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Truck stability on different types of horizontal curves combined with vertical alignments

Essam Mohamed S. A. E. A. Dabbour
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**TRUCK STABILITY ON DIFFERENT TYPES OF HORIZONTAL
CURVES COMBINED WITH VERTICAL ALIGNMENTS**

By

Essam Mohamed S. A. E. A. Dabbour, P. Eng.

B.Sc., Alexandria University, Egypt, 1988

A thesis

Presented to Ryerson University

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

Toronto, Ontario, Canada, 2003

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TRUCK STABILITY ON DIFFERENT TYPES OF HORIZONTAL CURVES COMBINED WITH VERTICAL ALIGNMENTS

Master of Applied Science (2003)

Essam Mohamed S. A. E. A. Dabbour, P. Eng.

Department of Civil Engineering, Ryerson University

ABSTRACT

The combination of horizontal curves with vertical alignments is commonly used in different classifications of highways; either on highway mainstream or on highway interchange ramps. The horizontal curves, combined with vertical alignments, may be single, compound or reverse horizontal curves. The current design guidelines do not adequately investigate vehicle stability on such three-dimensional (3D) alignments. Computer software that simulates vehicle behaviour on different geometrical alignments was employed to investigate vehicle stability on such 3D alignments. It was found that vehicle safety is questionable, especially for larger vehicles on reverse curves associated with vertical alignments. The critical speed, where the vehicle starts to rollover or skid, was found to be close to design speed for those 3D alignments. Design aids were then developed to address the recommended solutions to maintain the margin of safety required.

ACKNOWLEDGEMENTS

Many thanks go to Dr. Said Easa, my thesis supervisor. Dr. Easa gave me all the encouragement, guidance, support and inspiration throughout the study program at Ryerson University. Dr. Easa helped me in my thesis in many ways. He provided me with the thesis idea along with the selected software. He also provided me with many references related to the software to help me. Besides, Dr. Easa also guided me in all the publications related to the thesis that we have published together. This research has been financially supported by an Ontario Graduate Scholarship and a discovery grant from the Natural Sciences and Engineering Research Council of Canada.

My study program at Ryerson gave me excellent opportunity to meet great people with great minds and great attitudes. Those people are the faculty and staff members at Ryerson, including Dr. Mohamed Lachemi, Dr. Khaled Sennah, Dr. B. Persaud, Dr. Ahmed El-Rabbany, Leah Stanwyck and all other faculty and staff members at Ryerson University. I also had great opportunity to meet other students with great attitudes and passion for knowledge. Some of those students eventually became some of my closest friends, including Mohamed El-Diasty, Assem Hassan, Chandi Ganguly, Mahmoud Abd-El-Gelil and Y. F. Li. At last but not the least, I'd like to thank my great parents for their kind support, patience and encouragement.

DEDICATION

This work is dedicated to the three angels who give me love, happiness and hope for better future. To my wife Enas and my two daughters Olaa and Abir.

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CHAPTER 1: INTRODUCTION

Highway geometric design is a complicated process to design highways with different classifications. The design includes such elements as horizontal alignment, vertical alignment, cross section, intersections and interchanges, roadside facilities, and pedestrian and bicycle facilities. The primary consideration of geometric design is safety and smooth operation. The careful selection and coordination of various alignment components can significantly promote safety to the traveling public while maintaining smooth operation. Cost is also a significant consideration. Aesthetic and other environmental considerations play an important role in today's highway geometric design. In addition, there are many design controls that influence the design process, including design vehicle, driver performance, design speed, classification of the highway, topography, climatic conditions, traffic volumes, soil information, and land use.

Policies on highway geometric design in the United States are developed by the American Association of State Highway and Transportation Officials (AASHTO). These policies represent design guidelines agreed to by the state highway and transportation departments and the Federal Highway Administration (FHWA). Guidelines for highway geometric design are presented in "*A Policy on Geometric Design of Highways and Streets*" (AASHTO 2001), which is based on many years of experience and research. In Canada, geometric design guidelines are presented in

"Geometric Design Guide for Canadian Roads" (TAC 1999), which is published by the Transportation Association of Canada (TAC).

This chapter provides background on geometric design for different elements of highways in Section 1.1 and outlines the research problem to be addressed in this thesis in Section 1.2. Section 1.3 states the research objectives while Section 1.4 states the research plan and organizational structure of the remaining chapters of the thesis.

1.1 Background

Highway geometry is defined by horizontal alignment and vertical alignment. Horizontal alignment is the plan view of the highway along its path, while vertical alignment is the profile of the highway that shows its elevations along its path. Horizontal alignment includes tangents (i.e. straight paths without curvature) and circular horizontal curves that connect tangents together either with or without a transition spiral curve. Vertical alignment includes tangents (either flat, upgrade or downgrade) and parabolic curves that connect tangents. Another fundamental of highway geometric design is the highway cross section, which specifies the width and side-slope of the travelled way, shoulders, curbs, medians, sideslopes and backslopes, clear zones, pedestrian facilities, and bicycle facilities. More details on the design guidelines of both horizontal and vertical alignments, and cross sections will be covered later in the thesis.

When a vehicle travels along a horizontal circular curve, it experiences centrifugal force outward the centre of the horizontal curve. This centrifugal force is inversely proportional to the horizontal curve radius. Vehicle stability is achieved by the resistive forces that resist the centrifugal force. These forces include frictional interaction between the tires and pavement, and a component of the vehicle weight that acts parallel to the road surface. The frictional interaction between the tires and pavement depends on road surface side-friction factor, which in turn depends on many other factors, including road surface condition, weather and climatic condition, tire condition, and vehicle kinematics. The component of the vehicle weight that acts parallel to the road surface depends on the side slope of the highway, which is usually referred to as superelevation.

That approach is usually referred to as the point-mass (PM) model, which is adopted by North American design guides due to its simplicity. More details about the point-mass model will be covered in Chapter 2. Based on the point-mass model, when a vehicle travels along a vertical curve, there is obviously no centrifugal force, and consequently no potential risk for skidding or rollover. However, for 3D (combined) alignments, where a horizontal curve is superimposed by a vertical alignment, the vertical alignment affects the available side friction. For 3D alignments, traditional design guides (AASHTO 2001; TAC 1999) calculate the minimum radius assuming a side friction on a horizontal plane using the point-mass model, thus ignoring the effect of vertical alignment. This approach simplifies cornering dynamics by reducing the vehicle into a point mass travelling on a 2D horizontal alignment.

1.2 Problem Definition

The point-mass model, currently used in the design of minimum radius of horizontal curves, although relatively simple, has several limitations. The effect of an overlapping vertical alignment is totally ignored. Since the model approximates a vehicle to a point mass, there is no consideration given to the distribution of frictional forces between the inner and outer or front and rear tires. The point-mass model also is not based on any particular set of vehicle characteristics and is theoretically as applicable to trucks as it is to passenger cars (Harwood et al. 1994). However, in light of the differences between passenger cars and trucks in terms of size, number of tires, tire characteristics, and suspension characteristics, the suitability of the PM assumption for trucks has been re-examined. It was found that although the friction demands at the four tires of passenger car are approximately equal, the friction demands at various tires of a tractor-trailer truck vary widely (FHWA 1985). The net result of this tire-to-tire variation in side friction demand is that trucks typically demand approximately 10% higher side friction than that required for passenger cars.

The point-mass model does not explicitly consider vehicle rollover thresholds. The rollover threshold for passenger cars is relatively high, so a passenger car will normally skid off a road long before it would rollover (McGee 1981). However, tractor-trailer trucks have relatively higher mass centres and consequently tend to

have lower rollover thresholds. Furthermore, because of suspension characteristics, the rollover threshold of these trucks is substantially less than it would be if a truck were a rigid body. Studies have found that most unstable trucks had rollover threshold of about 0.30 g (Harwood et al. 1990). That is, if a truck travels at a minimum curve radius of 39 m with a design speed of 35 km/h, the lateral acceleration is 0.17 g. Then, this truck could undergo an additional lateral acceleration of only 0.13 g without rolling over.

Another limitation in the point-mass model is that it does not account for vehicle stability on complex alignments, including compound and reverse horizontal curves either individually or combined with vertical alignments. Such curves are commonly used in various highway classifications, especially at freeway interchanges. While current design guides provide brief and vague guidelines for the design of compound horizontal curves to have a ratio between the larger and smaller radii not to exceed 1.5, there are no guidelines for the design of reverse curves from the perspective of vehicle stability. In addition, current design guides deal with vehicle stability on compound or reverse curves as two individual curves, each with its own design requirements. Therefore, consistency is lacking in both the individual design of these curves and their 3D nature when they are combined with vertical alignments. The author is not aware of any research work that has been conducted on truck stability on compound or reverse curves.

One of the major limitations in the design of minimum radius of horizontal curves in general is that it is based on values of side friction that were developed over 60 years ago (Barnett 1936; Moyer and Berry 1940). The criterion used in establishing those values was based on the point at which the overturning is sufficient to cause the driver to experience a feeling of discomfort and cause the driver to react instinctively to avoid a higher speed. The speed on a curve, at which discomfort due to the overturning moment is evident to the driver, was accepted as a design control for the maximum allowable value of side friction. The ball-bank indicator has been used as a uniform measure for the point of discomfort, as well as for body roll, to set safe speeds on curves. Applying those design values of side friction for today's passenger cars is questionable. In addition, those values were originally developed for passenger cars and their application to today's fleet of trucks is definitely questionable. This is especially true since the new approach for modern truck design typically tends to provide optimum power/mass ratio for trucks to carry heavier cargo, resulting in higher mass centres and less rollover threshold.

Some research work has been conducted in Canada to evaluate side friction under different weather, climatic, and vehicle fleet conditions. A study in Calgary found that while the design values of side friction provided higher margin of safety for passenger cars and pick-up trucks against skidding on dry pavement, the margin of safety on wet and icy conditions was questionable (Morrall and Talarico 1995; Talarico and Morrall 1994). The authors also found that basing horizontal curve

guidelines on ball-bank angles might not be a conservative approach to highway design, because skid may occur at low speeds before discomfort is reached.

Another experimental study has been conducted in New Brunswick using actual five-axle semitrailer as a design vehicle with more consideration given to the evaluation of rollover threshold (Garcia et al 2003). The study found that the vehicle carrying less-than-truck-load displayed the highest propensity to rollover with some recorded values of lateral acceleration close to 80 to 90% of the corresponding rollover threshold (when traveling at or below the posted speed limit). Clearly, this raises questions about the adequacy of current design guidelines against rollover.

The roll performance of tractor-semitrailer combinations has been investigated using computer simulation on simple horizontal curves to evaluate the effect of transition spiral curves on vehicle stability (Blue and Kulakowski 1991). It was found that spiral curves, which provide a more gradual transition into the horizontal curve, resulted in smoother changes in lateral acceleration and roll angle, and less need for driver correction when the truck is entering the curve. Harwood and Mason (1993) made some recommendations regarding selection of appropriate ramp design speeds and geometries to avoid interchange operational and safety problems associated with driver behaviour. They used computer simulation, which replaced the traditional physical test-vehicle. Computer simulation provides economic and safe method to examine vehicle behaviour for different geometric design variables.

However, if a vehicle travels along a horizontal curve associated (or even not associated) with a vertical alignment, it may be subject to actual forces that exceed those determined by the point-mass model, and the vehicle favourable resistive forces may also be less than those determined by North American design guides (AASHTO 2001; TAC 1999). More details about the different approaches for vehicle stability will be discussed later.

The current design guidelines deal with alignment elements individually, which means if there is a complex horizontal curve, either reverse or compound, the design guidelines deal with it as two different simple horizontal curves. Thus, the vehicle dynamics characteristics on that complex curve are not accounted for. Furthermore, if that complex curve is associated with a vertical alignment, the design guidelines deal with it as separate horizontal and a vertical alignments, which does not account for the vehicle dynamics characteristics on such 3D alignment. However, it is a very common situation where a compound or reverse horizontal curve is superimposed by a vertical alignment forming a 3D alignment, either on the highway mainstream, or on the interchange ramps.

The following problems are defined, and will be addressed in this thesis:

1. The current design guidelines mostly deal with reverse or compound horizontal curves as separate elements and do not give adequate overview for the design requirements for those complex horizontal curves.

2. The current design guides mostly deal with any horizontal alignment associated with a vertical alignment as two separate alignments, each with its own design requirements. Therefore, there is no adequate considerations for the design requirements for the 3D alignments.
3. The lack for adequate guidelines either for reverse or compound horizontal curves by its own, or associated with vertical alignments, resulted in more rollover incidents on both highway curves and interchange ramps.

