis provides a good overview of geometric design research conducted in the 1990's and discusses more succinctly aspects of the literature review presented herein. Finally, it should be noted that only limited documentation is presented in the NCHRP Synthesis (2001) in the study area pertaining to this thesis. The research conducted by Harwood et al. (1994) and Felipe and Navin (1999) round out the discussion within the Synthesis on horizontal curves and the implications of vehicle stability on combined alignments are not discussed (NCHRP 2001).

CHAPTER 3

3.0 DATA COLLECTION

This chapter examines the data collection process used in this research. Section 3.1 begins by introducing the computer simulation software, VDM RoAD that was developed by UMTRI. The software limitations are discussed as are the assumptions that were made to facilitate the data collection. A discussion of the rationale used in the modeling procedure is also presented in Section 3.1. Section 3.2 expands on the modeling procedure discussed in Section 3.1 by highlighting the geometry of the 3D test alignments and the geometry of the 2D model calibration alignments. In addition, the characteristics of the Ford Taurus, the selected test vehicle, are briefly described to provide an appreciation of the required input needed to perform vehicle dynamics computer simulation. Finally, Section 3.3 documents the application of the software to combined alignments. A sample of the lateral acceleration plot used to gather data on flat, upgrade, downgrade and other combined alignment combinations is presented.

3.1 VDM RoAD Software

In the late 1980's the FHWA initiated development of the IHSDM to help with the evaluation of safe, cost-effective highway design alternatives. Two contracts were awarded by the FHWA in pursuit of this goal and the VDM RoAD software is the result of one of those contracts. VDM RoAD is capable of simulating the 3D vehicle dynamics behavior of cars, trucks and other two-unit combination vehicles on a user

defined road alignment. The software package is fairly sophisticated in that it is "...almost ideal for simulating what would happen if a particular vehicle is operated on a particular road with a given input path and speed." (Sayers 1999).

As would be expected with such a comprehensive simulation package, the amount of input data needed to use the software is extensive. Not only is roadway geometric data needed, but also quantities of detailed vehicle characteristics are required for a successful simulation run. Vehicle input data such as steering kinematics, suspension systems, nonlinear tire models in addition to more trivial items such as mass and wheelbase are all requisite. Fortunately, VDM RoAD provides data for several example vehicles taken from the AASHTO (1994) guide. A Ford Taurus is also included and it was this vehicle that was selected for trial simulation runs throughout the course of this research.

The Ford Taurus was selected as the test vehicle for two main reasons. First, the vehicle data on the Taurus was included with VDM RoAD but also its properties were recently measured to support other research work that is being conducted at the National Highway Traffic Safety Administration (NHTSA) on validation of computer vehicle dynamics models (Sayers 1999). Second, midsize sedans have been used in full-scale tests where researchers have had the option to use alternatives (Carlson et al. 1999, Felipe et al. 1999, Morrall et al. 1994). The Ford Taurus "...is assumed to be a reasonable representation of the fleet of passenger cars on American roads..." (Carlson et al. 1999).

3.1.1 Modeling Procedure

The basic modeling procedure used to simulate a vehicle navigating a roadway in VDM RoAD software is relatively logical, however, some of the details in each step of the procedure can be complicated. In order to successfully complete a dynamics simulation using the software, the user essentially completes a sequence of events as follows:

1. Roadway data must be selected and input in the proper format
2. A test vehicle complete with the required characteristics must be selected/input
3. A constant or varying test speed needs to be chosen for the trial run
4. The lateral position of the vehicle relative to the roadway centreline is needed
5. The simulation is run
6. The results are viewed through a wire frame animator or graphical plots

Each of these events will be explored in some detail in order to better explain the modeling procedure that was involved in the data collection phase of this research. It should be noted that sequence listed above must be completed each time a change is made in road geometry, vehicle characteristics, vehicle speed, or vehicle steering path.

Figure 3.1 illustrates the modeling procedure described above in a flowchart. Once viewing the results of the simulation run, the user can then rerun or restart the modeling sequence altering the input parameters as necessary in order to achieve the desired result.

Figure 3.1: Modeling Procedure Overview

Roadway geometry is described in a space delimited text file that contains 38 columns of data. The format, although quite cumbersome, was mandated by the FHWA in assigning the contract and the opportunity to simplify was not given to the software developer. As a minimum, two points can be used to describe a straight line or tangent alignment and this requires two lines of data in the roadway geometry file.

More complex alignments require more lines in the data file. A new line of data is necessary for changes in horizontal or vertical alignment, superelevation and roadway width. Figure 3.2 shows a portion of the geometry data file, known as an interchange

Input: Highway Data (IHSDM)

Data set: 4200u

Category: 5% Upgrade

IHSDM Highway Data File

Highway Design Data (Filename): C:\VDM_ROAD\ROADS\UPGRAD-14-200U.JH

Find IHSDM File

File Contents

--- IHSDM DATABASE ---

Job Number: tra Chain Name: ALT1 Initial Heading:

Number Regions: 1 Start Station: 0+000 End Station:

Station	X	Y	Z	Radius	DAngle	SAngle	VClen
0.000	500.000	500.000	100.000	0.00	0.00000	0.00	0.00
100.000	500.000	600.000	105.000	0.00	0.00000	0.00	0.00
126.400	500.000	626.400	106.320	0.00	0.00000	0.00	0.00
192.400	500.000	692.400	109.620	200.00	-45.00000	0.00	0.00
263.400	495.272	735.207	113.170	200.00	-45.00000	0.00	0.00
305.480	469.029	799.020	115.274	200.00	-45.00000	0.00	0.00
349.500	441.421	833.821	117.475	0.00	0.00000	0.00	0.00
415.480	394.752	880.491	120.774	0.00	0.00000	0.00	0.00
441.880	376.084	899.159	122.094	0.00	0.00000	0.00	0.00

Animator Links

Reference Frame: Fixed

Animator Data: 4200u

Figure 3.2: Sample of Geometric Alignment Data in VDM RoAD

file format that is used by VDM RoAD. As indicated in Figure 3.2, data requirements include stationing, X-Y-Z coordinates (S-L-Z coordinate projection described below), deflection angles, vertical curve lengths, and horizontal curve radii to name a few. The portion of the sample file shown in Figure 3.2 contains 8 of the 38 required data fields for each 3D centreline point used in the geometric alignment.

Integrating the vehicle dynamics model with the roadway data is a key component of accurately simulating a test run. The road design data performs the following functions

in the VDM RoAD simulation process (Sayers 1999):

1. Provides a continuous 3D surface to support and interact with the tires of the vehicle
2. Provides a horizontal path for the steering model to follow in VDM RoAD
3. Determines the initial conditions of the vehicle including location and orientation at the start of the simulation run

To locate a 3D point in space, X, Y and Z coordinates are all that is required, however VDM RoAD references the vehicle location relative to the roadway centreline using an S, L and Z coordinate system. The Z coordinates between systems are identical therefore transformations between the two systems involve the X-Y and S-L systems only.

Figure 3.3 shows the relationship between the coordinate systems integrated within VDM RoAD in the road design data. The 3D design line defined with the X, Y and Z

Figure 3.3: The S-L-Z and X-Y-Z Coordinate Systems (Sayers 1999)

coordinates is projected onto a 2D surface. The start of the line, representing the start of the design alignment, is given an S coordinate of 0 (zero). Stationing or distance is then measured along the 2D design line projection in meters or feet as required. The L coordinates are a perpendicular offset from the 2D projection of the design line. The right hand rule is employed so that when facing upchain (the direction of increasing S), points to the left of the design centreline will have a positive L coordinate. Points to the right of the centreline will have a negative L coordinate.

The right hand rule system is advantageous in this instance since for both systems, the Z coordinate is defined as being positive when moving upwards. This allows a more simplified 2D transformation to occur somewhat simplifying the creation of the IHSDM input files. Many commercial road design software packages employ the right handed coordinate system making it familiar to engineers and others who are likely to create and want to experiment with vehicle dynamics simulation (Sayers 1999).

The S-L-Z and X-Y-Z coordinate systems are not difficult to master, especially by individuals who have some background in roadway design or surveying. The coordinate transformation concept has been discussed because it is an essential component of the roadway input files and without proper referencing of the roadway and creation of these files, the simulation solvers in VDM RoAD will not operate.

The geometric alignment files are arduous to produce, even for simple alignments, since they must be created by hand. In addition, the quantity of redundant information

required by this type of file format makes it difficult "...for debugging and performing what-if analysis." (Sayers 1999). A complete sample input file for a 300 m radius 2D alignment with 8% superelevation can be found in Appendix A. There are other problems associated with the interchange file format that deal mainly with the increased computing time associated with reading in large data files. These types of problems will not be discussed further since they are largely unrelated to the research at hand and would contribute little to understanding the overall modeling procedure that was used.

Although the software developers do not document it explicitly, an impressive amount of research and effort must have gone into the vehicle models that are used by VDM RoAD software. These models were originally created by mechanical engineers for refining vehicle components and characteristics and were not intended solely for vehicle-road interaction analysis (Sayers 1999). This level of sophistication, although potentially very accurate, also requires that a considerable amount of information be known about a particular vehicle before it can be used in a simulation run.

Detailed vehicle data on steering systems, front and rear suspensions, tires, aerodynamics, inertia properties in addition to basic weights and dimensions are all required inputs. Figure 3.4 shows the main vehicle data screen within VDM RoAD. Note that every data field shown in the figure located next to an arrow, leads to another screen that is similar in appearance requiring input data for that particular item.