1.3 Research Objectives

The objectives of the thesis are as follows:

1. To conduct a literature review of the current design guidelines and emerging research work for the research point.
2. To use simulation software (*VDM RoAD*) to evaluate vehicle stability on different combinations of geometric alignments that are designed according to the current design guidelines. The geometric alignments to be evaluated include:
  - a. A single horizontal curve not associated with any vertical alignment (base scenario).
  - b. Base scenario with different types of vertical alignments (downgrade, upgrade, crest curve, and sag curve).
  - c. A reverse curve with minimum radius for the sharper arc equals the base scenario curve (reverse curve scenario).

- d. Reverse curve scenario with different types of vertical alignments (downgrade, upgrade, crest curve, and sag curve).
  - e. A compound curve with minimum radius for the sharper arc equals the base scenario curve (compound curve scenario).
  - f. Compound curve scenario with different types of vertical alignments (downgrade, upgrade, crest curve, and sag curve).
3. To calibrate mathematical models to calculate the required minimum radii for different types and combinations of horizontal curves (simple; reverse; compound). These models are capable of calculating the required minimum radius based on design speeds and geometric alignment data.

It should be noted that the experimental work in the thesis is based on side friction factors for freeways, which are different from those for urban streets or for interchange ramps. Further research is required to interpret the results from this thesis to be used for interchange ramps or urban streets. Many human factors are not covered in the scope of this research, including driver workload and perception-reaction time. Other situations where driver behavior should be investigated are not also covered, such as braking as a reaction to increasing lateral acceleration when it exceeds driver comfort limit, or steering as a reaction to increasing roll angle where a rollover is likely to occur. Aerodynamic forces are not also covered in this thesis. Many other factors that affect vehicle stability are not covered in the scope of this research. These include pavement condition (such as coarse or polished running surfaces) and weather and climatic conditions (such as rain, snow, and ice).

1.4 Thesis Organization

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

- Chapter 2 contains a detailed literature review of horizontal and vertical alignments, cross section, superelevation, interchange ramps and 3D alignments. It also covers design speeds, operating speeds, and posted speeds. Chapter 2 will also cover the current philosophy relating to the coordination of horizontal and vertical alignments and briefly touches on design consistency. An overview of different vehicle stability models and IHSDM project are also covered in Chapter 2.
- Chapter 3 highlights the simulation software used in the experimental work. It gives detailed description of the simulation process in VDM RoAD including data input, simulation processing, and reading the output data either as a wire frame animation or as a series of plots.
- Chapter 4 discusses the criteria used in the selection of all different variables of the simulation process, including selection of the design vehicle, test alignment, and all other considerations in selection criteria.
- Chapter 5 gives full details on simulation procedures for simple horizontal curves combined with vertical alignments, including 2D base scenario and different 3D scenarios for all different alignment combinations. It also gives complete discussion of the results and sensitivity analysis for all the variables involved in the simulation procedures. The resulting mathematical model

relates the required minimum radius, as dependent variable, to all geometric alignment configurations, as independent variables. Thus the minimum radius determined using this model is more reliable and more sensitive to different independent variables.

- Chapter 6 gives full details on simulation procedures for reverse horizontal curves combined with vertical alignments. This chapter gives more details about the simulation process and model calibration for such reverse horizontal curves combined or not combined with vertical alignments, including 2D base scenario and different 3D scenarios for all different alignment combinations. The combinations include the effect of reverse curvature by itself, without being combined with any vertical alignment, and the effect of introducing vertical alignment. It also gives complete discussion of the results and sensitivity analysis for all the variables involved in the simulation procedures. As there are no design guidelines for reverse curves, Chapter 6 highlights the minimum practical design value for the ratio between the radii of the flatter and the sharper arcs that compose the reverse curve.
- Chapter 7 gives full details on simulation procedures for compound horizontal curves combined with vertical alignments. It gives more details about the simulation process and model calibration for such compound curves combined or not combined with vertical alignments, including 2D base scenario and different 3D scenarios for all different alignment combinations. The combinations include the effect of compound curvature by itself, without being combined with any vertical alignment, and the effect of introducing

vertical alignment. It also gives complete discussion of the results and sensitivity analysis for all the variables involved in the simulation procedures.

- Chapter 8 represents the main findings of this research. Conclusions and recommendations to enhance safety are discussed. Areas that require 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: LITERATURE REVIEW

2.1 Background

There are three major elements involved in highway geometric design, namely: Vertical alignments, Horizontal alignments, and Cross sections. Vertical alignments mainly consist of straight alignments (either flat, upgrade or downgrade) connected together by vertical curves. Vertical curves are mainly parabolic curves, either crest or sag. Other options for vertical alignments include unsymmetrical curves and reverse parabolic curves (Easa 2002). Other option for vertical alignment is the curvilinear alignment, which is a series of successive curves with short tangents mainly found on older rural highways.

Horizontal alignments mainly consist of straight tangents connected together by circular horizontal curves, either with or without transitional spiral curves. Horizontal curves are either a simple curve or a complex curve. Complex curve may be a reverse curve, which consists of two successive simple curves in opposite directions, or a compound curve, which consists of two successive simple curves in the same direction.

For balance in highway design, all geometric elements should, as far as economically practical, be designed to provide safe, continuous smooth operation at

a speed likely to be observed under the normal conditions for that roadway (AASHTO 2001). The following sections will describe each of the design elements.

2.2 Vertical Alignments

The topography of the land traversed has an influence on highway alignments. Topography affects horizontal alignments, but has more pronounced effect on vertical alignment. To characterize variations in topography, engineers generally separate it into three classifications according to terrain (AASHTO 2001): -

- In level, highway sight distances are generally long or can be made to be so without construction difficulty or major expense.
- In rolling terrain, natural slopes consistently rise above and fall below the highway grade, and occasional steep slopes offer some restriction to normal horizontal and vertical roadway alignment.
- In mountainous terrain, longitudinal and transverse changes in the elevation of the ground with respect to the road or street are abrupt, and benching and side hill excavation are frequently needed to obtain acceptable horizontal and vertical alignments.

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 2.1 shows crest and sag vertical curves for the various entry and exit grade combinations.

Figure 2.1 Vertical curve combinations [Source: AASHTO 2001]

The vertical curves shown in Figure 2.1 are symmetric, which means 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. 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.

2.2.1 Vertical Curve Equation

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

$$y = ax^2 + bx + c \quad (2.1)$$

where,

- y = roadway elevation along the vertical curve
- x = distance from the BVC or PVC; (m)
- 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.

$$\frac{dy}{dx} = 2ax + b \quad (2.2)$$

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

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

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

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

The rate of change of the slope given by Equation 2.4 above can also be written as

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

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

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

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

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

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

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

Substituting Equation 2.8 into Equation 2.7 gives

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

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.

2.2.2 Vertical Alignments in AASHTO and TAC

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.

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.

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.

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 unfavourable. 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.

2.2.3 New Trends in Vertical Alignments

Asymmetrical vertical curves, although less common than symmetrical curves, are occasionally warranted in geometric design. A traditional asymmetrical vertical curve has unequal projection of its tangents and component parabolic arcs. Easa

(1994) developed a new asymmetrical vertical curve, with unequal projection of its tangents but equal component parabolic arcs, 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 and Hassan 2000). 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."

2.3 Horizontal Alignments

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).

A simple horizontal curve with radius R and deflection angle I is shown in Figure 2.2. The basic elements required for laying out a horizontal curve are tangent distance, T , external distance, E , middle ordinate, M , length of chord, C , and curve length, L (Easa 2002). These elements can be computed in terms of R and I . For example, the tangent distance, T , equals $R \tan (I/2)$ and the curve length, L , equals $\pi RI/180$.

Spiral transitions are sometimes used to introduce the vehicle to the directional change in a more gradual manner. 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.

Figure 2.2 Geometry of simple horizontal curves [Source: Easa 2002]

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.

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). When a vehicle moves in a circular path, it undergoes a centrifugal force that acts outward the center of curvature. This force is sustained by a component of the vehicle's weight related to the roadway superelevation and by the side friction developed between the vehicle's tires and the pavement surface. The centrifugal force is an imaginary force that motorists believe is pushing them outward while cornering while, in fact, they are truly feeling the vehicle being accelerated in an inward direction by centripetal acceleration that acts toward the center of the curvature (AASHTO 2001). However,

as a matter of conceptual convenience, the centrifugal force is used to illustrate vehicle stability as shown in Figure 2.3. 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.

Figure 2.3 Centrifugal and resisting forces on a cornering vehicle

2.3.1 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 2.4 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).

Wong (1978) explains that considering vehicle-cornering behaviour 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 that it is being carried around. Newton's Laws of motion quantify the centrifugal force as mass times acceleration (Scott 1991).

Figure 2.4 Vehicle axis system [Source: Wong 1978]

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.

Figure 2.5 The 7 degree-of-freedom ride model for passenger cars

[Source: Wong 1978]

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 2.5. As shown, the model has independent suspension for the front axle and has seven degrees of freedom (DOF). 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 2.5 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 2.5 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 2.6 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. 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).

Figure 2.6 Two degree-of-freedom ride model [Source: Wong 1978]

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 traction force.
4. Vehicle characteristics such as weight distribution, operating speed and driver behaviour.
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.

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 Morral 1994).

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).

The main factor that affects the cornering behaviour 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_u), which is the derivative of the cornering force with respect to slip angle (Wong 1978).

Tire behaviour 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 of 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).

2.3.2 Vehicle Stability on Horizontal Curves

There are numerous variables involved in vehicle behaviour and cornering, which makes the stability problem 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 2.7 illustrates the radial forces that act on a vehicle as it travels around a constant radius horizontal curve at a constant speed.

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

a = angle of roadway incline (analogous to superelevation (e))

- Q = vehicle weight force (N); $Q = \text{mass} \times \text{gravity acceleration}$
- 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):

$$\frac{v^2}{R} \quad (2.10)$$

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):

$$F = \frac{mv^2}{R} \quad (2.11)$$

Accurate estimates of the geometric variables involved are readily available to the designer. The process used to estimate the values of the resistive and traction 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 favourable 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 the lateral sliding or skidding and the second is the 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 2.7, if the resisting force provided by the radial friction falls below the traction 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 traction forces are less than resisting forces (Lamm et al. 1999).

Recognizing that the radial frictional forces (F_R) that act at the tire-pavement interface are not shown in Figure 2.7, summing forces in the radial direction, parallel to the pavement surface yields:

$$\cos \alpha \left(\frac{mv^2}{R} \right) = F_R + Q \sin \alpha \quad (2.12)$$

Figure 2.7 Vehicle cornering forces – Radial direction

[Source: Lamm et al. 1999]

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:

$$f = \frac{F_R}{N} \quad (2.13)$$

Where N is defined as the vertical force acting normal to the pavement surface. By summing forces in Figure 2.7, N is equal to (Banks 1998):

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

Rearranging Equation 2.13 to solve for F_{rad} and substituting Equation 2.14 and 2.13 into Equation 2.12 yields:

$$\cos \alpha \left(\frac{mv^2}{R} \right) = f \left(\sin \alpha \left(\frac{mv^2}{R} \right) + Q \cos \alpha \right) + Q \cos \alpha \quad (2.15)$$

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

$$\left(\frac{v^2}{gR} \right) = f \tan \alpha \left(\frac{v^2}{gR} \right) + f + \tan \alpha \quad (2.16)$$

Recognizing that $\tan \alpha$ is simply the cross-slope of the road or superelevation (e), Equation 2.16 reduces to:

$$\left(\frac{v^2}{gR}\right) = fe\left(\frac{v^2}{gR}\right) + f + e \quad (2.17)$$

Simplifying further yields:

$$\frac{v^2}{gR}(1 - fe) = f + e \quad (2.18)$$

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 2.18.

Therefore, Equation 2.18 can be reduced to:

$$\frac{v^2}{gR} = f + e \quad (2.19)$$

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/s²)

R = horizontal curve radius (m)

A manipulation of Equation 2.19 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. Note that the relationships presented in Equation 2.19 are based on a plain flat 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.

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 2.8 illustrates the overturning moment that is generated as a vehicle navigates a horizontal curve. Vehicle rollover will not occur as long as the traction or

overturning moment is smaller than the resisting moment. Referring again to Figure 2.7 and neglecting the effect of body roll and other contributing factors such as wind, this can be stated mathematically as (Lamm et al. 1999):

$$\left(\left(\frac{mv^2}{R} \right) \cos \alpha - Q \sin \alpha \right) h \leq \left(\left(\frac{mv^2}{R} \right) \sin \alpha + Q \cos \alpha \right) b \quad (2.20)$$

Figure 2.8 Different forces and moments experienced by a vehicle negotiating a circular curve [Source: TAC1999]

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 2.20 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 be discussed in more detail.