Car's

Date set: **Taurus#1**

Category:

mass center of entire vehicle

542

2690

dimensions are in millimeters

Masses and Inertia Front and Rear

	Front	Rear	
Vehicle mass supported by tires:	1049.0	655.7	kg
Unsprung mass (both sides):	98.1	79.7	kg
Spin moment of inertia for one wheel:	0.99	0.99	kg-m2

Inertia Properties: Entire Vehicle

Roll Inertia (I _{xx}):	555.5	kg-m2
Pitch Inertia (I _{yy}):	2832.7	kg-m2
Yaw Inertia (I _{zz}):	3048.1	kg-m2
Product (I _{xz}):	0.0	kg-m2

(Optional) Radii of Gyration

Estimate I _{xx}	R _{xx}	Estimate I _{yy}	R _{yy}	Estimate I _{zz}	R _{zz}
	.8		.9		1

Estimate inertia with eq. $I = M \cdot R^2$

Front Suspension: Taurus Front Suspension

Rear Suspension: Taurus rear suspension

Front Tire: table-lookup model: Taurus Tires

Rear Tire: table-lookup model: Taurus Tires

Front Brakes: Taurus front

Rear Brakes: Taurus rear

Steering System: Taurus Steering

Aerodynamics: Taurus Aero Data (drag only)

Animator Group: Generic Car (normal size)

Notes:

Figure 3.4: Sample of Vehicle Data in VDM RoAD

Following the input and selection of roadway geometry and vehicle data, the next steps in the modeling sequence involve the integration of physical or tactile decisions that in the real world are made by the driver. These decisions are mainly speed and steering controls. These two inputs are relatively straightforward as far as the modeling procedure is involved. Both can vary as a function of roadway stationing or alternately remain a constant throughout the simulation.

The software uses speed and lateral positioning input and in turn produces the corresponding torque and steering wheel angles that are consistent with the “full

dynamic response of the vehicle.” (Sayers 1999). The controller algorithm within VDM RoAD then does whatever is necessary to strictly follow the speed and steering values that were input. If the algorithm attempts to navigate an alignment using speeds and steering inputs that are beyond the capabilities of the roadway or automobile, then the vehicle crashes (Sayers 1999). A crash in this sense refers to either a vehicle rollover or a skid off the road surface depending on vehicle roadway interaction.

Next in the modeling procedure 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. The solver program reads in the files previously described (roadway, vehicle, speed and lateral position) as input for the numerical integration that used to solve the corresponding equations of motion.

The Runge-Kutta numerical method is used by the solver program to perform the necessary calculations (Sayers 1999). Runge-Kutta integration was selected because of its good accuracy compared to Euler and Modified Euler methods. This improved accuracy is the result of taking smaller step sizes and VDM RoAD provides the advanced user with the opportunity to adjust the step size. For this research the default step size of 0.002 seconds was used for all data runs.

Figure 3.5 shows the main data screen in VDM RoAD where access to input data, run data and output of post-processing data can be obtained. A user essentially works from left to right on this screen to perform a simulation run and view the results of the run.

Figure 3.5: Sample of Main Run Screen in VDM RoAD

It is from this main run screen that the user can access the vehicle models, input highway geometry, vehicle speed and positioning relative to the roadway centreline. Various utility functions within the program can be accessed from the toolbar at the top of the run screen. These functions allow the user to quickly access the data libraries built into VDM RoAD to facilitate efficient modification to the simulation parameters as required. The 'Data set' and 'Category' fields found near the top of the screen are specified by the user for identification purposes and quick reference. All associated model and input parameters are saved in a group for any given simulation run.

Figure 3.6: Output Plots and Wire-Frame Animation

The output and post-processing features in VDM RoAD allow the user to view the simulation run using a wire-frame animator or alternately produce a variety of graphical plots. Figure 3.6 shows a sample of the output plots as well as a sample animator frame from a simulation run.

The numerous predefined output plots include tire and steering responses, vehicle motion and tracking and select road geometry to name a few. The wire-frame animator, although rudimentary, does allow the user to view vehicle responses from a variety of angles. When the output plots and animation are combined together, it can provide a useful alignment evaluation tool to road designers.

The preceding discussion has trivialized the modeling procedure to some extent yet the basic sequence of steps have been presented in the order consistent with a user performing an actual simulation. As for any technical software, care must be taken by the user in understanding the required inputs and mechanism used by the software in solving the problem so that the results are as meaningful as possible.

Certain assumptions have been made to facilitate this research using VDM RoAD and these will be highlighted in the next section, as will some of the limitations associated with the software and data collection. In addition, a brief discussion about the rationale used to select test alignments will be presented to provide a more thorough explanation on the selection of alignments that were actually used in the data collection phase.

3.1.2 Limitations and Assumptions

Perhaps the most significant assumption made in this research is the use of a Ford Taurus to adequately represent the North American passenger vehicle fleet. This assumption is also the greatest limitation. However, given the scope of this thesis it is more than sufficient to meet the study objectives. As noted earlier, the Ford Taurus has been used by other researchers for full-scale tests and given the availability of the required input vehicle data it was the only logical choice for the simulation work that was conducted.

Completed in 1997, the National Highway Traffic Safety Administration's (NHTSA) Vehicle Research and Test Centre (VRTC) documented the entire data set for a 1994 Ford Taurus. Various parameters were measured and tested experimentally to ensure that the resulting model was accurate and valid for complete vehicle dynamics simulation (Salanni et al. 1997, Chrstos et al. 1997). The results of the VRTC model for the Ford Taurus have been incorporated into VDM RoAD software resulting in a proved and precise simulation model (Sayers 1999).

The VDM RoAD Ford Taurus was used for all simulation runs in this research providing vehicle consistency between the different alignments under consideration. By keeping the simulated vehicle constant, any changes in vehicle response could be attributed to alignment effects only allowing a more meticulous data analysis to be performed.

It is well known that drivers tend to steer a spiral or transitional path upon entering a horizontal curve. This statement is reasonable since it is physically impossible to drive along a tangent, and then upon traversing the start of the horizontal curve, to instantaneously steer the exact radius of the curve without steering some sort of transition between the two. Modeling this type of driver behavior is more closely related to the area of human factors expertise, however, VDM RoAD software accounts for this in its driver model.

The driver model in VDM RoAD uses two parameters, namely the preview time and driver lag time to adjust how the vehicle is steered. The preview time refers to amount of time or roadway distance viewed in advance of the current location. The driver lag time is used to "...simulate the neuro-muscular delay..." that drivers experience when the brain issues a command (Sayers 1999). Sayers (1999) recommends for realistic driver control that a value of 1 and 0.15 seconds be used for preview time and driver lag time respectively. These values were used consistently throughout the data collection phase.

Another important assumption made during the course of this research was the pavement condition and associated maximum side friction factor or coefficient. A dry pavement with an average roughness was simulated by assuming the maximum amount of side friction that the pavement is capable of providing is equal to 0.9. This value is consistent with maximum friction factors obtained by other researchers for dry pavements (Morrall et al. 1994, Harwood et al. 1994).

The findings of this research are easily transferable to wet pavement conditions by simply comparing the output obtained against the appropriate side friction factor. A good estimation of the maximum friction that results from pavement tire interaction on wet pavements is approximately 0.58 for speed of 30 km/hr while a lower value of 0.41 is more representative of 113 km/hr speeds (Harwood et al. 1994). One of the major problems in estimating maximum side friction coefficients for wet pavements is that there is currently no accurate method for measuring water depths on pavements (Morrall et al. 1994). Vehicle stability on combined alignments is discussed in a subsequent chapter for both wet and dry pavement conditions.

The other assumptions that were made are associated mainly with the geometric features of the roadway. The normal crown or pavement cross slope was taken in all cases to be 2%. Superelevation was provided as recommended in the TAC (1999) guide with 60% of the required length distributed on the tangent and the remaining 40% being introduced on the curve itself. The tangent runout was removed at a constant rate of 1:400 again consistent with TAC (1999) procedures.

It should also be noted that the research objectives were considered paramount and the test alignments chosen for the analysis attempt to reflect this. Efforts were taken in selecting horizontal and vertical combinations that, although within the acceptable limits of current geometric design guides, potentially amplify the problematic issues associated with vehicle stability on combined alignments.

Two existing examples of curvilinear alignments were examined to ensure that the test alignments studied were at least representative of current designs in place or proposed for Canadian roadways. The Wasagamack-St. Teresa access road in Northern Manitoba and a portion of the Sea-to-Sky Highway in British Columbia were selected for this purpose. These two-lane rural highway facilities provide excellent examples of combined alignments. All the test alignments used in the data collection process, including the related alignment combinations that were examined as a result of reviewing these Canadian facilities, will be discussed in Section 3.2.

3.2 Test Alignments

Numerous alignment combinations were used in the data collection process to ensure the results obtained were as meaningful as possible. A series of alignments were produced based on the TAC (1999) minimum horizontal curve radius guidelines for all design speeds equal to and greater than 60 km/hr listed in Table 2.7. These test alignments included 4, 6 and 8% superelevation rates, and were produced in 2D as a base or control group in order to assist with model calibration and validation process.

Superelevation runoff and tangent runout was applied in accordance with the recommended TAC (1999) procedures for this control group and all other alignments considered. Deflection angles in all cases were selected as 45-degrees because this angle resulted in a sufficiently large curve length for the maximum superelevation rate to be achieved along a portion of the curve. Also, under some circumstances, smaller

deflection angles would have prevented reaching the design superelevation rate due to runoff lengths and this would have caused problems in interpreting results due to lack of consistency.

The control group alignments (0% grade) described above were used as a basis for comparison to 3D alignments. The same horizontal, superelevation and tangent runout geometry used for the base alignments was then superimposed on constant favorable and adverse vertical alignments. Constant grades of +6% and -6% were selected since this grade line was the controlling limit used on the Trans Canada Highway.

Moreover, advance-warning grade signs are typically not provided on grades less than 6% (TAC MUTCD 1998). Grades in excess of 6% were not considered in the analysis since the effects of signing on driver behavior would then need to be taken into account. The human factors component of driving on adequately signed combined alignments is beyond the scope of this research.

The selection of appropriate sag and crest curves to superimpose on the horizontal curves was perhaps the most challenging aspect of formulating the test alignments. Suitable entry and exit grades in addition to reasonable curve lengths were needed and since the potential geometric combinations are essentially endless, the selection process was quite difficult. The TAC (1999) guide was consulted to determine the suggested K values for crest and sag curves. Stopping sight distance K values governed design on crest curves while headlight control was the limiting factor for sag curves.

Table 3.1: Crest and Sag K Values - Test Alignments

Design Speed (km/hr)	SSD Crest K Value	Headlight Control Sag K Value
60	13	18
70	23	25
80	36	32
90	53	40
100	80	50
110	110	62
120	150	73
130	200	85

The TAC (1999) guide provides a range of minimum K values for all crest and sag vertical curves to encourage the design engineer to think rather than ‘table-pick’. The ranges provided are calculated based on the anticipated operating speed for the facility in question. Upper end K values coincide with the design speed while lower end values are representative of an operating speed that falls below the design speed.

Table 3.1 shows the values used to create the test alignments for this research. The minimum K values selected are the maximum range values provided in TAC (1999). In other words, taking the 130 km/hr design speed for crest curves for example, the minimum K value range indicated in TAC (1999) is 95-200. The value of 200 was selected for use in the test alignments as shown in Table 3.1.

Although using the lower end range values may have provided a more pronounced impact on lateral acceleration and vehicle stability it was felt that matching the geometry to the design speed would be more representative of actual design conditions.

In addition, since TAC (1999) does not provide a range for the minimum horizontal curve radius based on operating speed, an exact match based on design speed was deemed to be most appropriate for the test alignments.

Having established suitable K values for both crest and sag vertical curves, the next step involved selection of realistic entry and exit grades. An almost endless number of grade combinations exist so efforts were taken in selecting combinations that were realistic. The review of the Wasagamack-St. Teresa access road in Northern Manitoba and a portion of the Sea-to-Sky Highway in British Columbia were very useful in obtaining such combined test alignments.

Recall that A denotes the maximum algebraic difference in entry and exit grades as can be seen in Equation 2.8 which defines the K value. Very flat entry and exit grades with opposite algebraic signs results in a lower value for A and subsequently a smaller vertical curve length. The opposite is true for steep entry and exit grades with different signs. After careful deliberation, it was decided that a value of A equal to 8 would be used for both the crest and sag curve combinations.

Figure 3.7 shows a frequency histogram that presents the percentage difference between vertical curve entry and exit grades. Vertical curves that had opposite entry and exit grades on the Wasagamack-St. Teresa and section of the Sea-to-Sky Highway Alignments were used to create the histogram. Over half of the vertical curves in these alignments have opposite sign entry and exit grades within 1% of each other.

Figure 3.7: Frequency Histogram Showing Entry and Exit Grade Difference

On this basis, equal, opposite sign entry and exit grades for both crest and sag vertical curves were selected for analysis in this research. A value of 8 for the algebraic difference between grades was selected providing equal entry and exit grades of 4% for the test alignments. It was not deemed appropriate to select a larger value for the difference between grades since extreme topography would be needed for such an alignment making it an exception rather than a common occurrence. A value smaller than 8 was also not chosen since equally distributing lower values to the exit and entry grades would result in relatively flat gradients.

Flat gradients may not accentuate the potential problematic issues on combined alignments sufficiently to be discernable from 2D alignments and therefore would be of no use in supporting the research objectives.

Table 3.2: Test Alignments

Design Speed (km/hr)	Super-elevation Rate (%)	Minimum Horizontal Curve Radius (m)	Test Alignment K Values				
			0% Grade (2D road) _a	Constant +6% grade _a	Constant -6% grade _a	A = 8 Crest Vertical Curve	A = 8 Sag Vertical Curve
60	4	150	-	-	-	13	18
70	4	200	-	-	-	23	25
80	4	280	-	-	-	36	32
90	4	380	-	-	-	53	40
100	4	490	-	-	-	80	50
60	6	130	-	-	-	13	18
70	6	190	-	-	-	23	25
80	6	250	-	-	-	36	32
90	6	340	-	-	-	53	40
100	6	440	-	-	-	80	50
110	6	600	-	-	-	110	62
120	6	750	-	-	-	150	73
130	6	950	-	-	-	200	85
60	8	120	-	-	-	13	18
70	8	170	-	-	-	23	25
80	8	230	-	-	-	36	32
90	8	300	-	-	-	53	40
100	8	390	-	-	-	80	50
110	8	530	-	-	-	110	62
120	8	670	-	-	-	150	73
130	8	830	-	-	-	200	85

^a Not Applicable

Table 3.2 summarizes the test alignments considered for the various design speeds and superelevation rates. Five different alignments were considered for each design speed, superelevation rate and minimum horizontal curve radius combination.

3.2.1 Model Calibration

Since this research is essentially based on vehicle-roadway interaction, both the vehicle and the roadway models were examined to ensure model calibration. The vehicle model was tested and calibrated by the VRTC division of the NHTSA prior to its

integration into VDM RoAD. The cumbersome interchange file format used to specify roadway geometry made the calibration procedure difficult and time consuming since trial and error was in essence the only method available for the process. Simple 2D alignments were created initially and using the various graphical output facilities provided by VDM RoAD, they were examined and modified as needed.

One of the most useful plots used to calibrate the roadway models was the vertical profile followed by the vehicle axle as shown in Figure 3.8. The upper figure shows the overall roadway under consideration. There is a substantial vertical exaggeration factor used in the plots, which is why the profile appears more dramatic than the actual 4% entry and exit grades it represents. The sag vertical curve that occurs near the middle of the alignment is superimposed on a horizontal curve complete with superelevation although this is not discernable from the vertical profile shown in Figure 3.8.

The lower plot in Figure 3.8 shows a close-up view of the alignment in the middle of the sag curve. In this case for a right hand turn, the upper profile line represents the front left vehicle tire while the lower line represents the front right vehicle tire. Using the vertical difference between the profile lines, and knowing the vehicle axle width, the superelevation on the roadway surface can be determined. This type of trial and error process was used to confirm the 4, 6 and 8% superelevation rates used in the test alignments.

Figure 3.8: Overall and Close-up Views of a Roadway Profile

Similarly, other graphical outputs provided by VDM RoAD were used to verify that the vehicle model was navigating the desired roadway model. Vertical tire loads for the vehicle axle were examined to ensure, that even though the correct superelevation was modeled on the roadway, that in fact the vehicle was following the roadway geometry. Finally, once it could be ascertained that the vehicle model was interacting properly with the desired roadway geometry, the wire frame animator was used to view the trial runs from a variety of angles. This served to increase the user level of comfort by providing a visual confirmation of vehicle-roadway interaction during a given simulation run.

3.2.2 Application to 3D Alignments

Using the test alignments previously described, simulation runs were performed using the Ford Taurus vehicle model. Test runs were initiated at the design speed for the alignment under consideration and then increased in 5 km/hr increments until a vehicle speed of 20 km/hr over the design speed was reached. Values of peak lateral acceleration were recorded for each speed run on each test alignment. Figure 3.9 shows lateral acceleration plotted against roadway stationing for a given test run.

The rectilinear line in Figure 3.9 represents the friction factor value obtained through manipulation of the point-mass formula while the curvilinear line is the resulting lateral acceleration obtained based on the 3D vehicle dynamics model within VDM RoAD. The curve radius, superelevation and vehicle test speed are used by VDM RoAD to

Figure 3.9: Lateral Acceleration vs. Roadway Stationing

calculate point-mass lateral acceleration values, while roadway geometry and the differential equations of motion are used to estimate the 3D vehicle dynamics values.

Following the point-mass line (rectilinear profile) in Figure 3.9 from left to right, shows the roadway model alignment consisting of a tangent, followed by a horizontal curve, followed by another tangent. The curvilinear function indicates that the lateral acceleration actually experienced by the vehicle does not coincide precisely with the

subject alignment: point-mass estimation. The difference between the two profiles can be attributed to a few factors that will be discussed in the next chapter.

A plot similar to that shown in Figure 3.9 was produced for every test alignment and vehicle speed combination examined. These plots were collected digitally, viewed and data from them was recorded in order to obtain the peak lateral acceleration values that were used in assessing and comparing vehicle stability on combined alignments.

CHAPTER 4

4.0 RESULTS ANALYSIS

This chapter examines the collected data to investigate the effects of combined horizontal and vertical alignments on vehicle stability. Section 4.1 documents the lateral acceleration differences between 2D and 3D alignments and highlights the most problematic combinations out of the test alignments considered. In addition, the effects of overdriving and pavement condition on vehicle stability are discussed. Section 4.2 takes the findings of Section 4.1 one step further by using the 3D lateral acceleration values to calculate minimum horizontal radii. A simple back analysis is performed to estimate the required radius based on 3D vehicle dynamics and on maintaining the TAC (1999) recommended friction factors.