Baker et al. (2001) examined the effectiveness of truck rollover warning systems. Those warning systems range from speed reduction signs, rollover warning signs,

and chevrons to warn all drivers of a potentially dangerous curve ahead. Although signing is present at most problem locations, many drivers become desensitized to a specific warning among the multitude of other roadside signs. Intelligent rollover safety systems are designed to calculate the rollover potential for the specific vehicle and direct a warning if required. The directed message is achieved by activating a sign or flashing lights only when a potential rollover vehicle is detected. Modern intelligent rollover systems can incorporate several vehicle parameters, such as speed, weight, live load, non-live load, vehicle height, and vehicle configuration, into the threshold equation, which significantly increases the effectiveness and accuracy of the rollover warning system. A case study was then carried out using three types of rollover warning systems: -

- Static signing.
- Speed-based rollover system.
- Rollover system based on speed and weight.

It was found rollover incidents are complex events that require a large number of inputs for a rollover warning system to be effective and efficient. The case study revealed that there is an added advantage to incorporating weight in addition to speed and classification when warning commercial vehicles of potential rollover. Including weight in an intelligent rollover system reduced the number of false alarms from the speed-based system by 44-49%. Thus, there is dramatic increase in effectiveness for the rollover system. In the long run, accurate system performance

will ensure that the public will continually respond to the message of the intelligent rollover warning system.

2.3.3 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 favoured 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).

$$F \leq F_{\max} = \mu_P Q \quad (2.21)$$

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 behaviour.

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.

Geometric design guides (AASHTO 2001; TAC 1999) distinguish between side friction factors for rural freeways and high-speed urban streets from those for low-speed urban streets. The research work in this thesis is based on design values for side friction factors for rural freeways. Further research is required to interpret the findings of this thesis to be adopted for urban streets or for freeway interchange ramps.

2.3.4 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.

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 manoeuvres 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 manoeuvres 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 behaviour in accordance with the prevailing conditions.

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.

2.3.5 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 inside in a curved glass tube (Figure 2.9). There is a dampening liquid in the tube yet the steel ball is free to roll in response to the forces acting on it (Merritt 1998).

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 2.10. 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 influencing the neutral position of the steel ball. The movement of the steel ball is measured in degrees and in a neutral position and the reading should be zero degrees. The ball bank indicator shown in Figure 2.9 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 70mph. 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).

In western Canada, a number of curves have been examined using various vehicles. It was 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).

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):

$$BBI = 1.115 + 52.627(ULA) \quad (2.22)$$

Where,

BBI = Ball bank indicator reading (degrees), and

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 re-examine 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. These criteria assume 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).

Figure 2.9 A typical ball bank indicator [Source: Furtado 2002]

Figure 2.10 Geometric considerations for a ball bank indicator
[Source: Merritt 1998]

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. After all, it should be noted that the current side friction factors used in North American design guides are based on research work that have been done more than six decades ago.

2.3.6 Minimum Radius for Horizontal Curves

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 can be derived to describe vehicle dynamics. Although the curve radii given by point mass formula 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).

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:

$$R_i = \frac{V^2}{121(0.5e + f)} \quad \text{for passenger cars} \quad (2.23)$$

$$R_i = \frac{V^2}{122.5(0.75e + f)} \quad \text{for trucks} \quad (2.24)$$

where,

- V = vehicle speed (km/hr)
- e = superelevation rate (% /100)
- f = side friction factor
- R_i = minimum horizontal curve radius

The minimum horizontal curve radius in equations 2.23 and 2.24 is based on a sprung vehicle. By inspection of Equations 2.23 and 2.24 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 and the approach presented by Chang (2001), curve radii values were calculated and plotted in Figure 2.11. A superelevation rate of 6 % was used and values of side friction were taken for design speeds of 20 - 130 km/hr from AASHTO (2001). As shown by Figure 2.11, 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 Equations 2.23 and 2.24, 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 2.11, Chang (2001) suggests increasing superelevation rates or implementing speed reduction methods as potential solutions to maintaining the desired margin of safety.

2.3.7 Operating Speeds on Horizontal Curves

There are three terms related to speed. These terms should be distinguished from each other, namely: design speed, posted speed, and operating speed. The selection of design speed and posted speed have many tradeoffs, that the practice of basing posted speed limits on statistical analysis of individual vehicular speeds observed at a spot on the roadway was initiated long time ago.

Figure 2.11 Minimum curve radii by using Chang and AASHTO [Source: Furtado 2002]

An assumption basic to the procedure is that motorists can decide the appropriate speed at which to travel, and the 85th percentile speed is assumed as a reasonable speed for use as the posted speed limit. Because of differences in design and operations criteria, there are locations where the posted speed limit based on an 85th percentile speed exceeds the roadway's design speed (Fitzpatrick et al 1997). This situation is a result of the fact that criteria used in highway design incorporate a significant factor of safety. Consequently, it is not surprising that motorists feel comfortable travelling at speeds greater than the roadway's design speed during good weather conditions; however, when posted speed exceeds design speed, exposure to tort liability is a concern.

Fitzpatrick et al. (1997) collected data to carry out a study to examine the relationships between design speed, operating speed, and posted speed on rural two-lane highways. The authors used three different techniques of data collection, namely: -

- Mail-out surveys distributed to each TxDOT district and state DOT, and 130 cities and counties across the United States.
- Personal interviews with two Federal Highway Administration (FHWA) engineers, six members of the American Association of State Highway and Transportation Officials (AASHTO) task force on geometric design, and 13 traffic engineers from ten states.

- Field studies conducted on suburban highways with horizontal and vertical curves with low inferred design speed.

It was found that for design speeds lower than 70 km/h, operating speeds tend to be higher than design speed, resulting in a posted speed exceeding design speeds, which raise liability concerns. 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.

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-behaviour 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 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.

2.4 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.

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 favourable.

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 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).

An inspection of the point-mass equation 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 impractical 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).

2.5 Coordination Between 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.

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).

Extensive research work has been published for the coordination of horizontal and vertical alignments. Easa (1994) quantified the improvement in sight distance by using horizontal or vertical reverse curves. Hassan and Easa (1998) developed new methodology to determine sight distance red zones on combined horizontal and vertical alignments. Other research work was published for vehicle stability on combined horizontal and vertical alignments. Furtado et al. (2002) examined stability of passenger cars on simple horizontal curves combined with vertical alignments, and it was found that an increase is required in the minimum radius of the horizontal curve if it was combined with a vertical alignment. Easa and Dabbour (2003) examined truck stability on simple horizontal curves combined with vertical alignments. Dabbour et al (2003) examined truck stability on reverse horizontal

curves combined with vertical alignments and it was found that an increase is required for minimum radius of the sharper arc of the reverse curves to compensate for both the effects of reverse curvature and vertical alignment.

2.6 Interchange Ramps

Interchange ramps are good application for the coordination between horizontal and vertical alignments. They usually have sharp horizontal curves of different types all combined with large difference in elevation, either rise or fall. Interchanges vary from single ramps connecting local streets to complex and comprehensive layouts involving two or more highways (AASHTO 2001). There are numerous combinations of interchange types. An important element of interchange design is the assembly of one or more of the basic types of ramps. The layout for any specific ramp and type of traffic movement will reflect surrounding topography and culture, cost, and degree of flexibility in desired traffic operation.

2.6.1 Design Considerations for Interchange Ramps

The design guidelines applied for ramp curves are the same applied for horizontal alignments (AASHTO 2001; TAC 1999). Compound or spiral curve transitions are desirable to: -

- Obtain the desired alignment of ramps,

- Provide for a comfortable transition between the design speeds of the through and turning roadways, and
- Fit the natural paths of vehicles.

Caution should be exercised in the use of compound curvature to prevent unexpected and abrupt speed adjustments. Sight distance along a ramp should be at least as great as the design stopping sight distance. Sight distance for passing is not needed. There should be a clear view of the entire exit terminal. The profile of a typical ramp usually consists of a central portion on an appreciable grade, coupled with terminal vertical curves and connections to the profiles of the intersection legs.

Usually, ramp profiles assume the shape of the letter “S” with a sag vertical curve at the lower end and a crest vertical curve at the upper end (AASHTO 2001). Additional vertical curves may be needed, particularly on ramps that overpass or underpass other roadways. Where a crest or sag vertical curve extends onto the ramp terminal, the length of curve should be determined by using a design speed between those on the ramp and the highway.

2.6.2 Design Speed on Interchange Ramps

Desirably, ramp design speeds should approximate the low-volume running speed on the intersecting highways (AASHTO 2001; TAC 1999). This design speed is not always practical, and lower design speeds may be selected, but they should not be

less than the low range presented in Table 2.1. Values in Table 2.1 apply to the sharpest, or controlling, ramp curve.

2.6.3 Operating Speed on Interchange Ramps

Ramp design speed criteria were evaluated (Hunter et al. 2001) by observing six ramps in three Texas cities. Ramp and freeway traffic speed-distance relationships were observed in the field using videotaping methods. Traffic operations were described in terms of ramp and freeway speeds and accelerations, as well as merging locations, accepted time-gap sizes, and freeway time headways. Performance characteristics for numerous ramp features and volume combinations were compared. It was found that ramp driver speeds are consistently greater than 50% of the freeway design speed, even though if the design ramp speed was 50% of the freeway design speed, which might have negative safety implications. This finding stresses the importance of revising the current criteria used in selecting ramp design speeds.

2.7 Different Models for Vehicle Stability

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.

Table 2.1 Guide values for ramp design speeds [Source: AASHTO 2001]

	Metric							
Highway design speed (km/h)	50	60	70	80	90	100	110	120
Ramp design speed (km/h)								
Upper range (85%)	40	50	60	70	80	90	100	110
Middle range (70%)	30	40	50	60	60	70	80	90
Lower range (50%)	20	30	40	40	50	50	60	70
Corresponding minimum radius (m)	see Exhibit 3-43							

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.

2.7.1 Point-Mass Model

The point-mass model, as was discussed earlier, 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, 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 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 manoeuvre (Kontaratos et al. 1994).

2.7.2 Bicycle Model

An extension for 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 is known as the bicycle model. There are 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 2.12 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 additional friction force that has been introduced. Assuming rear and front friction factors to be identical, simply distributing the frictional forces to a front and rear tire result in friction factors that are identical to the values resulting from the use of the point-mass formula (MacAdam et al. 1985).

2.7.3 Two-Axle Model

The two-axle model shown in Figure 2.13 "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 2.13. The elevated mass centre assumed for this model allows the weight of the vehicle to shift laterally.

Figure 2.12 Bicycle model [Source: MacAdam et al. 1985]

Figure 2.13 Two-axle model [Source: MacAdam et al. 1985]

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 2.12 and 2.13 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 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.

2.7.4 Vehicle Dynamics Model

The United States Federal Highway Administration (FHWA) is currently developing a multi-year project to apply new computer technologies towards the process of highway design. This project is called the Interactive Highway Safety Design Model

(IHSDM). It contains different software packages. One of them was VDM RoAD. This computer program was developed at University of Michigan Transportation Research Institute (UMTRI) to be included in the IHSDM module. It accurately simulates a vehicle traveling through a user-defined alignment. Estimates of the lateral acceleration can be obtained allowing the designer to evaluate geometric alternatives. More details about IHSDM project and VDM RoAD will be discussed in the following sections.

2.8 IHSDM Project

The Interactive Highway Safety Design Model (IHSDM) is a multi-year project under development by Federal Highway Administration (FHWA) to apply new computer technologies toward the process of highway design. IHSDM is composed of several modules focused on various aspects of roadway safety that gives highway planners and designers the never-before-available opportunity to quickly and easily access an innovative suite of modules to assess the potential safety effects of specific geometric design decisions.

In its current form, available information on the safety effects of highway planning and design decisions is not readily usable to evaluate and compare design alternatives. The lack of convenient evaluation tools limits a designer's ability to detect potential safety problems in design, to select safety cost-effective design parameters, and "to compare the safety of various alternatives or to optimize the

safety of a particular design." From a broader perspective, it is difficult to assess tradeoffs between safety versus social, environmental, and economic impacts. As a result, differences in safety performance may not be given due consideration in deciding among design alternatives.