4.1 2D vs. 3D Lateral Acceleration

A total of 105 test alignments were created including both the 21 2D base group alignments and the 84 combined 3D alignments. Peak lateral acceleration values were obtained for each simulation run beginning with the design speed and increasing in 5 km/hr increments until 20 km/hr over the design speed was reached. This methodology resulted in 525 peak lateral acceleration values in both 2D and 3D.

Complete data tables for the test alignment combinations can be found in Appendix A. Table 4.1 shows the data obtained for the 2D base group alignments.

Table 4.1: Simulation Results for 2D Base Group Alignments

Values from the TAC (1999) Guide					VDM RoAD Results				
Design Speed (km/hr)	e_{max} (m/m)	$f_{tolerable}$	$e + f$	Min. Radius (m)	Peak Lateral Acceleration (g's) for overdriving the design speed of the curve by:				
					0 km/hr	5 km/hr	10 km/hr	15 km/hr	20 km/hr
60	0.04	0.15	0.19	150	0.192	0.228	0.268	0.312	0.358
70	0.04	0.15	0.19	200	0.200	0.233	0.268	0.305	0.345
80	0.04	0.14	0.18	280	0.191	0.218	0.247	0.278	0.310
90	0.04	0.13	0.17	380	0.182	0.205	0.229	0.254	0.280
100	0.04	0.12	0.16	490	0.177	0.197	0.218	0.240	0.262
60	0.06	0.15	0.21	130	0.222	0.264	0.310	0.359	0.414
70	0.06	0.15	0.21	190	0.211	0.246	0.283	0.322	0.363
80	0.06	0.14	0.20	250	0.214	0.245	0.277	0.311	0.347
90	0.06	0.13	0.19	340	0.203	0.229	0.256	0.284	0.313
100	0.06	0.12	0.18	440	0.197	0.219	0.242	0.267	0.292
110	0.06	0.10	0.16	600	0.178	0.196	0.214	0.234	0.254
120	0.06	0.09	0.15	750	0.172	0.187	0.204	0.221	0.238
130	0.06	0.08	0.14	950	0.161	0.175	0.189	0.203	0.218
60	0.08	0.15	0.23	120	0.241	0.286	0.336	0.389	0.450
70	0.08	0.15	0.23	170	0.236	0.275	0.316	0.359	0.406
80	0.08	0.14	0.22	230	0.233	0.266	0.301	0.338	0.376
90	0.08	0.13	0.21	300	0.230	0.259	0.289	0.321	0.354
100	0.08	0.12	0.20	390	0.222	0.247	0.273	0.306	0.329
110	0.08	0.10	0.18	530	0.201	0.221	0.242	0.265	0.288
120	0.08	0.09	0.17	670	0.192	0.209	0.228	0.247	0.267
130	0.08	0.08	0.16	830	0.184	0.199	0.215	0.232	0.249

Inspection of the VDM RoAD results shows that even for the simulation runs performed at the design speed, the resulting peak lateral acceleration values are larger than the maximums recommended in the TAC (1999) guide. The difference between values results from the transient effects of a vehicle traveling along a horizontal curve. Recall that a vehicle steers through a tangent to circular curve transition even on simple horizontal curve alignments. The peak lateral acceleration occurs at a distance into the curve when oversteering is at the maximum. These transient effects, due to changes in road geometry, include driver steering, vehicle off tracking and vehicle roll (Sayers 1999). Figure 4.1 shows an overview and detailed lateral acceleration values plotted

Figure 4.1: Lateral Acceleration vs. Stationing – 20km/hr over design speed

against roadway stationing. The geometric characteristics used to create Figure 4.1 include a 4% superelevation and a minimum horizontal radius of 280 m. Three simulation runs have been overlaid on the rectilinear point-mass estimation as shown in Figure 4.1.

The upper frame in Figure 4.1 shows the rectilinear and curvilinear profiles as discussed in Chapter 3. The three curvilinear profiles are more clearly seen in the lower frame where from top to bottom, a sag curve, crest curve and the 2D alignment are represented respectively. While the shapes of the three curvilinear profiles are similar, the vertical shift between them is the result of roadway geometry differences among the test alignments. The test speed, steering, and other vehicle components were kept constant for all three test runs.

The slight rise visible in the rectilinear point-mass estimation is due to slight fluctuations in the vehicle speed (Sayers 1999). These changes are a direct result of the adjustments made by the vehicle model as it negotiates the horizontal curve. These variations are essentially negligible since the magnitude of change is very small.

As can be seen in Figure 4.1, the maximum or peak lateral acceleration values occur a distance into the curve and not instantaneously at the beginning of the curve. This is consistent with the findings of Morrall (1994) who conducted full-scale tests using a ball bank indicator and G-analyst to obtain lateral acceleration values. Harwood and Mason (1994) also found that the maximum lateral acceleration experienced by a

vehicle occurs several hundred feet into a curve and can exceed the values estimated with the point-mass formula by approximately 0.02 g's. This is again consistent with the results obtained from VDM RoAD when simulating a vehicle run on a test alignment at or above the design speed.

4.1.2 Effect of Overdriving

Figure 4.2 shows the simulated lateral acceleration values obtained from VDM RoAD plotted against the point-mass estimations for a typical alignment. The divergence between the lines shows that the point-mass estimate becomes less representative of 3D vehicle dynamics values as the overdriving speed increases.

Figure 4.2: 2D Lateral Acceleration vs. Overdriving Speed

Figure 4.3 shows both 2D and 3D peak lateral acceleration values plotted against the minimum horizontal curve radius. In addition, the recommended maximum side friction or lateral acceleration values from TAC (1999) have also been included. The data used to create Figure 4.3 is based on an 8% superelevation rate driven at the design speed for all minimum horizontal curve radii. It can be seen in all cases, including the 2D, 0% grade alignment, that negotiating the test alignment resulted in lateral acceleration values larger than the recommended TAC (1999) threshold values.

Figure 4.3: Design Speed Lateral Acceleration vs. Horizontal Curve Radius

Figure 4.4: Overdriving Peak Lateral Acceleration vs. Horizontal Curve Radius

Figure 4.4 shows the same information as Figure 4.3 except that the test alignment was navigated at 20 km/hr above the corresponding design speed. The effect of overdriving further increases the differential between the recommended TAC (1999) lateral acceleration values and those obtained based on 3D vehicle dynamics in VDM RoAD.

Overdriving, which is a common occurrence on many roadways, can potentially reduce vehicle stability by increasing friction demand. Figures 4.3 and 4.4 were based on an 8% superelevation rate and were simulated using the design speed and 20 km/hr over the design speed respectively. Similar pairs of plots for the 4% and 6% superelevation rates can be found in the Appendix. Overdriving the test alignments by a 5 km/hr

Table 4.2: 2D vs. 3D Lateral Acceleration – Driven at Design Speed

Minimum Radius (m)	e (m/m)	TAC $e + f$	Peak Lateral Acceleration Values (g's) – Roadway Driven at the Design Speed				
			0% Grade	6% Grade	-6% Grade	Crest	Sag
150	0.04	0.19	0.192	0.193	0.192	0.196	0.189
200	0.04	0.19	0.200	0.197	0.199	0.199	0.197
280	0.04	0.18	0.191	0.187	0.190	0.190	0.188
380	0.04	0.17	0.182	0.177	0.181	0.180	0.177
490	0.04	0.16	0.177	0.172	0.176	0.175	0.172
130	0.06	0.21	0.222	0.223	0.222	0.227	0.220
190	0.06	0.21	0.211	0.207	0.210	0.211	0.207
250	0.06	0.20	0.214	0.209	0.213	0.213	0.210
340	0.06	0.19	0.203	0.198	0.202	0.201	0.198
440	0.06	0.18	0.197	0.192	0.196	0.195	0.191
600	0.06	0.16	0.178	0.173	0.178	0.176	0.166
750	0.06	0.15	0.172	0.166	0.172	0.169	0.170
950	0.06	0.14	0.161	0.156	0.162	0.158	0.160
120	0.08	0.23	0.241	0.242	0.241	0.245	0.238
170	0.08	0.23	0.236	0.232	0.234	0.237	0.231
230	0.08	0.22	0.233	0.228	0.231	0.232	0.228
300	0.08	0.21	0.230	0.224	0.229	0.229	0.225
390	0.08	0.20	0.222	0.216	0.221	0.221	0.217
530	0.08	0.18	0.201	0.195	0.200	0.199	0.195
670	0.08	0.17	0.192	0.186	0.192	0.190	0.180
830	0.08	0.16	0.184	0.178	0.184	0.181	0.165

increment less than 20 km/hr only served to shift the test alignment curves closer to the TAC (1999) values and therefore have not been included.