IHSDM is envisioned as a computer-based tool that facilitates evaluation of the safety implications of design alternatives throughout the planning, design, and review phases of typical highway construction or reconstruction. For ease of use, it is being developed in a CAD environment, in which most design work is currently performed. The initial development efforts are restricted to two-lane rural highways. Two-lane rural highways are the largest single class of highways, representing approximately two-thirds of all federal-aid highways in the US. Because of their age, condition, and crash experience, they are common targets for improvement projects. As such, they are also a logical initial focus for IHSDM. A second phase of IHSDM development will add the capability to evaluate multilane design alternatives. IHSDM consists of five different modules:

- *Accident Analysis Module* will consist of three models: one that estimates the number and severity of crashes on specified highway segments; a second that performs safety benefit-versus-cost analyses of alternative roadside designs; and a third that uses an expert systems approach to evaluate intersection designs, identify geometric deficiencies that may impact safety, and suggest improvements that correct these deficiencies.

- *Design Consistency Module*, will evaluate the consistency of a design relative to drivers' speed expectations.
- *Driver/Vehicle Module* will consist of a Driver Performance Model linked to a Vehicle Dynamics Model, which will permit the designer to evaluate how various drivers would operate a given vehicle (e.g., passenger car or tractor-trailer) through a design and identify whether conditions exist that could result in loss of vehicle control (e.g., skidding or rollover).
- *Traffic Analysis Module* will use existing microscopic traffic simulation models to estimate likely operating conditions on a design under current and projected traffic volumes.
- *Policy Review Module* will evaluate design elements for conformance with established design policies (AASHTO, state, or local), identify elements that deviate from those policies, and aid the designer in documenting the safety implications of those deviations.

IHSDM provides an integrated collection of evaluation tools that estimate safety-related measures of effectiveness, identify areas for improvement, and compare design alternatives. A users group representing state departments of transportation and FHWA field offices provides periodic input to ensure that these tools are responsive to the needs of the user community. FHWA is also working with civil design software vendors through cooperative research and development agreements so that IHSDM can be integrated into those software packages for delivery to highway planners and designers.

2.9 Vehicle Stability Simulation Software

There are many computer programs available for vehicle stability simulation. The most competing ones are those that were competing to be included in the IHSDM project. The three computer programs that were potentially included in the IHSDM project are: NADSdyna, VDANL, and VDM RoAD. The following section will give a brief description for each of them.

2.9.1 NADSdyna

Starting in 1989, the National Highway Traffic Safety Administration (NHTSA) began researching the use and construction of a new state-of-the-art driving simulator, the National Advanced Driving Simulator, NADS, (Haug et al, 1990). A site selection competition among United States universities was conducted, and in 1992, the University of Iowa was selected. As part of the University of Iowa's cost sharing, they contributed the core vehicle dynamics software for the NADS. This software program, called the Real Time Recursive Dynamics, RTRD, (Tsai and Haug, 1989), is a minimum coordinate set, multi-body dynamics program that breaks the vehicle chassis and suspension system into a tree topology by cutting closed kinematics loops. This tree topology allows the multi-body system to be solved on parallel computers in real time.

After winning the NADS site selection competition, the University of Iowa supplemented the core vehicle dynamics (RTRD) with a collection of sub-modules that model portions of the vehicle such as the powertrain, tires, brakes, steering system, and aerodynamics that cannot be modeled using multi-body dynamics. The RTRD program containing these sub-modules is called NADSdyna (The University of Iowa, 1995).

The Vehicle Research and Test Centre (VRTC) in East Liberty, Ohio has validated NADSdyna as part of the National Advanced Driving Simulator (NADS) program (Chrstos and Crygier, 1997). The Ford Taurus GL 1994 was used for the validation. It was found that the variation in lateral acceleration was $\pm 0.05 \text{ m/s}^2$ at 95-percent confidence level.

2.9.2 VDANL

The vehicle dynamics code VDANL (Vehicle Dynamics Analysis, Non Linear) was developed in the 1980's by the National Highway Traffic Safety administration (NHTSA) and Systems Technology, Incorporated (STI). It is a comprehensive vehicle dynamics simulation program that runs on a P.C. in a windows environment. It was designed for the analysis of passenger cars, light trucks, articulated vehicles and multi-purpose vehicles and has been upgraded over the years to expand and improve its capabilities. It now permits analysis of driver induced manoeuvring up

through limit performance conditions defined by tire saturation characteristics, as well as driver feedback control features.

VDANL was chosen by the Federal Highway Administration (FHWA) for use in the Interactive Highway Safety Design Model (IHSDM). The IHSDM program is used to assess new roadway designs by using a driver performance model to simulate the vehicle/driver response when traversing the proposed roadway configuration (Paniati and True, 1996). The Driver Performance Model in IHSDM estimates drivers' speed and path choice along a roadway and this information is provided as input to VDANL, which estimates vehicle kinematics such as lateral acceleration, friction demand, and rolling moment. The information from VDANL is used to identify conditions that could result in loss of vehicle control (i.e., skidding or rollover).

2.9.3 VDM RoAD

One of the most advanced vehicle stability models is VDM RoAD (Vehicle Dynamic Models Roadway Analysis and Design). The model, referred to throughout as VDM, was developed at University of Michigan. It accurately simulates a vehicle traveling through a user-defined alignment, taking into account vehicle characteristics such as body roll, pitch, yaw, and lateral weight distribution. VDM can analyze lateral acceleration, directional control, roll stability, and stopping sight distance. Both braking performance and directional control stability are mainly related to dynamics analysis in the yaw plan, while roll stability and lateral skidding are related to the roll

plan. VDM is an extension to two commercially distributed computer programs, namely TruckSim and AutoSim. VDM accurately simulates vehicle dynamics behaviour of passenger cars, trucks, and two-unit combination vehicles on 3D road alignments. The software runs under Windows 95 and later versions, including Windows 98, Windows 2000, Windows ME, Windows NT, and Windows XP. Although of the more comprehensive and accurate details of the model, the simulations run in real-time or faster on Pentium-Pro computers (or later versions). In other words, a run simulating a 2 minute test will finish in less than 2 minutes. VDM uses mathematical models similar to those in the commercial programs TruckSim and CarSim. The models are based on research conducted over the past few decades at the University of Michigan. The TruckSim and CarSim software packages were further developed and commercialized by Mechanical Simulation Corporation (MSC), a private company in Ann Arbor, MI. MSC licenses, maintains, and supports TruckSim and CarSim.

The equations used in VDM are identical to those in the corresponding models in TruckSim and CarSim. In addition, the VDM versions include the capability of reading the IHSDM files to define the 3D alignment for the vehicle to be simulated. The VDM versions of the equations also have subtle modifications in the input requirements to make them more convenient for interacting with a road coordinate system based on station number and lateral position in addition to the conventional X, Y coordinate required for vehicle models. More details about VDM will be discussed in Chapter 3.

CHAPTER 3: SIMULATION SOFTWARE - VDM RoAD

VDM RoAD (Vehicle Dynamic Models Roadway Analysis and Design) is one of the most advanced vehicle stability models. It was developed at University of Michigan with the intention to be included within the IHSDM module. This chapter gives more details about VDM RoAD, including its capabilities and limitations and the criteria used to select it to be utilized throughout the research work. This chapter also gives description for all data required to be input to the software, including vehicle type, road geometry, human factor and cargo load distribution. This chapter also gives more details about data processing and obtaining results in VDM RoAD.

3.1 Capabilities of VDM RoAD

VDM can analyze lateral acceleration, directional control, roll stability, and stopping sight distance. Both braking performance and directional control stability are mainly related to dynamics analysis in the yaw plan, while roll stability and lateral skidding are related to the roll plan. A diagram for a four-wheeled tractor as viewed from the top (yaw plane) is shown in Figure 3.1a. There are three equations that govern vehicle behaviour in the yaw plan. Summing forces along both X and Y directions result in two equations, while the third equation can be derived by summing the moments about the vehicle mass centre. Yaw behaviour is also affected by the rotary motion of the vehicle body in roll and pitch.

The mechanical energy is transferred as the vehicle rolls and pitches, and those motions contribute to the vehicle transient response. A diagram for the vehicle body as viewed from the rear (roll plane) is shown in Figure 3.1b. The external forces acting on the vehicle are due to the tires and hitches. The vehicle is also subject to roll moments from the hitches and from the vertical component of the tire forces.

Figure 3.1 Analytical model for truck stability in VDM RoAD: (a) in yaw plane (X-Y) as viewed from the top and (b) in roll plane (Y-Z) as viewed from the rear [Source: Sayers 1999]

Vehicle stability in the roll plane may be determined by balancing the moments about the vehicle mass centre. When tires lift off the ground, the forces are zero on one side of the axle. When lift-off has occurred for more than a critical number of axles, roll stability cannot be achieved and the vehicle will potentially rollover. This condition is considered to be the onset of rollover. Thus, the basic factors for predicting stability against rollover are the vertical tire forces and their points of application relative to the vehicle mass centre, which changes as the vehicle rolls.

VDM generates a set of 3D equations to simulate vehicle stability at every time interval. These include vehicle equations (in both yaw and roll planes), hitch models, tire models, steering system models, suspension models, and braking models. Care has to be taken, however, in reaching a balance between model simplicity and model 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). The flexibility of VDM model allows for one body element or sprung mass and up to six axles or unsprung masses. The input parameters for the model include vehicle characteristics, highway geometry, lateral offset, braking, and speed control.

VDM RoAD was validated, along with VDANL and NADSdyna (Sayers 1999) by measuring actual lateral acceleration and other vehicle dynamics and comparing them with what was predicted by VDM RoAD. However, it should be noted that the

validation was mainly based on flat alignments. The author is not aware of any validation for vehicle stability on 3D alignments.

3.2 Selection Criteria for VDM RoAD

VDM was selected to be the simulation software used through this study. The selection was based on many factors, including: -

- Its ability to input any user-defined alignment, including combined horizontal and vertical alignments in 3D environment, which is the case in this study.
- VDM has a comprehensive vehicle library, including the most common vehicles on North American roads. The library includes also a real truck (3a/2a) and a real passenger car (Ford Taurus).
- VDM was used before to examine the stability of passenger cars on combined horizontal and vertical alignments (Furtado, 2002).
- The output results can be seen as an animation or as a series of charts, which gives more flexibility and easy-to-read results even for non-professionals. However, the input process is complicated and needs through understanding of the software capabilities and of Transportation Engineering in general.

3.3 Input Data to VDM RoAD

The VDM program starts with the main screen shown in Figure 3.2 with three screen areas. The input area in the left-hand side column, the simulation processing area in the middle column and the output area in the right-hand side column. The user essentially works from left to right on this screen to perform a simulation run and view the results of that run. From the main run screen, 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.

Figure 3.2 Main screen in VDM RoAD

These functions allow the user to quickly access the data libraries built into VDM to facilitate efficient modification to the simulation parameters as required. The following sections will discuss the procedures of data input in VDM.

3.3.1 Vehicle Type Input

Starting from the left in the input area in the main screen (Figure 3.2) there are many variables to be input. The first variable to be input is the detailed vehicle data, it should be input either manually or from the built-in library. Vehicle data includes steering systems, front and rear suspensions, tires, inertia properties in addition to basic weights and dimensions.

Figure 3.3 Vehicle screen in VDM RoAD for NHTSA 3a/2a combination truck.

The main data screen for NHTSA 3a/2a vehicle is shown in Figure 3.3. It should be noted that every data field shown in the figure next to a pull-down arrow is leading to another screen to input detailed data for the particular item.

As would be expected with such a comprehensive simulation package, the amount of input data needed to input a vehicle into the software is extensive. The data includes detailed vehicle characteristics, such as steering kinematics, suspension systems, nonlinear tire models and other more trivial items such as mass and wheelbase. The VDM computer program has a built-in vehicle library that contains the following truck components and combinations:

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

The properties of heavy-truck combination were measured from the field to support research at the National Highway Traffic Safety Administration (NHTSA) to validate the vehicle dynamics model (Figure 3.3). Every vehicle property includes all dynamic characteristics of the vehicle as mentioned earlier, such as brakes (and ABS), wheels, tires, axles, suspension, hitch, steering, and kinematics.

3.3.2 Road Geometry Input

Following the input and selection of vehicle data, the geometric alignment data are input to VDM in the Interactive Highway Safety Design Model (IHSDM) interchange file format. The IHSDM input file is a text file that was constructed such that each record (row) in the file is a 'critical point' along the candidate road alignment. A critical point is any location where there is a change in horizontal alignment, vertical alignment, or cross-section. The data are tabulated in 37 columns, each represents a variable that describes the roadway geometry including horizontal alignment, vertical profile, pavement width and cross-slope, shoulder width and slope, and side-slope information. Unfortunately, VDM is designed to import the IHSDM file, and does not have the capability to prepare it by itself. This means that the IHSDM file had to be pre-prepared either manually or using a simple FORTRAN computer program. The IHSDM file format has the capability to input any 3D roadway geometry, which is the main benefit of using VDM in this research.

3.3.3 Human Factor Input

The next steps in the modeling sequence involve the integration of physical or tactile decisions that involve the human factor to some extent. These decisions are mainly speed and steering controls. Both can vary as a function of roadway stationing or alternately remain constant throughout the simulation. The VDM screen for the

speed control is shown in Figure 3.4. Note that VDM has the capability to input the speed control function, as well as any other function, as a series of plot points, which gives more flexibility to the user to input virtually any function, either linear, second degree, or any other type of functions.

It should be noted that VDM was not designed to handle more complex human factors, such as braking as a reaction to increasing lateral acceleration when it exceeds driver comfort limit, or steering as a reaction to increasing roll angle where a rollover is likely to occur. Driver's workload is not included at all in VDM.

Figure 3.4 Speed control screen in VDM RoAD

Lane change manoeuvre should also be input into VDM, either as a constant lateral position, single lane change manoeuvres or irregular series of lane change manoeuvres (king-of-the-road manoeuvre). The VDM 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 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 the vehicle, 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 and vehicle characteristics.

3.3.4 Cargo-Load Distribution Input

One of the most important features in VDM is that it allows the user to input user-defined cargo load dimensions and distribution, including length, width, and height of the cargo. Although the cargo mass is determined by truck characteristics, the height and lateral shift of the mass centre are very important factors in truck stability against rollover as was found before (Garcia et al 2003). The higher and more eccentric the mass centre of the cargo is, the lower the speed for truck to rollover.

3.3.5 Side Friction Factor in VDM

An important assumption used by VDM is the pavement condition and associated maximum side friction factor. The VDM assumes a dry pavement with an average roughness. The VDM uses the maximum amount of side friction for dry pavement assumed by the design guides for freeways. The findings of this research can easily be 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 pavement 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 factors for wet pavements is that there is currently no accurate method for measuring water depths on pavements (Morrall et al. 1994).

The findings of this research can also be transferable to side-friction factors for urban streets or for freeway interchanges by simply comparing the output obtained against the appropriate side friction factor using the same design speed for both urban streets and freeways. For example, the side-friction factor for design speed 70 km/h on freeways is 0.140, while the side-friction factor for the same design speed on urban streets is 0.163. Hence, the findings of this research can properly be transferable to side-friction factors for urban streets or for freeway interchanges by knowing those two values.

3.4 Data Processing in VDM RoAD

Next in the modelling process is the simulation run event where the simulation solver programs within VDM are initiated to resolve the various differential equations that define vehicle motion and associated forces and reactions. The solver program reads in the files previously described in the input area as input for the numerical integration that used to solve the corresponding equations of motion.

3.5 Obtaining Results in VDM RoAD

The output and post-processing features in VDM allow the user to view the simulation run using a wire frame animator or alternately produce a variety of graphical plots. A sample of animator frame from a simulation run is shown in Figure 3.5, while a sample of the output plots is shown in Figure 3.6. The numerous predefined output plots include tire and steering responses, vehicle motion and tracking and select roadway geometry to name a few. The wire-frame animator, although rudimentary, does allow the user to view vehicle responses from a variety of angles. The output plots and animations all together can provide a useful alignment evaluation tool for road designers.

3.6 Limitations In VDM RoAD

Although VDM has many features and advantages, it also has some limitations. Human factors are included in VDM to some extent. VDM was not designed to handle complex human factors, such as braking as a reaction to increasing lateral acceleration when it exceeds driver comfort limit, or steering as a reaction to increasing roll angle where a rollover is likely to occur. Driver's workload is not included at all in VDM. Aerodynamic forces are not also included in truck model in VDM.

It should be also noted that VDM is mainly designed to be used by road designers, hence, options that would not be of interest to road designers have been removed. Output variables involving suspension and tire behaviour are not included in the output files. Inputs that would be used to simulate proving ground tests (e.g. steering wheel angle as a function of time) are not supported. In order to automatically set initial conditions for the moving vehicle to match the target speed and road geometry, the option to set arbitrary initial conditions is not available.

The vehicle library in VDM RoAD does not include Sports Utilities Vehicle (SUV) design vehicle. During the 1990's Sport Utility Vehicles (SUV) gained tremendous popularity. In the decade of the 1990's, the number of registered SUV in the United States more than tripled. SUV, as a class of vehicles, typically have a higher centre of gravity than passenger cars. This characteristic frequently relates to a higher

propensity to rollover when compared to passenger cars under similar circumstances. However, North American design guides do not include SUV as a design vehicle.

Another limitation in VDM RoAD is that it uses the maximum amount of side friction for dry pavement assumed by the design guides for freeways. However, the findings of this research can easily be transferable to side-friction factors for urban streets or for freeway interchanges as was explained earlier.

Figure 3.5 Wire frame animation in VDM RoAD

Figure 3.6 Sample of output plots for a single run in VDM RoAD

CHAPTER 4: DATA PREPARATION

This chapter reviews the criteria used to select all the different variables used in experimental work, including the selection of the simulation software, test alignment configurations, design vehicle, cargo-load distribution, and other considerations. Truck stability is required to be evaluated on different 3D alignments. Three cases are included, namely: -

- Truck stability on simple horizontal curves combined with vertical alignments;
- Truck stability on compound horizontal curves combined with vertical alignments. This case includes two sub-cases:
  - The evaluation of the effect of the compound curvature by its own, without being combined with any vertical curvature.
  - The evaluation of the effect of introducing vertical alignment to the compound curve.
- Truck stability on reverse horizontal curves combined with vertical alignments.

This case includes two sub-cases:

- The evaluation of the effect of the reverse curvature by its own, without being combined with any vertical curvature.
- The evaluation of the effect of introducing vertical alignment to the reverse curve.

4.1 Selecting Test Alignment Configurations

Many alignment combinations were used in the data collection process to ensure that the results represented most practical situations. VDM was used to evaluate truck stability on different types of simple, reverse and compound horizontal curves. The horizontal curves are combined with different types of vertical alignments (upgrade, downgrade, crest curve, and sag curve). For the test alignments, the input variables included maximum superelevation, horizontal deflection angle, maximum vertical grades, minimum length of vertical curves, and maximum rate-of-change of grade, for vertical curves

4.1.1 Minimum Radius and Superelevation

A series of alignments were produced in 2D as a basic control group based on Canadian minimum horizontal curve radius guidelines for $e_{max} = 0.04$ and 0.06 (TAC 1999). Superelevation runoff and tangent runout were applied for this control group and all other test alignments. Deflection angles in all cases were selected as 45° degrees. A 45-degree deflection angle resulted in a sufficiently long curve for maximum superelevation to be achieved along a portion of the curve. Under some circumstances, smaller deflection angles would have prevented reaching the design superelevation due to runoff lengths and this would have caused problems in interpreting the results.

Maximum superelevation of 0.04 and 0.06 were selected for evaluation. No higher values of maximum superelevation were evaluated, as it appears that $e_{max} = 0.06$ is gaining more acceptance because it results in better horizontal alignment in cases where minimum radii are used. It also improves operational characteristics for vehicles traveling at lower speeds during adverse weather conditions (TAC 1999).

Spiral transitions are sometimes used to introduce the vehicle to the directional change in a more gradual manner. The contribution to vehicle stability provided by spiral transitions has been found to be relatively minimal (Harwood et al. 1994). Evaluating vehicle stability on two identical alignments, one with spiral curve and the other one without spiral curve validated this previous finding. It was found that the lateral acceleration values in both cases were almost identical, which confirms the previous finding that spiral curves have minimal effect on vehicle stability on horizontal curves. Hence, spiral transitions were not used in the configuration throughout the experimental work.

4.1.2 Vertical Alignments

The effect of grades on truck speeds is much more pronounced than on speeds of passenger cars. On level grades, truck speeds are approximately the same as passenger car speeds. On downgrades, truck speeds are about 5% higher than on level terrain. On upgrades, there is a large variance in truck speeds depending on the severity and the grade length as well as the mass/power ratio of the vehicle,

which may require the use of climbing lanes in some cases. The maximum grades of +6% and -6% were selected since this grade was the controlling limit used on Canadian rural highways (TAC 1999).

The minimum lengths of vertical curves were selected as to satisfy the maximum rate-of-change of grade, r . Since vertical curves are usually parabolic, r is constant for the curve. The inverse of r is denoted by K , which is the curve length corresponding to a change in grade of 1% (TAC 1999). For crest vertical curves, K was selected to satisfy the minimum stopping sight distance, while headlight control was the limiting factor for sag curves. It was found that the most conservative results were obtained when the vertical curve length extends along the entire test segment.

4.1.3 Ratio Between Flatter and Sharper Radii in Complex Curves

A major factor that affects truck stability on complex horizontal curves is the ratio between the radius of the flatter arc and the radius of the sharper arc composing complex curves. For compound curves, that ratio was required not to exceed 1.5 (AASHTO, 2001). The reason for selecting that ratio was for the purpose of superelevation runoff. No evidence for any consideration given to vehicle stability when selecting that ratio. A series of different alignments with different ratios between the radii of the flatter and the sharper arcs were selected with ratios ranging from 1.05 to 1.50. The maximum value of that ratio was selected to be 1.50 to comply with the design guidelines for the purpose of superelevation runoff. The

minimum value of that ratio was selected to be 1.05. Note that selecting the ratio to be 1.0 is the case of simple curve.

No design guidelines were provided at all for the design of reverse curves. No evidence for any preference given when selecting the ratio between the radius of the flatter arc and the radius of the sharper arc that composing the reverse curve. For the purpose of this study, that ratio was selected to range from 1.0 and 2.0. The minimum value of that ratio was selected to be 1.0, which means that both arcs have minimum radii. The maximum value of the ratio was selected to be 2.0 because it was found that the effect of reverse curvature was not evident for ratio values higher than 2.0.

4.2 Selecting Design Vehicle

As would be expected with such a comprehensive simulation package, the amount of input data needed to input a vehicle into the software is extensive. The data includes detailed vehicle characteristics, such as steering kinematics, suspension systems, nonlinear tire models and other more trivial items such as mass and wheelbase. The VDM computer program has a built-in vehicle library that contains the following truck components and combinations:

- AASHTO design trucks (SU, 2-axle tractor & 3-axle tractor).
- AASHTO design trailers (WB-12, WB-15, WB-19 & WB-20 trailers).
- Any user-defined combination among those tractors and trailers.

- AASHTO design combinations (WB-12, WB-15, WB-19 & WB-20 combinations).
- A heavy-truck combination (3a/2a) combo.

Every vehicle property includes all dynamic characteristics of the vehicle as mentioned earlier, such as brakes (and ABS), wheels, tires, axles, suspension, hitch, steering, and kinematics. In this study, the following design vehicles were selected:

- WB-15 from the AASHTO design guide, as it represents the moderate design truck on North American roads.
- WB-20 from AASHTO design guide, as it is the largest design truck in AASHTO.
- The heavy truck combination (3a / 2a) combo, as it is a real truck on North American roads.

Although Sports Utilities Vehicle (SUV) has higher centre of gravity, which gives it potential risk for rollover, neither the vehicle library in VDM RoAD nor the design vehicle libraries in design guides include SUV as a design vehicle. Hence, it was not selected in this research. Further research may be required to examine SUV stability against rollover on 3-D alignments.

4.3 Selecting Cargo-Load Distribution

VDM allows the user to input user-defined cargo load dimensions and distribution, including length, width, and height of the cargo. Although the cargo mass is determined by truck characteristics, the height and lateral shift of the mass centre are very important factors in truck stability against rollover as was found before (Garcia et al 2003). The higher and more eccentric the mass centre of the cargo is, the lower the speed for truck to rollover. Three different rollover incidents were encountered during the experimental work for simple horizontal curves combined with vertical alignments. All the incidents were associated with higher V under eccentric cargo loading conditions. When the same vehicles were re-examined on the same alignment with the same V with no load eccentricity, no rollover occurred. This stresses the importance of cargo size and load distribution in truck stability against rollover.