Table 4.2 shows the simulation results for the test alignments considered using the design speed that corresponds to the minimum horizontal curve radius. The lateral acceleration values shown in bold text govern. Coincidentally, there are several instances of alignment combinations where the peak lateral acceleration is the same for two different test alignments. As noted, the largest values of lateral acceleration are predominantly found in the 2D alignment scenario. However, the crest curve and

Table 4.3: 2D vs. 3D Lateral Acceleration – 20 km/hr over the Design Speed

Minimum Radius (m)	e (m/m)	TAC $e + f$	Peak Lateral Acceleration Values (g's) – Roadway Driven 20 km/hr over the Design Speed				
			0% Grade	6% Grade	-6% Grade	Crest	Sag
150	0.04	0.19	0.358	0.353	0.359	0.364	0.355
200	0.04	0.19	0.345	0.339	0.344	0.343	0.341
280	0.04	0.18	0.310	0.303	0.307	0.308	0.305
380	0.04	0.17	0.280	0.274	0.277	0.279	0.274
490	0.04	0.16	0.262	0.256	0.258	0.261	0.255
130	0.06	0.21	0.414	0.409	0.415	0.407	0.408
190	0.06	0.21	0.363	0.357	0.363	0.361	0.358
250	0.06	0.20	0.347	0.340	0.345	0.346	0.341
340	0.06	0.19	0.313	0.306	0.309	0.312	0.304
440	0.06	0.18	0.292	0.285	0.287	0.291	0.282
600	0.06	0.16	0.254	0.247	0.251	0.253	0.241
750	0.06	0.15	0.238	0.231	0.236	0.236	0.234
950	0.06	0.14	0.218	0.211	0.216	0.215	0.219
120	0.08	0.23	0.450	0.444	0.451	0.443	0.442
170	0.08	0.23	0.406	0.400	0.406	0.406	0.401
230	0.08	0.22	0.376	0.369	0.375	0.376	0.370
300	0.08	0.21	0.354	0.346	0.351	0.353	0.347
390	0.08	0.20	0.329	0.321	0.324	0.328	0.321
530	0.08	0.18	0.288	0.280	0.283	0.286	0.277
670	0.08	0.17	0.267	0.258	0.263	0.265	0.251
830	0.08	0.16	0.249	0.241	0.247	0.247	0.227

downgrade test alignments are also associated with several maximum values. The upgrade and sag curve test alignments were found to be less critical than the others when simulated at the design speed.

Table 4.3 shows peak lateral acceleration values when the test alignments are simulated using a vehicle speed equal to 20 km/hr over the design speed. The 0% grade or 2D alignment again provided the majority of governing values. The downgrade, crest and sag vertical curve combinations also resulted in lateral acceleration maximums.

Consistent with the simulation results at the design speed, the 6% upgrade test alignment did not provide any governing lateral acceleration values.

It is clear that the limiting lateral acceleration values provided in current North American geometric design guides are exceeded even when a vehicle is traveling at the design speed and on level terrain (0% grade). Overdriving an alignment further increases the differential between the recommended maximum lateral acceleration and the actual lateral acceleration experienced by a driver. This fact alone does not necessarily mean a vehicle will have stability issues resulting in a skid off the road or rollover type accident but as drivers become more tolerant towards larger lateral acceleration their implicit margin of safety is reduced. This will be explored further in the following section, which deals with vehicle stability considerations on both wet and dry pavements.

4.1.2 Vehicle Stability Considerations

Both AASHTO (2001) and TAC (1999) geometric design guides limit the unbalanced lateral acceleration on horizontal curves based on research conducted in the late 1930's and early 1940's. This research focused mainly on comfort limits experienced by drivers and passengers within the vehicle and did not concern itself with the maximum tire-pavement friction that could be developed.

Unlike the comfort limits on lateral acceleration mentioned above, vehicle stability is very much associated with the maximum tire-pavement friction that can be developed. Morrall (1994) found that the maximum amount of side friction on dry pavement to be approximately 0.9 g's while Harwood and Mason (1994) suggest that a value of 0.94 g's provides a good indication of the maximum value. A value of 0.9 was used in this research as the maximum side friction value. Since the lateral acceleration results obtained in all cases, including the overdriving simulations, were well below this value as far as skidding stability is concerned, the precise value is of little consequence.

Inherent in the use of a maximum dry pavement side friction factor is the assumption of good pavement and tire condition, the assumption that the vehicle is steering a radius equal to the radius of the curve and that no emergency or erratic driving maneuvers are required. Smooth or polished pavements, bald tires, oversteering the horizontal curve, lane changing, braking or swerving all affect the lateral acceleration demands placed on the vehicle and if severe enough could cause stability problems. Although a thorough examination of these factors is beyond the scope of this research a designer should be aware of their potential implications on vehicle stability.

The maximum side friction factor that can be obtained on wet pavements is difficult to ascertain for two main reasons. First, a standard method for determining water thickness on pavements has not been developed and second, the superelevation present on a horizontal curve makes it impossible for the water layer to remain a uniform thickness (Morrall 1994). Harwood and Mason (1994) suggest that a maximum side

Table 4.4: Side Friction Provided by Wet Pavement – Various Speeds

Speed (km/hr)	Maximum Side Friction Factor – Wet Pavement
60	0.520
70	0.498
80	0.477
90	0.455
100	0.434
110	0.412
120	0.391
130	0.369

friction factor on wet pavements of 0.580 at 32 km/hr and 0.406 for 113 km/hr is reasonable.

Linearly interpolating between the Harwood and Mason (1994) values and extrapolating for speeds higher than 113 km/hr results in the maximum wet pavement friction factors shown in Table 4.4. Unlike the maximum lateral acceleration value suggested for dry pavement conditions, the maximum side friction provided by wet pavements is thought to deteriorate with increased vehicle speed. The wet pavement value for 130 km/hr is almost a third of the maximum attainable value on a dry pavement.

Using data from Tables 4.2, 4.3 and 4.4, a plot was created to illustrate the potential vehicle stability concerns on wet pavements for passenger vehicles. Figure 4.5 shows the side friction maximums assumed for wet pavements and the output from the 8% superelevation simulation runs both at the design speed and 20 km/hr over the design speed.

Wet Pavement Side Friction vs. Design Speed

Figure 4.5: Wet Pavement Side Friction vs. Design Speed

As can be seen in Figure 4.5, the simulated lateral acceleration values fall below the available side friction values for wet pavements. Overdriving the design speed appears to be more critical for minimum horizontal radius alignments with a lower design speed since the differential between curves is increased slightly at higher design speeds.

Similar to the assumptions made for dry pavements, the effects of oversteering and erratic or emergency maneuvers still apply to the wet pavement scenario. As expected, vehicle skidding stability is much lower on a wet pavement than a dry one.

Assuming a dry pavement maximum side friction factor of 0.9, it can be seen that for speeds over 90 km/hr that the wet pavement maximum is less than half. This dramatic reduction in side friction supply if combined with less than optimal pavement or tire conditions has the potential to result in vehicle instabilities especially when the design speed for the facility in question is exceeded.

Based on the preceding discussion it appears that the current geometric design guides are functioning as intended by providing a margin of safety to vehicles traveling on wet pavements with respect to skidding instability. The margin of safety is perhaps less than intended, since the lateral acceleration experienced by a vehicle traveling at the design speed exceeds the recommended maximums in the current guides for the particular design speed and associated horizontal curve radius.

The other major form of vehicle instability is rollover and although rollover tends to be more critical with respect to trucks, a brief synopsis of the issue will now be presented in light of the simulation results obtained.

Passenger vehicles tend to have very high rollover thresholds in the order of 1.2 g's compared to unstable trucks that can rollover at 0.26 to 0.34 g's (Harwood et al. 1994, FHWA 1990). Under most circumstances a passenger vehicle will skid off of the roadway long before it would rollover. This does not mean that rollover could not occur once the vehicle leaves the traveled roadway and contacts a tripping obstacle such

as a curb or other object within the right-of-way. However, from a lateral acceleration point of view only, a rollover event is not likely for passenger vehicles.

All simulated lateral acceleration values obtained from the test alignments were well under the 1.2 g passenger vehicle rollover threshold and therefore rollover instability to passenger vehicles is not critical for the combined test alignments considered. Certain trucks on the other hand would have significant rollover stability issues especially when overdriving the alignment. Although truck instability is beyond the scope of this research, consideration should be given to all anticipated vehicle types that may operate on an alignment during the design stage.

Based on the simulation results obtained and the current skidding and rollover thresholds for passenger vehicles, stability does not appear to be a problem under dry or wet pavement conditions on minimum horizontal curves and combined alignments. The recommended maximum lateral acceleration values were exceeded on both 2D and 3D alignments suggesting that current geometric design guides are providing less than the intended margin of dynamic safety to vehicles operating on these low-end standard facilities.

One of the main objectives of this research was to determine the differences between 2D and 3D lateral acceleration values on minimum horizontal radii curves. The statements made with respect to vehicle stability considerations apply only to minimum

horizontal curves since the lateral acceleration that would result on larger radii curves would be lower than the values collected herein.

Finally, stability concerns on larger radii curves would be reduced and the margin of safety provided on these larger curves would be greater all else remaining the same.

There is also no evidence in the literature to suggest that the Ford Taurus is the most or least stable passenger vehicle and obviously no extension of the results found here can be made to other vehicle types without further investigation.

4.2 Modification to Minimum Horizontal Curve Radius

It was shown in the last section that the Taurus passenger vehicle exceeded the design guideline maximum lateral acceleration values on minimum radii horizontal curves even when traveling at the design speed. It should be noted that the lateral acceleration levels are exceeded for a relatively short period of time and do not occur for the entire duration of the horizontal curve. Recall the curvilinear lateral acceleration profile shown in Figure 4.1. Many of these lateral acceleration versus stationing plots were reviewed and it was determined on average that the simulated vehicle exceeds the point-mass value of lateral acceleration for 1 second when traveling at the design speed and 1.5 seconds when traveling 20 km/hr over the design speed.

The transient effects that cause the lateral acceleration to exceed guideline values, due to changes in road geometry, include driver steering, vehicle off tracking and vehicle

roll. All of these factors slightly alter the resulting lateral acceleration that can be computed based only on road geometry and vehicle speed. While these effects occur in a relatively short period of time, perhaps less than what is required by the driver to perceive and react, they do result in a larger minimum horizontal radius than suggested by the current guides.