Selecting eccentric cargo-load distribution for compound or reverse curves resulted in unrealistic minimum curve requirements. Hence, uniform cargo-load distribution was selected for compound and reverse curves. That was based on the assumption that cargo with eccentric load distribution are usually transported by professional truck drivers who usually adjust their speeds according to their perception of the rolling angle which increase with the eccentric cargo-load distribution. However, eccentric cargo-load distribution may be used as a design control for further

research for some roads leading to mines. Those roads may have a design vehicle to be a truck transporting mining rocks or heavy equipment.

4.4 Other Considerations

Human factors are included in VDM to some extent. This is represented by an input variable for lane change manoeuvre: single lane change manoeuvres or irregular series of lane change manoeuvres (king-of-the-road manoeuvre). Both cases were examined along with the traditional case of constant lateral offset. It was found that the most conservative results corresponded to the irregular series of lane change manoeuvres, where the vehicle inertia resulted in more lateral shift of the vehicle. VDM was not designed to handle more complex human factors, such as braking as a reaction to increasing lateral acceleration when it exceeds driver comfort limit, or steering as a reaction to increasing roll angle where a rollover is likely to occur. It should be noted that the aerodynamic forces are not considered in VDM's truck models.

CHAPTER 5: MODEL DEVELOPEMENT - SIMPLE CURVES

5.1 Simulation Procedures

VDM was used to evaluate truck stability on different simple horizontal curves combined with vertical alignments. To generate the results, the following procedures were followed for each design vehicle (Figure 5.1):

- Input a flat alignment (without being superimposed by vertical alignment) as a basis for comparison. This flat alignment includes simple horizontal curve, superelevation, and tangent runout.
- Run the simulation using design speed, V , for that horizontal alignment.
- Read the results (base scenario): rollover, if any, and lateral acceleration.
- Superimpose the same alignment by a specific vertical alignment (Figure 5.2).
- Re-read the results (Scenario 1): rollover, if any, and lateral acceleration.
- Compare the results together.
- Increase the radius of horizontal curve, R , and re-read the results (Scenario 2). Continue to increase the radius until two conditions are satisfied: (a) the lateral acceleration obtained from scenario 2 matches the lateral acceleration obtained from base scenario and (b) the rollover, if any, is avoided.

Figure 5.1 Evaluation procedures for vehicle stability on simple horizontal curves combined with vertical alignments.

Figure 5.2 Different study cases for simple horizontal curves

- Record the radius from Scenario 2, and compare it with the radius from Scenario 1.
- Repeat previous steps for different values of R and V .
- Repeat previous steps for a different superelevation.

5.2 Required Increase in Existing Design Guidelines

Following the procedures shown earlier, the results for the required minimum radii and the required increase in design minimum radii for different design vehicles are shown in Tables 5.1 and 5.2 for $e_{max} = 0.04$ and 0.06 , respectively. The values

shown are the maximum radii values required based on all study configurations of vertical alignments (flat grade, upgrade, downgrade, crest curve, and sag curve). It was found that the most critical cases occur when the horizontal alignment is superimposed by either a sag curve or a downgrade where both the vehicle weight and the traction forces are in the same direction, resulting in maximum variation in the load distribution on different tires of the vehicle.

The tables show that the TAC and AASHTO minimum radius guidelines need to be increased. The increase ranges from 5.3% to 19.4%, depending on the design vehicle, design speed, and maximum superelevation. Figure 5.3 and Figure 5.4 show the required increases in radius for trucks and passenger cars for $e_{max} = 0.04$ and $e_{max} = 0.06$, respectively. The results for passenger cars (no data were available for $V = 110$ km/h) are based on previous study dedicated for passenger cars (Furtado et al. 2002). It is noted that the required increases in horizontal curve radii for trucks are generally larger than that for passenger cars. As mentioned earlier, this is expected in light of the increased friction required for trucks due to sprung suspension and force distribution on various tires. Note also that while rollover incidents were encountered for trucks, no rollover incident was found for passenger cars. The truck rollover may be caused by the truck suspension characteristics and the higher mass centres. Two cases of rollover were encountered for trucks negotiating horizontal curves with the minimum radii required by the design guides for $e_{max} = 0.04$. WB-20 rolled over at $V = 100$ km/h on a horizontal curve with $R_{min} = 490$ m and at $V = 110$ km/h on a horizontal curve with $R_{min} = 680$ m. In both cases,

the horizontal curve was combined with a sag vertical curve with -6% downgrade, 6% upgrade, and $e_{max} = 0.04$ m/m.

A third rollover incident was encountered for WB-20 on a horizontal curve with the minimum radius required by design guides for $e_{max} = 0.06$ at $V = 110$ km/h and a minimum horizontal radius of 600 m, and the horizontal alignment was also superimposed by a sag vertical curve with -6% downgrade and 6% upgrade. Those rollover incidents occurred under eccentric cargo loading conditions. Furthermore, the radii were increased from 490 m to 562 m, from 680 m to 805 m, and from 600 m to 710 m for all cases, respectively, where the same level of lateral acceleration was retrieved as if there were no vertical alignment. However, the rollover was still encountered. The minimum radii to avoid rollover were found to be 574 m, 812 m, and 715 m, respectively (without any safety margin) as shown in Tables 2 and 3. Although no skidding incidents were encountered during the simulation, a large increase in side friction demand is likely to reduce the skidding threshold for trucks negotiating horizontal curves imposed by vertical alignments. Note that $V = 120$ Km/h and 130 Km/h were not investigated in this study, as they are not realistic for truck operations.

5.3 Sensitivity Analysis

The design radii shown in Table 5.1 are sensitive to both side friction and superelevation. As there is high correlation between side friction and superelevation,

so that the following sensitivity analysis is conducted to evaluate the sensitivity of the required design radii against different variables.

5.3.1 Side Friction Variation

The assumed and actual values of side friction demand for WB-20 and passenger car are shown in Figure 5.5a and Figure 5.5b, respectively. The rectilinear line represents the side friction obtained through manipulation of the PM model based on driver comfort for specific R and V (assumed by the design guides). The scattered line represents the resulting lateral acceleration obtained based on the 3D vehicle dynamics model within VDM for the same R and V . The curve radius, superelevation, and design speed are used by VDM to calculate lateral acceleration based on the PM model, while roadway geometry and differential equations of motion are used to estimate the actual lateral acceleration experienced by the vehicle.

For WB-20, which consists of a tractor and a trailer, the actual lateral acceleration deviates from the assumed lateral acceleration for both tractor and trailer. The deviation for the trailer is generally greater than that for the tractor, which is expected due to the mechanical differences between them. The traction forces are transferred from the tractor to the trailer through the hitch, which can be thought of as a ballpoint joining the trailer to the tractor in a multi-body tree topology. Thus, the traction forces are applied on the trailer through a single point, which is the hitch,

while the traction forces on the tractor are distributed on all different wheels. This difference in mechanical characteristics may explain the increase in lateral acceleration on the trailer compared with the tractor.

The actual lateral acceleration experienced by WB-20 is generally greater than that experienced by the passenger car. There is also a difference in the shape of actual lateral acceleration. A passenger car experiences a small increase in lateral acceleration for a short period only (approximately 30% of the curve length), and then the deviation decreases or diminishes for the rest of the curve length. Then the deviation starts again at the start of the tangent and extends along the tangent for a length ranging from 30 m to 120 m (depending on R , V , and runoff length).

On the other hand, WB-20 experiences a larger increase in lateral acceleration for a longer period (approximately 70% of the curve length), and then the deviation slightly decreases, but does not diminish, for the rest of the curve length. Then the deviation increases again at the start of the tangent and extends along the tangent for a larger length ranging from 80 m to 260 m. The differences between WB-20 and passenger car may be explained in light of the increase in the side friction required for trucks. This increase is caused by the sprung suspension and force distribution on various truck tires, which widely vary from tire to tire (FHWA 1985). This variation in load distribution on different truck tires increases with the increase in downgrade.

5.3.2 Effect of Superelevation

A comparison between the required increase in minimum radius for different rates of maximum superelevation is shown in Figure 5.6, Figure 5.7 and Figure 5.8 for design vehicles (WB-15), (WB-20) and (3a/2a), respectively. It was found that the required increase in radius for $e_{max} = 0.06$ was generally more than that required for $e_{max} = 0.04$, especially for lower speeds. This may be explained in light of the conservative rates of side friction used in the design guides, especially for low speeds. Those conservative rates may result in relatively unrealistic design radii associated with higher superelevation. For example, for $e_{max} = 0.04$ and $V = 60$ km/h, the assumed f_{max} is 0.15 (TAC 1999), and $R_{min} = 150$ m. On the other hand, for $e_{max} = 0.06$ and the same design speed, $R_{min} = 130$ m (86.7% of R_{min} required for $e_{max} = 0.04$). If the actual side friction supply (maintaining the same margin of safety) was more than f_{max} (say 0.18), then R_{min} will be 130 m and 120 m for $e_{max} = 0.04$ and 0.06, respectively, for the same $V = 60$ km/h. In this case, R_{min} required for $e_{max} = 0.06$ is 92.3% of that required for $e_{max} = 0.04$. Note also that the variation in load distribution on different tires increases with the increase in superelevation, where the load increases on the inner tires and decreases on the outer tires. The variation in load distribution results in more variation in the side friction demand at various tires of trucks.

5.4 Model Calibration

The values in Tables 5.1 and 5.2 represent conservative values based on all test alignments and configurations, where sag curves and downgrades generally controlled in most cases. Thus, these values will be more conservative for upgrades and crest curves. To determine the minimum radius requirements based on specific vertical alignment configurations, which are not explicitly included in Tables 5.1 and 5.2, two mathematical models for trucks and passenger cars were developed and may be used for specific alignment configurations.

The simulation results obtained from VDM were used to establish the mathematical models for the design radius requirements for horizontal curves on 3D alignments. The required minimum radius (dependent variable) was related to design speed and alignment data (independent variables). Two different models were developed using regression analysis: one for trucks based on the results presented in the previous section and the other for passenger cars based on the results of previous study (Furtado et al. 2002). Note that attempts to calibrate a single model for both types of vehicles resulted in unsatisfactory model. The developed models for passenger cars and trucks were verified using three additional runs of VDM for each model using input data different from those used in model calibration. Many combinations of independent variables were examined and the final models were selected based on the following criteria:

- The coefficient of determination, R^2 , must be significant at the 0.95 confidence level.
- Each of the independent variables used in the model must have a coefficient that is significantly different from zero at the 0.95 confidence level.
- The algebraic signs of the coefficients of the independent variables must have a logical explanation.

5.4.1 Trucks Model

A total of 180 observations were used to calibrate a mathematical model for trucks.

The final model is given by

$$R_{\min} = 9.872 + 0.155(V - 50)^2 + \frac{6.300}{e_{\max}} - 294.097g_1 - 17.708g_2, \quad [R^2 = 0.959] \quad (5.1)$$

where g_1 and g_2 are the first and second grades of vertical alignment (in decimal), respectively, positive for upgrade and negative for downgrade. For a flat horizontal alignment (not superimposed by vertical alignment), both g_1 and g_2 have a value of zero. For an upgrade, both g_1 and g_2 have the same positive value, and for a downgrade they have the same negative value. For crest curves, g_1 has a positive value and g_2 has a negative value. For sag curves, g_1 has a negative value and g_2 has a positive value. The relevant statistics are shown in Table 5.3.

This model has logical explanation for the effect of each independent variable on R_{min} . The positive sign for the coefficient of V means that R_{min} increases with the increase in V . The positive sign for the coefficient of e_{max} means that R_{min} decreases with the increase of e_{max} , as expected. The negative signs for the coefficient of g_1 and g_2 means that R_{min} increases when one or both of g_1 and g_2 is a downgrade. In this case, both vehicle weight and traction forces are in the same direction, resulting in maximum variation in the load distribution on different tires of the vehicle.

Noted that g_2 is not as significant as g_1 , which is expected since the largest increase in the actual lateral acceleration occurs in the first segment of the horizontal curve while the vehicle is still negotiating the first grade of the vertical alignment (Figure 5.5b). The calibration results show that the vehicle type was not highly correlated with R_{min} and therefore was not included in the model.

5.4.2 Passenger Cars Model

The results for the minimum radius required for different alignments for passenger cars (Furtado 2002) were used to establish this model. A total of 55 observations were used in the calibration, resulting the following model:

$$R_{min} = 65.673 + 0.144(V - 50)^2 + \frac{3.835}{e_{max}} - 65.875g_2, \quad [R^2 = 0.992] \quad (5.2)$$

The relevant statistics are shown in Table 5.3. The coefficients of V and e_{max} are significantly different from zero at the 95% confidence level. Note that g_1 was originally included in the model, but the results showed that it was not highly correlated with R_{min} .

Figure 5.9 shows a comparison of the minimum radius requirements by TAC and those obtained from the mathematical models for trucks and passenger cars (Equations 5.1 and 5.2). The vertical alignment is assumed to be a downgrade ($g_1 = -6\%$ and $g_2 = -6\%$) As noted, the minimum radius requirements of the mathematical models were found to be greater than those of the current design guides, which is expected due to the 3D nature included in the models. Also, the results for trucks are generally higher than those for passenger cars due to the variation in the load distribution on different truck tires, as discussed earlier.

Table 5.1 Required 3D minimum radius for different design vehicles ($e_{max} = 0.04$)

Design Speed (km/hr)	Design Minimum Radius (m)	3D Required Minimum Radius (m)^a	Required Increase in Radius (%)	3D Design Radius^c (m)
<i>(a) WB-15</i>				
60	150	161	7.3	170
70	200	219	9.5	220
80	280	311	11.1	320
90	380	428	12.6	430
100	490	568	15.9	570
110	680	791	16.3	800
<i>(b) WB-20</i>				
60	150	166	10.7	170
70	200	224	12.0	230
80	280	323	15.4	330
90	380	444	16.8	450
100	490	562 (574) ^b	17.1	580
110	680	805 (812) ^b	19.4	820
<i>(c) NHTSA 3a/2a</i>				
60	150	162	8.0	170
70	200	221	10.5	230
80	280	319	13.9	320
90	380	434	14.2	440
100	490	572	16.7	580
110	680	808	18.8	810

^a To maintain the same level of lateral acceleration.

^b To avoid rollover

^c Rounded Values

Table 5.2 Required 3D minimum radius for different design vehicles ($e_{max} = 0.06$)

Design Speed (km/hr)	Design Minimum Radius (m)	3D Required Minimum Radius (m)^a	Required Increase in Radius (%)	3D Design Radius^c (m)
(a) WB-15				
60	130	140	7.7	140
70	190	209	10.0	210
80	250	279	11.6	280
90	340	384	12.9	390
100	440	511	16.1	520
110	600	703	17.2	710
(b) WB-20				
60	130	147	13.1	150
70	190	218	14.7	220
80	250	288	15.2	290
90	340	394	15.9	400
100	440	522	18.6	530
110	600	710 (715) ^b	19.2	720
(c) NHTSA 3a/2a				
60	130	144	10.8	150
70	190	211	11.1	220
80	250	284	13.6	290
90	340	390	14.7	390
100	440	514	16.8	520
110	600	706	17.7	710

^a To maintain the same level of lateral acceleration.

^b To avoid rollover

^c Rounded Values

Table 5.3 Summary statistics of truck and passenger car models

<i>Independent Variable</i>	<i>Coefficient</i>	<i>t-Stat</i>
(a) Truck Model		
Intercept	9.872	0.175
V^2	0.155	35.585
$(e_{max})^{-1}$	6.300	4.935
g_1	-294.097	-2.967
g_2	-17.708	-0.179
Coefficient of determination, $R^2 = 0.959$		
Significance of F statistic = 0.000		
(b) Passenger Car Model		
Intercept	65.673	5.398
V^2	0.144	78.064
e_{max}^{-1}	3.835	7.880
g_2	-65.875	-1.781
Coefficient of determination, $R^2 = 0.991$		
Significance of F statistic = 0.000		

Figure 5.3 Required increases in minimum curve radii for different design vehicles ($e_{max} = 0.04$)

Figure 5.4 Required increases in minimum curve radii for different design vehicles ($e_{max} = 0.06$)

Figure 5.5 Lateral acceleration calculated by VDM RoAD on 3-D alignment: (a) WB-20 and (b) passenger car

Figure 5.6 Required increase in design minimum radii for different rates of maximum superelevations (WB-15)

Figure 5.7 Required increase in design minimum radii for different rates of maximum superelevations (WB-20)

Figure 5.8 Required increase in design minimum radii for different rates of maximum superelevations ($3a/2a$)

Figure 5.9 Required minimum radius using different models ($e_{max} = 0.04$, $g_1 = -0.06$, and $g_2 = -0.06$)

CHAPTER 6: MODEL DEVELOPEMENT - REVERSE CURVES

There are mainly two types of complex horizontal curves, namely: -

- Horizontal reverse curves; and
- Horizontal compound curves.

This chapter will give more details about vehicle stability on reverse curves while Chapter 7 will cover vehicle stability on compound curves.

6.1 Background

While current design guides provide brief and vague guidelines for the design of compound horizontal curves to have a ratio between the radii of the flatter and shaper arcs not to exceed 1.5, there are no guidelines for the design of reverse curves from the perspective of vehicle stability. Horizontal compound or reverse curves combined with vertical alignments are commonly used in different highway classifications. Current design guides deal with vehicle stability on compound or reverse curves as two or more individual curves, each with its own design guidelines. Therefore, consistency is lacking in both the individual design of these curves and their 3D nature when combined with vertical alignment. No research work has been published on truck stability on compound or reverse curves. In order to proceed with the experimental work, the variables selected in Chapter 4 were

used throughout the experimental work for all study cases. This chapter will describe the procedures followed to evaluate vehicle stability on reverse horizontal curves combined with vertical alignments. Chapter 7 will describe the procedures followed to evaluate vehicle stability on compound horizontal curves combined with vertical alignments

In using VDM to evaluate vehicle stability on reverse curves, two different schemes were followed as shown in Figure 6.1. The first scheme was to investigate the effect of reverse curvature by itself, and the second scheme was to investigate the effect of being superimposed by vertical alignment. A description of each scheme is illustrated herein after to examine the effect of both schemes, either individually or combined together.

6.2 Effect of Reverse Curvature

In order to investigate the effect of reverse curvature, the following procedures were followed for every design vehicle (WB-15, WB-20 and 3a/2a combination) as shown in Figure 6.2:

- Input a flat alignment (horizontal curve with radius R_1) as a basis for comparison. This flat alignment includes simple horizontal curve and superelevation.
- Run simulation using design speed (V) for that horizontal alignment.
- Read the results (base scenario): rollover (if any) and lateral acceleration.

- Introduce another horizontal curve in the opposite direction (with radius R_2) to form a reverse curve consisting of two horizontal curves in opposite directions with two different radii (R_1 & R_2).
- Read the results (scenario 1): rollover (if any) and lateral acceleration.
- Increase the horizontal curve radius (R_1) and re-read the results (scenario 2). Continue to increase the radius until the results obtained from (scenario 2) match the results obtained from base scenario.
- Record the radius R_1 from scenario 2, and compare it with the radius from scenario 1.
- Repeat previous steps with different radii of the opposite curve (R_2) keeping R_1 the same, so that the effect of the radius of the opposite curve on vehicle stability can be measured and quantified.
- Repeat previous steps for different values of R_1 and V .
- Repeat previous steps for a different superelevation.

Following the procedures shown, the required increase in sharper arc radii (R_1) corresponding to different values of opposite flatter arc radii (R_2) are shown in Tables 6.1 and Table 6.2 for $e_{max} = 0.04$ and $e_{max} = 0.06$, respectively. The required increase is also shown graphically in Figures 6.3 through 6.8 for different types of design vehicles and different rates of superelevation. The tables and figures show that reverse curvature, by itself, has significant impact on truck stability.

An increase in horizontal curve radius is needed to compensate for the effect of reverse curvature. That increase ranges from 1.7% to 24.8%, depending on the design vehicle, superelevation and the radius of the second arc in the opposite direction (R_2). More increase is generally required in sharper arc radius (R_1) where the opposite arc radius (R_2) is smaller. As shown in the tables, the required increase in radius R_1 also depends on superelevation rate. The increase for $e_{max} = 0.06$ was found to be generally greater than that required for $e_{max} = 0.04$. This may be explained in light of the conservative rates of side friction used in the design guides as was discussed earlier in Chapter 5. It was also found that more increase is required for curves with low design speeds, which is a particular concern for freeway ramps, where horizontal reverse curves are usually used to minimize land use. Those ramps are usually having low design speed. It should be noted that the increase required in this stage is to compensate for the effect of the reverse curvature only.

6.3 Effect of Vertical Alignment

In order to quantify the effect of vertical alignment, the reverse curves used in scenario 2 were superimposed by different types of vertical alignments (i.e. upgrade, downgrade, crest curve, and sag curve) as shown in scenario 3 – Figure 6.1. The procedures followed were as follows:

- Superimpose the reverse alignment (obtained from scenario 2) by a specific vertical alignment (Figure 6.1).

- Re-read the results: rollover and lateral acceleration.
- Increase the horizontal curve radius (R_1) and re-read the results (scenario 3). Continue to increase the radius until the results (obtained from scenario 3) matches the results obtained from (scenario 2).
- Record the radius from scenario 3, and compare it with the radius from scenario 2.

Following the procedures shown earlier, the results for the required minimum radii and the required increase in curve radius R_1 are shown in Table 6.3 and Table 6.4 for $e_{max} = 0.04$ and $e_{max} = 0.06$, respectively. The results are also shown graphically in Figures 6.9 through 6.14. The values shown are the maximum radii values required based on all study configurations of vertical alignments (i.e. flat grade, upgrade, downgrade, crest curve, and sag curve). It was found that the most critical cases occur when the horizontal alignment is superimposed by either a sag curve or a downgrade where both the vehicle weight and the traction forces are in the same direction, resulting in maximum variation in the load distribution on different tires of the vehicle.

The results show that another increase in the minimum radius is required to compensate for the effect of being superimposed by a vertical alignment. The overall increase in the minimum radius R_1 ranges from 3.3% to 27.3% as shown in Tables 6.3 and 6.4. It should be noted that the increase shown in Tables 6.3 and 6.4 is the overall increase to compensate for both the effect of reverse curvature and the effect

of being superimposed by a vertical alignment. Comparing Tables 6.1 and 6.3, and Tables 6.2 and 6.4, it is noticed that the effect of reverse curvature overpowers the effect of vertical alignment, especially for lower speeds. This finding magnifies the impact of reverse curvature and the need to establish design guidelines for reverse curves.

6.4 Rollover Speeds

As truck rollover may be caused by suspension characteristics and the higher mass centres, so that rollover for trucks is more likely to occur prior to skidding (Harwood et al. 1990). The rollover speeds are important measure of safety and should be considered as a design control for trucks. The rollover speeds and margin of safety for different design vehicles (WB-15, WB-20 and 3a/2a) negotiating reverse curves with R_2 / R_1 ratio equals 100%, 150% and 200% are shown in Tables 6.5, 6.6 and 6.7, respectively. The critical speeds for (WB-20) design vehicle is also shown graphically in Figure 6.15 for $e=0.06$. It was found that for lower design speeds, rollover speeds are closer to design speeds with smaller margins of safety, even without being superimposed by a vertical alignment. For example, for a truck negotiating a reverse curve with design speed 40 km/h and superelevation rate 6%, the rollover speed is 50 km/h with a margin of safety 10 km/h. If the same reverse curve was superimposed by a vertical alignment, the rollover speed decreases to 46 km/h with a margin of safety only 6 km/h. This is a particular concern for freeway ramps, where horizontal reverse curves superimposed by vertical alignments are

commonly used. Noting that many ramps have unrealistically low design speeds in comparison with the design speed of the mainline roadway, safety problems may be encountered due to driver behaviour to overdrive ramps if their design speeds were not appropriately selected. It should be noted that the values shown in Tables 6.5, 6.6 and 6.7 are for eccentric load distribution condition, which could be a design control for some situations like trucks transporting mining rocks or heavy equipment.

6.5 Sensitivity Analysis

The values shown in Tables 6.1 through 6.4 represent conservative values based on all test alignments and configurations, where sag curves and downgrades generally controlled in most cases with eccentric cargo load distribution. Thus, these values will be more conservative for upgrades and crest curves. To determine the minimum radius required based on specific alignment configuration, which are not explicitly included in Tables 6.1 through 6.4, a mathematical model should be developed to be used for specific alignment configurations. In order to establish such proper mathematical model with logical explanation for all the variables, some sensitivity analysis should be conducted to determine the sensitivity of such proposed model towards different variables.

6.5.1 Effect of Reverse Curvature vs. Vertical Alignment

Comparing Table 6.1 with Table 6.3 and Table 6.2 with table 6.4, it is noticed that the effect of reverse curvature overpowers the effect of vertical alignment on the required increase in minimum radius. A graphical representation for this finding is shown in Figure 6.16 for vehicle type (WB-15), rate of superelevation 4% and $R_2 = 100\% R_1$. The effect of reverse curvature is approximately as twice as the effect of the vertical alignment. This finding magnifies the effect of reverse curvature on vehicle stability and the importance of establishing design guidelines for reverse curves.