A simple back analysis is described in the following section which uses the peak lateral acceleration values obtained from the simulation runs performed at the design speed for each test alignment to calculate a new horizontal curve radius. The new horizontal radii values are such that the current guideline thresholds will not be exceeded for a given vehicle speed and superelevation rate. The intent is to present suitable revised minimum horizontal curve radii that are valid for both 3D vehicle dynamics and combined alignments. It must be stressed that the revised horizontal curve radii presented are not recommended for design since they are based on a single vehicle simulation.

4.2.1 Back Analysis

It was not deemed appropriate to use any of the lateral acceleration values obtained in the overdriving simulations for this analysis since the resulting curve radii would be far too conservative under all but the most extreme circumstances. A road designer should not be required to accommodate drivers who blatantly disregard the rules of the road to the point where they are traveling far over the posted speed limit. It is common practice

to design a road using a design speed roughly 10 km/hr over the intended posted speed (AASHTO 2001, TAC 1999).

The governing lateral acceleration values from Table 4.2 were used in the analysis process and the following simple example will show how the new minimum horizontal curve radii were calculated. The values obtained from Table 4.2 are the unbalanced lateral acceleration values and the analysis presented here is based on maintaining the current superelevation rates of 4, 6 and 8%.

A horizontal curve with a design speed of 100 km/hr and a maximum superelevation rate of 6% has a maximum tolerable lateral acceleration ($e+f$) of 0.18 in accordance with TAC (1999). The minimum horizontal curve radius based on this input data using Equation 2.25 is:

$$R_{\min} = \frac{100^2}{127(0.06 + 0.12)} = 437.4m$$

This radius is rounded up to 440 m which is the value listed in TAC (1999). The peak lateral acceleration based on 3D vehicle dynamics was found to be 0.197 as shown in Table 4.2. Dividing the guideline lateral acceleration value by the simulated value yields:

$$\frac{0.18}{0.197} = 0.91371$$

This ratio when multiplied by 100% results in the percentage of the guideline lateral acceleration value that should be used to calculate the minimum horizontal curve radius in order to limit the lateral acceleration to the established levels. In other words, 91.37

% of 0.18 should be used in Equation 2.25 to calculate the new minimum horizontal radius in the calculation.

$$0.91371(0.18) = 0.164$$

The new minimum horizontal radius is:

$$R_{\min} = \frac{100^2}{127(0.164)} = 480.1m = 480m(\text{rounded})$$

Therefore running a simulation with a 480 m radius, 6% superelevation rate and 100 km/hr vehicle speed, the summation of resulting side friction and superelevation components total 0.18. This is consistent with the limiting comfort threshold values provided in the TAC (1999) guide. There are other, perhaps more direct ratio calculations that would yield the same results as shown above but this method provides a good interaction between the variables involved and is believed to improve the overall understanding of the process.

As shown in Table 4.5, increases to the TAC (1999) minimum horizontal curve radii up to 16.6% are required to limit the lateral acceleration levels to those recommended in the guideline. Since there are no stability concerns on dry or even wet pavements with the current minimums, the increases noted above are based on maintaining user comfort rather than vehicle control. Vehicle stability would be improved given that a 3D vehicle dynamics simulation on larger radii curves would yield lower lateral acceleration values. Even though the majority of the governing values $e + f$ values occurred on 2D alignments it should be emphasized that the revised radii presented in Table 4.5 are based on 3D vehicle dynamics and maintaining comfort control limits.

Table 4.5: Revised Requirements of the Minimum Horizontal Curve Radius

Design Speed (km/hr)	TAC Minimum Horizontal Radius	<i>e</i> (m/m)	TAC <i>e + f</i>	Design Speed Governing 3D <i>e + f</i>	Calculated Minimum Radius	Rounded Minimum Horizontal Radius	Percentage (%) Increase from TAC Radius
60	150	0.04	0.19	0.196	154	160	6.7
70	200	0.04	0.19	0.200	214	220	10.0
80	280	0.04	0.18	0.191	297	300	7.1
90	380	0.04	0.17	0.182	402	410	7.9
100	490	0.04	0.16	0.177	544	550	12.2
60	130	0.06	0.21	0.227	146	150	15.4
70	190	0.06	0.21	0.211	185	190	0.0
80	250	0.06	0.20	0.214	270	270	8.0
90	340	0.06	0.19	0.203	359	360	5.9
100	440	0.06	0.18	0.197	480	480	9.1
110	600	0.06	0.16	0.178	662	670	11.6
120	750	0.06	0.15	0.172	867	870	16.0
130	950	0.06	0.14	0.162	1100	1100	15.8
60	120	0.08	0.23	0.245	131	140	16.6
70	170	0.08	0.23	0.237	173	180	5.9
80	230	0.08	0.22	0.233	243	250	8.7
90	300	0.08	0.21	0.230	333	340	13.3
100	390	0.08	0.20	0.222	437	440	12.8
110	530	0.08	0.18	0.201	591	600	13.2
120	670	0.08	0.17	0.192	753	760	13.4
130	830	0.08	0.16	0.184	956	960	15.7

The percentage increases were calculated by comparing the rounded 3D dynamics radii values with those presented in the TAC (1999) guide. The percentage increases are not uniform nor are they consistent between superelevation rates. This is due to the rounding that has occurred in the point-mass calculation of the TAC (1999) values and the rounding associated with the values based on 3D vehicle dynamics.

Generally, the larger increases are associated with the higher design speeds. Note that in all cases the calculated radii values were rounded up to the nearest value of 10. For example, 402 m was rounded up to 410 m and 359 m was rounded to 360 m.

Figure 4.6 illustrates the radii differences for the 4, 6 and 8% superelevation rates for each of the corresponding design speeds. As noted, the differences between radii are more distinct at the higher design speeds.

It is evident that even for the slower design speed that 3D vehicle dynamics results in minimum horizontal curve radii larger than those presented in the TAC (1999) guide. The following section discusses the revised minimum horizontal curve radius and design speed in a little more detail and presents a comparison between the minimum horizontal radii obtained from 3D vehicle dynamics, and the TAC (1999) and Chang (2001) formulations.

Figure 4.6: TAC (1999) and 3D Dynamics Minimum Curve Radii

4.2.2 Curve Radius and Design Speed

Recall in Section 2.5 that research by Chang (2001) was presented where modifications to the minimum horizontal curve radius were suggested based on the inclusion of vehicle body roll in dynamics calculations. He recommended modified point-mass formulae to account for the sprung body mass.

Based on the VDM RoAD lateral acceleration values, which use a sophisticated vehicle dynamics model and facilitates simulation on 3D roadway alignments, modifications to the minimum horizontal curve radii were calculated in the preceding section. Figure 4.7 shows a comparison of the TAC (1999), Chang (2001) and 3D dynamics values for minimum horizontal curve radius for a 6% superelevation rate.

The horizontal curve radii based on the 3D vehicle dynamics and lateral acceleration values fall between the Chang (2001) and TAC (1999) or AASHTO (2001) values. Unlike the Chang (2001) formulation, no modified point-mass formula has been presented since the increases suggested here apply only to the minimum horizontal radii curves and are not required on larger curves because the 3D lateral acceleration would be much lower on larger curves. Similar figures showing the curve radii differences for 4% and 8% superelevation rates can be found in Appendix A.

Based on the variables in the point-mass formula it was decided that modification to only the horizontal curve radius was appropriate in maintaining the lateral acceleration

Figure 4.7: Different Formulations for Minimum Curve Radius

levels recommended in the current guides. Although increases in superelevation or reduction in design speeds would have ultimately achieved the same results of limiting the lateral acceleration it was not deemed to be a realistic solution.

Since it is obvious based on the research results presented here that the limiting values of lateral acceleration in the guides are based on comfort rather than vehicle stability, a reexamination of comfort thresholds should be conducted. A study of this kind is overdue since the values found in the current North American guides are based on research that was conducted in the late 1930's and early 1940's. Vehicle design and dynamics have changed significantly in the last 70 years and perhaps the associated user comfort threshold has changed as well.

CHAPTER 5

5.0 CONCLUSIONS AND RECOMMENDATIONS

Computer simulation was used in this research to examine vehicle stability through lateral acceleration levels with the main objective of comparing the minimum flat horizontal curve radius used in traditional geometric design guides with the minimum radius based on three-dimensional vehicle dynamics and alignments. Conventional geometric design practice considers horizontal and vertical road elements separately. Furthermore, the design vehicle is reduced to a point-mass in order to simplify vehicle dynamics calculations.

A vehicle dynamics simulation program called VDM RoAD (Vehicle Dynamic Models Roadway Analysis and Design) developed by the University of Michigan Transportation Research Institute was used to examine vehicle dynamics and stability on 3D alignments. The program uses a sophisticated two-axle dynamics model that traditionally has only been used by mechanical engineers in experimenting with vehicle characteristics.

A simulated Ford Taurus was the vehicle used during the data collection phase for two main reasons. First, the required input data such as suspension, tire, steering kinematics and axle information was readily available and recently verified by the National Highway Traffic Safety Administration's Vehicle Research and Test Centre. Second, the Ford Taurus is an average passenger vehicle that adequately represents the standard

North American passenger vehicle fleet and mid-size sedans, like the Taurus, have been used in full-scale tests by numerous researchers.

Flat, minimum radius, simple horizontal curves superelevated at 4, 6, and 8 % were used as the base control group. Vehicle dynamics were recorded for these 2D alignments in order to facilitate a comparison with subsequent 3D alignments. Minimum horizontal radii curves were selected as subject alignments because the superelevation would reach the maximum rate during a portion of the curve in all cases. Deflection angles were kept constant at 45 degrees in order to provide a sufficiently large curve length to facilitate development of maximum superelevation while still complying with suggested tangent run out and superelevation runoff lengths.

The 3D alignment groups consisted of the same base 2D alignment elements with various overlapping vertical alignments. Constant upgrades and downgrades were overlaid on the base 2D alignments as were crest and sag vertical curves. In all cases, various vehicle speeds were examined with the design speed for the horizontal curve in question being the lowest. In all, a total of 105 test alignments were considered and with the various vehicle speeds examined, a total of 525 lateral acceleration values were obtained from both 2D and 3D alignments.

It was found that vehicle stability is not compromised on combined alignments compared to 2D alignments for the test cases examined. However differences in minimum horizontal radii were obtained due to the transient effects of a vehicle

traveling along a horizontal curve. These transient effects, due to changes in road geometry, include driver steering, vehicle off tracking and vehicle roll. All of these factors slightly alter the resulting lateral acceleration that can be computed based only on road geometry and vehicle speed. While these effects occur in a relatively short period of time, perhaps less than what is required by the driver to perceive and react, they do result in a larger minimum horizontal radius than suggested by the current design guides.

In order to maintain the recommended comfort threshold suggest by the current North American geometric design guides, increases in minimum horizontal radii up to 16.6 % are required. In general, it was found that the higher design speeds require radii increases on the high end of the range, while the opposite is true of the lower design speed alignments. The 3D vehicle dynamics revisions to the minimum horizontal curve radii apply only to the minimum values since lateral acceleration levels are much lower for curve with larger radii.

Based on the findings of this research and on some of the issues that developed in exploring the background material, the following recommendations are provided:

- Design consistency between horizontal and vertical alignment elements should be promoted in both the AASHTO and TAC design guides. Quantitative measures of consistency for operating speed on combined alignments, driver workload and alignment indices should find their way into future design guides.

- There is a strong need to review and research comfort thresholds for today's vehicles. Other researchers have expressed this in the past. However, the evolution of geometric design guides is seriously hindered by current knowledge of lateral acceleration effects on human comfort in modern vehicles. At the very least, it should be determined whether or not a ball bank angle of 10 degrees still causes discomfort.
- Additional research should be conducted on the effects of environmental conditions, particularly snow and ice, on vehicle stability for the larger superelevation rates. TAC (1999) limits superelevation rates to 8% to mitigate inward sliding by vehicles on horizontal curves under adverse pavement conditions. Vehicle stability in some cases may benefit from greater superelevation on horizontal curves.
- Truck stability on combined alignments should be examined. This research has focused on the passenger vehicle, in particular the Ford Taurus, therefore an extension to truck and trailer combinations should be considered. It was shown through discussion of skidding and rollover thresholds that passenger vehicles and trucks respond differently to lateral acceleration and these differences may be significant enough to impact design criteria for combined alignments.
- Other 3D alignment combinations should be considered. A relatively limited variety of combined alignments were investigated in this research. While the objective was not to provide final revisions to horizontal curve design criteria, the work would benefit with the examination of additional alignments. It must be stressed that the revised horizontal radii values presented herein are not recommended for design since they are based on a single vehicle simulation.

- This research used a sophisticated (and accurate) vehicle dynamics model and compared the output of this model against approximate friction or lateral acceleration values obtained many years ago. A comparison of this nature is less than ideal as accurate model outputs are better compared to accurate lateral acceleration thresholds. Additional research should be conducted into quantifying accurate lateral acceleration thresholds.
- Finally, the results of this research clearly showed that 3D dynamics do impact alignment elements and may in fact play a role in the improvement and development of geometric design guides. In the future, 3D design heuristics should find their way into the TAC and AASHTO documents.

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APPENDIX A

Table A1: Lateral Acceleration Data 0 % Grade (2D Alignment)

Values from TAC (1999) Guidelines					VDM RoAD Results				
Design Speed (km/hr)	e_{max} (m/m)	$f_{tolerable}$	$e + f$	Min. Radius (m)	Peak Lateral Acceleration (g's) for overdriving the design speed of the curve by:				
					0 km/hr	5 km/hr	10 km/hr	15 km/hr	20 km/hr
60	0.04	0.15	0.19	150	0.192	0.228	0.268	0.312	0.358
70	0.04	0.15	0.19	200	0.200	0.233	0.268	0.305	0.345
80	0.04	0.14	0.18	280	0.191	0.218	0.247	0.278	0.310
90	0.04	0.13	0.17	380	0.182	0.205	0.229	0.254	0.280
100	0.04	0.12	0.16	490	0.177	0.197	0.218	0.240	0.262
60	0.06	0.15	0.21	130	0.222	0.264	0.310	0.359	0.414
70	0.06	0.15	0.21	190	0.211	0.246	0.283	0.322	0.363
80	0.06	0.14	0.20	250	0.214	0.245	0.277	0.311	0.347
90	0.06	0.13	0.19	340	0.203	0.229	0.256	0.284	0.313
100	0.06	0.12	0.18	440	0.197	0.219	0.242	0.267	0.292
110	0.06	0.10	0.16	600	0.178	0.196	0.214	0.234	0.254
120	0.06	0.09	0.15	750	0.172	0.187	0.204	0.221	0.238
130	0.06	0.08	0.14	950	0.161	0.175	0.189	0.203	0.218
60	0.08	0.15	0.23	120	0.241	0.286	0.336	0.389	0.450
70	0.08	0.15	0.23	170	0.236	0.275	0.316	0.359	0.406
80	0.08	0.14	0.22	230	0.233	0.266	0.301	0.338	0.376
90	0.08	0.13	0.21	300	0.230	0.259	0.289	0.321	0.354
100	0.08	0.12	0.20	390	0.222	0.247	0.273	0.306	0.329
110	0.08	0.10	0.18	530	0.201	0.221	0.242	0.265	0.288
120	0.08	0.09	0.17	670	0.192	0.209	0.228	0.247	0.267
130	0.08	0.08	0.16	830	0.184	0.199	0.215	0.232	0.249

Table A2: Lateral Acceleration Data –6% Grade (3D Alignment)

Values from TAC (1999) Guidelines					VDM RoAD Results				
Design Speed (km/hr)	e_{max} (m/m)	$f_{tolerable}$	$e + f$	Min. Radius (m)	Peak Lateral Acceleration (g's) for overdriving the design speed of the curve by:				
					0 km/hr	5 km/hr	10 km/hr	15 km/hr	20 km/hr
60	0.04	0.15	0.19	150	0.192	0.227	0.267	0.311	0.359
70	0.04	0.15	0.19	200	0.199	0.231	0.265	0.304	0.344
80	0.04	0.14	0.18	280	0.190	0.217	0.245	0.275	0.307
90	0.04	0.13	0.17	380	0.181	0.203	0.226	0.251	0.277
100	0.04	0.12	0.16	490	0.176	0.196	0.216	0.237	0.258
60	0.06	0.15	0.21	130	0.222	0.262	0.308	0.359	0.415
70	0.06	0.15	0.21	190	0.210	0.243	0.280	0.320	0.363
80	0.06	0.14	0.20	250	0.213	0.243	0.274	0.308	0.345
90	0.06	0.13	0.19	340	0.202	0.227	0.253	0.280	0.309
100	0.06	0.12	0.18	440	0.196	0.218	0.240	0.263	0.287
110	0.06	0.10	0.16	600	0.178	0.195	0.213	0.232	0.251
120	0.06	0.09	0.15	750	0.172	0.187	0.202	0.219	0.236
130	0.06	0.08	0.14	950	0.162	0.175	0.188	0.202	0.216
60	0.08	0.15	0.23	120	0.241	0.284	0.334	0.389	0.451
70	0.08	0.15	0.23	170	0.234	0.272	0.313	0.358	0.406
80	0.08	0.14	0.22	230	0.231	0.263	0.297	0.335	0.375
90	0.08	0.13	0.21	300	0.229	0.256	0.285	0.317	0.351
100	0.08	0.12	0.20	390	0.221	0.245	0.270	0.296	0.324
110	0.08	0.10	0.18	530	0.200	0.220	0.240	0.261	0.283
120	0.08	0.09	0.17	670	0.192	0.208	0.226	0.244	0.263
130	0.08	0.08	0.16	830	0.184	0.199	0.214	0.230	0.247

Table A3: Lateral Acceleration Data 6% Grade (3D Alignment)

Values from TAC (1999) Guidelines					VDM RoAD Results				
Design Speed (km/hr)	e_{max} (m/m)	$f_{tolerable}$	$e + f$	Min. Radius (m)	Peak Lateral Acceleration (g's) for overdriving the design speed of the curve by:				
					0 km/hr	5 km/hr	10 km/hr	15 km/hr	20 km/hr
60	0.04	0.15	0.19	150	0.193	0.226	0.264	0.307	0.353
70	0.04	0.15	0.19	200	0.197	0.229	0.263	0.300	0.339
80	0.04	0.14	0.18	280	0.187	0.214	0.242	0.272	0.303
90	0.04	0.13	0.17	380	0.177	0.199	0.223	0.248	0.274
100	0.04	0.12	0.16	490	0.172	0.192	0.212	0.233	0.256
60	0.06	0.15	0.21	130	0.223	0.262	0.306	0.355	0.409
70	0.06	0.15	0.21	190	0.207	0.241	0.277	0.250	0.357
80	0.06	0.14	0.20	250	0.209	0.239	0.271	0.305	0.340
90	0.06	0.13	0.19	340	0.198	0.223	0.249	0.277	0.306
100	0.06	0.12	0.18	440	0.192	0.213	0.236	0.260	0.285
110	0.06	0.10	0.16	600	0.173	0.190	0.208	0.227	0.247
120	0.06	0.09	0.15	750	0.166	0.181	0.197	0.214	0.231
130	0.06	0.08	0.14	950	0.156	0.169	0.182	0.196	0.211
60	0.08	0.15	0.23	120	0.242	0.283	0.332	0.385	0.444
70	0.08	0.15	0.23	170	0.232	0.270	0.310	0.353	0.400
80	0.08	0.14	0.22	230	0.228	0.260	0.295	0.331	0.369
90	0.08	0.13	0.21	300	0.224	0.253	0.283	0.314	0.346
100	0.08	0.12	0.20	390	0.216	0.240	0.266	0.293	0.321
110	0.08	0.10	0.18	530	0.195	0.215	0.235	0.257	0.280
120	0.08	0.09	0.17	670	0.186	0.203	0.220	0.239	0.258
130	0.08	0.08	0.16	830	0.178	0.193	0.208	0.224	0.241

Table A4: Lateral Acceleration Data Crest Curve (3D Alignment, A=8)

Values from TAC (1999) Guidelines					VDM RoAD Results				
Design Speed (km/hr)	e_{max} (m/m)	$f_{tolerable}$	$e + f$	Min. Radius (m)	Peak Lateral Acceleration (g's) for overdriving the design speed of the curve by:				
					0 km/hr	5 km/hr	10 km/hr	15 km/hr	20 km/hr
60	0.04	0.15	0.19	150	0.196	0.231	0.272	0.315	0.364
70	0.04	0.15	0.19	200	0.199	0.232	0.267	0.303	0.343
80	0.04	0.14	0.18	280	0.190	0.217	0.246	0.277	0.308
90	0.04	0.13	0.17	380	0.180	0.203	0.227	0.253	0.279
100	0.04	0.12	0.16	490	0.175	0.195	0.216	0.238	0.261
60	0.06	0.15	0.21	130	0.227	0.266	0.309	0.357	0.407
70	0.06	0.15	0.21	190	0.211	0.244	0.281	0.319	0.361
80	0.06	0.14	0.20	250	0.213	0.244	0.276	0.310	0.346
90	0.06	0.13	0.19	340	0.201	0.227	0.254	0.283	0.312
100	0.06	0.12	0.18	440	0.195	0.217	0.241	0.265	0.291
110	0.06	0.10	0.16	600	0.176	0.194	0.213	0.233	0.253
120	0.06	0.09	0.15	750	0.169	0.185	0.201	0.218	0.236
130	0.06	0.08	0.14	950	0.158	0.171	0.185	0.200	0.215
60	0.08	0.15	0.23	120	0.245	0.289	0.335	0.387	0.443
70	0.08	0.15	0.23	170	0.237	0.275	0.316	0.359	0.406
80	0.08	0.14	0.22	230	0.232	0.265	0.300	0.337	0.376
90	0.08	0.13	0.21	300	0.229	0.258	0.289	0.320	0.353
100	0.08	0.12	0.20	390	0.221	0.246	0.272	0.299	0.328
110	0.08	0.10	0.18	530	0.199	0.219	0.241	0.263	0.286
120	0.08	0.09	0.17	670	0.190	0.207	0.225	0.245	0.265
130	0.08	0.08	0.16	830	0.181	0.197	0.213	0.229	0.247

Table A5: Lateral Acceleration Data Sag Curve (3D Alignment, A=8)

Values from TAC (1999) Guidelines					VDM RoAD Results				
Design Speed (km/hr)	e_{max} (m/m)	$f_{tolerable}$	$e + f$	Min. Radius (m)	Peak Lateral Acceleration (g's) for overdriving the design speed of the curve by:				
					0 km/hr	5 km/hr	10 km/hr	15 km/hr	20 km/hr
60	0.04	0.15	0.19	150	0.189	0.225	0.264	0.308	0.355
70	0.04	0.15	0.19	200	0.197	0.229	0.263	0.309	0.341
80	0.04	0.14	0.18	280	0.188	0.214	0.243	0.305	0.305
90	0.04	0.13	0.17	380	0.177	0.200	0.223	0.248	0.274
100	0.04	0.12	0.16	490	0.172	0.191	0.219	0.237	0.255
60	0.06	0.15	0.21	130	0.220	0.259	0.304	0.354	0.408
70	0.06	0.15	0.21	190	0.207	0.241	0.277	0.316	0.358
80	0.06	0.14	0.20	250	0.210	0.240	0.271	0.305	0.341
90	0.06	0.13	0.19	340	0.198	0.222	0.249	0.276	0.304
100	0.06	0.12	0.18	440	0.191	0.212	0.235	0.258	0.282
110	0.06	0.10	0.16	600	0.166	0.183	0.201	0.221	0.241
120	0.06	0.09	0.15	750	0.170	0.185	0.206	0.217	0.234
130	0.06	0.08	0.14	950	0.160	0.174	0.189	0.204	0.219
60	0.08	0.15	0.23	120	0.238	0.281	0.330	0.384	0.442
70	0.08	0.15	0.23	170	0.231	0.269	0.310	0.354	0.401
80	0.08	0.14	0.22	230	0.228	0.261	0.295	0.332	0.370
90	0.08	0.13	0.21	300	0.225	0.254	0.283	0.314	0.347
100	0.08	0.12	0.20	390	0.217	0.241	0.267	0.293	0.321
110	0.08	0.10	0.18	530	0.195	0.214	0.234	0.255	0.277
120	0.08	0.09	0.17	670	0.180	0.197	0.214	0.232	0.251
130	0.08	0.08	0.16	830	0.165	0.179	0.194	0.210	0.227

Table A6: Sample VDM RoAD Input Data File (2D Alignment, 300 m Radius, 8% Superelevation)

---		IHSDM		DATABASE		---							
Number:		Chain		Name:		Initial		Heading:					
Regions:		Start		Station:		End		Station:					
Station	X	Y	Z	ALT1	Radius	DAngle	SAngle	VClen	Bgrade	Fgrade	L1Width	L1Type	L1Slope
0	500	500	100	0	0	0	0	0	0	0	0	0	0
100	500	600	100	0	0	0	0	0	0	0	0	0	0
126.4	500	626.4	100	0	0	0	0	0	0	0	0	0	0
236.4	500	736.4	100	300	300	45	0	0	0	0	0	0	0
310.4	508.244	809.431	100	300	300	45	0	0	0	0	0	0	0
398.019	541.002	888.902	100	300	300	45	0	0	0	0	0	0	0
472.119	587.868	948.532	100	0	0	0	0	0	0	0	0	0	0
582.019	665.65	1026.314	100	0	0	0	0	0	0	0	0	0	0
608.419	684.317	1044.981	100	0	0	0	0	0	0	0	0	0	0
708.419	755.028	1115.692	100	0	0	0	0	0	0	0	0	0	0

Table A6 Continued: Sample VDM Road Input Data File (2D Alignment, 300 m Radius, 8% Superelevation)

L2Width	L2Slope	MWidth	MType	MSlope	L3Width	L3Slope	L4Width	L	4Type	L4Slope	LSWidth	LSSlope
3.3	0	0	0	0	3.3	-2	0	0	0	0	0	0
3.3	0	0	0	0	3.3	-2	0	0	0	0	0	0
3.3	0	0	0	0	3.3	-2	0	0	0	0	0	0
3.3	0	0	0	0	3.3	-4.8	0	0	0	0	0	0
3.3	0	0	0	0	3.3	-8	0	0	0	0	0	0
3.3	0	0	0	0	3.3	-8	0	0	0	0	0	0
3.3	0	0	0	0	3.3	-4.8	0	0	0	0	0	0
3.3	0	0	0	0	3.3	-2	0	0	0	0	0	0
3.3	0	0	0	0	3.3	-2	0	0	0	0	0	0
3.3	0	0	0	0	3.3	-2	0	0	0	0	0	0

RSWidth	RSSlope	LBSlope	LBSWidth	LDWidth	LFSlope	LFWidth	RFSlope	RFWidth	RDWidth	RBSlope	RBSWidth
0	0	0	0	0	0	0	0	0	0	0	0
0	0	0	0	0	0	0	0	0	0	0	0
0	0	0	0	0	0	0	0	0	0	0	0
0	0	0	0	0	0	0	0	0	0	0	0
0	0	0	0	0	0	0	0	0	0	0	0
0	0	0	0	0	0	0	0	0	0	0	0
0	0	0	0	0	0	0	0	0	0	0	0
0	0	0	0	0	0	0	0	0	0	0	0
0	0	0	0	0	0	0	0	0	0	0	0
0	0	0	0	0	0	0	0	0	0	0	0

NOTE: INPUT FILE HAS BEEN CUT INTO 3 BLOCKS TO FACILITATE VIEWING ON ONE PAGE. THE ACTUAL INPUT FILE HAS 38 COLUMNS OF DATA.

Figure A1: Peak Lateral Acceleration vs. Curve Radius (4% Superelevation driven at the design speed)

Figure A2: Peak Lateral Acceleration vs. Curve Radius (4% Superelevation driven 20 km/hr over the design speed)

Figure A3: Peak Lateral Acceleration vs. Curve Radius (6% Superelevation driven at the design speed)

Figure A4: Peak Lateral Acceleration vs. Curve Radius (6% Superelevation driven 20 km/hr over the design speed)

Figure A5: Minimum Horizontal Curve Radius vs. Design Speed (4% Superelevation)

Figure A6: Minimum Horizontal Curve Radius vs. Design Speed (8% Superelevation)