6.5.2 Effect of Superelevation

The required increase in minimum radius for $e_{max} = 0.06$ was found to be generally more than that required for $e_{max} = 0.04$, especially for lower speeds. This finding is true either for the case of reverse curvature by itself, or the case of reverse curvature combined with vertical alignment. This may be explained in light of conservative rates of side friction used in the design guides as was explained earlier in Chapter 5. The effect of superelevation on vehicle stability is shown in Figures 6.17 through 6.19 for design vehicles WB-15, WB-20 and 3a/2a, respectively and $R_2 = 100\% R_1$. As shown in the figures, the effect of superelevation is not as significant as other effects. However, special care should be taken for superelevation runoffs in reverse curves due to the complexity associated with them.

6.5.3 Effect of Design Vehicle

The required in minimum curve radii for different design vehicles are shown in Figures 6.20 and 6.21 for superelevation rate 0.04 and 0.06, respectively. It was found that the increase required for WB-20 is generally greater than what is required for WB-15. As was mentioned before, WB-20 is the largest design vehicle in AASHTO tables, so that it is expected to have the largest variation in tire forces and reactions between front and rear tires, as well as between inside and outside tires. The required increase for NHTSA 3a/2a truck is generally lying between that for WB-20 (the largest design truck in AASHTO) and WB-15 (the moderate design truck running on North American roads). It should be noted that the properties of NHTSA 3a/2a heavy-truck combination were measured from the field, so that it is more appropriate to be used as a design vehicle.

6.5.4 Effect of The Ratio Between Sharper and Flatter Arcs

The effect of the ratio between sharper and flatter arcs on required increase in minimum curve radii is shown in Figures 6.3 through 6.14. This effect is clearly the most paramount effect affecting vehicle stability on reverse curves. The smaller the ratio, the more increase required for minimum radius. That effect almost disappears for ratio greater than 200%, so that this maximum ratio was taken as an upper limit throughout experimental work.

6.6 Model Calibration

The values in Tables 6.1 through 6.4 represent conservative values based on all test alignments and configurations, where sag curves and downgrades generally controlled in most cases. Thus, these values will be more conservative for upgrades and crest curves. To determine the minimum radius requirements based on specific vertical alignment configurations, which are not explicitly included in Tables 6.1 through 6.4, a mathematical model was developed and may be used for specific alignment configurations. The model was calibrated using the simulation results obtained from VDM. The required minimum radius (dependent variable) was related to design speed and alignment data (independent variables). Many combinations of independent variables were examined and the final model was selected based on the following criteria:

- The coefficient of determination, R^2 , must be significant at the 0.95 confidence level.
- Each of the independent variables used in the model must have a coefficient that is significantly different from zero at the 0.95 confidence level.
- The algebraic signs of the coefficients of the independent variables must have a logical explanation.

A total of 540 observations were used to calibrate the mathematical model. The final model is given by

$$R_{min} = -11.09 + 0.107(V - 30)^2 + \frac{3.693}{e_{max}} - 93.23g_1 - 13.333r, \quad [R^2 = 0.995] \quad (6.1)$$

where R_{min} = the required minimum radius for the first arc of a reverse curve (m), V = design speed (km/h), e_{max} = maximum rate of superelevation, r = ratio between the sharper and flatter arcs in the reverse curve and g_1 is the first grade of vertical alignment (in decimal), positive for upgrade and negative for downgrade. For a flat horizontal alignment (not superimposed by vertical alignment), g_1 has a value of zero. For an upgrade or crest curve, g_1 has positive value, and for a downgrade or sag curve it has negative value. The relevant statistics are shown in Table 6.8. The coefficients of all independent variables are significantly different from zero at the 95% confidence level.

This model has logical explanation for the effect of each independent variable on R_{min} . The positive sign for the coefficient of V means that R_{min} increases with the increase in V . The positive sign for the coefficient of $1/e_{max}$ means that R_{min} decreases with the increase of e_{max} , as expected. The negative signs for the coefficient of g_1 means that R_{min} increases when g_1 is a downgrade. In this case, both vehicle weight and traction forces are in the same direction, resulting in maximum variation in the load distribution on different tires of the vehicle. The negative sign for the reverse curve ratio means that R_{min} increases with the decrease of that ratio, which is also expected.

It should be noted that g_2 was not included in the model as it was found to have insignificant effect on the model, which is expected since the largest increase in the actual lateral acceleration occurs in the first segment of the horizontal curve while the vehicle is still negotiating the first grade of the vertical alignment. This model was calibrated using the results for 3a/2a design vehicle as that vehicle represents a majority of trucks really operated on North American roads. Since the properties of this heavy-truck combination were measured from the field, so that it is more appropriate to be used as a design vehicle as was discussed before.

Figure 6.22 shows a comparison between the minimum radii required for reverse curves using both TAC and the mathematical model. The design vehicle is NHTSA 3a/2a and the reverse curve ratio is 100%. As it is shown in the figure, the required minimum radii calculated using the mathematical model are generally higher than those calculated using the current design guidelines (TAC). It is also noted that the required minimum radius for a downgrade (i.e. $g_1 = -0.06$) is higher than that for a crest curve (i.e. $g_1 = +0.06$).

Table 6.1 Required increase in curve radius R_1 corresponding to different values of opposite curve radii R_2 for different design vehicles ($e_{max} = 0.04$)

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 100% R1		R2 = 150% R1		R2 = 200% R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	60	64	6.7%	62	3.3%	61	1.7%
50	100	108	8.0%	105	5.0%	102	2.0%
60	150	166	10.7%	159	6.0%	154	2.7%
70	200	223	11.5%	215	7.5%	209	4.5%
80	280	314	12.1%	305	8.9%	294	5.0%
90	380	436	14.7%	416	9.5%	402	5.8%
100	490	572	16.7%	544	11.0%	526	7.3%
110	680	809	19.0%	762	12.1%	745	9.6%
(b) WB-20							
40	60	70	16.7%	68	13.3%	65	8.3%
50	100	116	16.0%	111	11.0%	109	9.0%
60	150	177	18.0%	172	14.7%	168	12.0%
70	200	238	19.0%	232	16.0%	226	13.0%
80	280	336	20.0%	328	17.1%	318	13.6%
90	380	462	21.6%	458	20.5%	442	16.3%
100	490	598	22.0%	588	20.0%	574	17.1%
110	680	848	24.7%	826	21.5%	814	19.7%
(c) NHTSA 3a / 2a							
40	60	67	11.7%	64	6.7%	63	5.0%
50	100	112	12.0%	108	8.0%	107	7.0%
60	150	170	13.3%	164	9.3%	162	8.0%
70	200	228	14.0%	223	11.5%	217	8.5%
80	280	324	15.7%	318	13.6%	312	11.4%
90	380	446	17.4%	437	15.0%	431	13.4%
100	490	581	18.6%	571	16.5%	566	15.5%
110	680	822	20.9%	794	16.8%	792	16.5%

Table 6.2 Required increase in curve radius R_1 corresponding to different values of opposite curve radii R_2 for different design vehicles ($e_{max} = 0.06$)

Design Speed (km/hr)	Design Minimum Radius (m)	R2 =100% R1		R2 = 150% R1		R2 = 200% R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	55	59	7.3%	58	5.5%	56	1.8%
50	90	98	8.9%	96	6.7%	94	4.4%
60	130	144	10.8%	140	7.7%	136	4.6%
70	190	214	12.6%	208	9.5%	203	6.8%
80	250	284	13.6%	276	10.4%	269	7.6%
90	340	392	15.3%	384	12.9%	371	9.1%
100	440	522	18.6%	511	16.1%	494	12.3%
110	600	724	20.7%	712	18.7%	691	15.2%
(b) WB-20							
40	55	64	16.4%	63	14.5%	60	9.1%
50	90	106	17.8%	104	15.6%	99	10.0%
60	130	154	18.5%	152	16.9%	148	13.8%
70	190	230	21.1%	224	17.9%	218	14.7%
80	250	304	21.6%	296	18.4%	289	15.6%
90	340	419	23.2%	406	19.4%	396	16.5%
100	440	544	23.6%	529	20.2%	517	17.5%
110	600	749	24.8%	736	22.7%	722	20.3%
(c) NHTSA 3a / 2a							
40	55	61	10.9%	61	10.9%	58	5.5%
50	90	104	15.6%	101	12.2%	97	7.8%
60	130	151	16.2%	147	13.1%	141	8.5%
70	190	223	17.4%	217	14.2%	209	10.0%
80	250	294	17.6%	288	15.2%	278	11.2%
90	340	408	20.0%	397	16.8%	388	14.1%
100	440	533	21.1%	519	18.0%	507	15.2%
110	600	731	21.8%	726	21.0%	702	17.0%

Table 6.3 Required increase in curve radius R_1 for reverse curves combined with vertical alignment (superelevation rate is 4%)

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 100% R1		R2 = 150% R1		R2 = 200% R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	60	66	10.0%	64	6.7%	62	3.3%
50	100	115	15.0%	111	11.0%	109	9.0%
60	150	166	10.7%	162	8.0%	158	5.3%
70	200	227	13.5%	218	9.0%	212	6.0%
80	280	319	13.9%	316	12.9%	307	9.6%
90	380	448	17.9%	438	15.3%	426	12.1%
100	490	582	18.8%	569	16.1%	554	13.1%
110	680	824	21.2%	794	16.8%	772	13.5%
(b) WB-20							
40	60	72	20.0%	68	13.3%	66	10.0%
50	100	126	26.0%	117	17.0%	112	12.0%
60	150	179	19.3%	174	16.0%	171	14.0%
70	200	239	19.5%	234	17.0%	230	15.0%
80	280	342	22.1%	336	20.0%	331	18.2%
90	380	471	23.9%	467	22.9%	455	19.7%
100	490	611	24.7%	597	21.8%	584	19.2%
110	680	856	25.9%	844	24.1%	836	22.9%
(c) NHTSA 3a / 2a							
40	60	69	15.0%	67	11.7%	64	6.7%
50	100	118	18.0%	115	15.0%	109	9.0%
60	150	172	14.7%	168	12.0%	166	10.7%
70	200	234	17.0%	231	15.5%	227	13.5%
80	280	335	19.6%	329	17.5%	326	16.4%
90	380	457	20.3%	452	18.9%	445	17.1%
100	490	594	21.2%	587	19.8%	579	18.2%
110	680	836	22.9%	818	20.3%	816	20.0%

Table 6.4 Required increase in curve radius R_1 for reverse curves combined with vertical alignment (superelevation rate is 6%)

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 100% R1		R2 = 150% R1		R2 = 200% R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	55	62	12.7%	60	9.1%	58	5.5%
50	90	102	13.3%	100	11.1%	97	7.8%
60	130	149	14.6%	146	12.3%	142	9.2%
70	190	218	14.7%	211	11.1%	209	10.0%
80	250	291	16.4%	284	13.6%	280	12.0%
90	340	399	17.4%	392	15.3%	386	13.5%
100	440	534	21.4%	526	19.5%	511	16.1%
110	600	741	23.5%	730	21.7%	715	19.2%
(b) WB-20							
40	55	68	23.6%	66	20.0%	62	12.7%
50	90	112	24.4%	109	21.1%	104	15.6%
60	130	166	27.7%	160	23.1%	154	18.5%
70	190	241	26.8%	237	24.7%	226	18.9%
80	250	318	27.2%	309	23.6%	301	20.4%
90	340	438	28.8%	423	24.4%	411	20.9%
100	440	559	27.0%	550	25.0%	538	22.3%
110	600	764	27.3%	757	26.2%	737	22.8%
(c) NHTSA 3a / 2a							
40	55	66	20.0%	64	16.4%	61	10.9%
50	90	109	21.1%	105	16.7%	101	12.2%
60	130	158	21.5%	153	17.7%	148	13.8%
70	190	232	22.1%	226	18.9%	218	14.7%
80	250	308	23.2%	299	19.6%	290	16.0%
90	340	420	23.5%	411	20.9%	398	17.1%
100	440	546	24.1%	534	21.4%	519	18.0%
110	600	748	24.7%	732	22.0%	712	18.7%

Table 6.5 Rollover speeds for different design vehicles and superelevation rates ($R_2 = 100\%$ R_1)

Design Speed (km/hr)	Reverse Curve Only				Reverse Curve & Vertical Alignment			
	Superelevation 0.04		Superelevation 0.06		Superelevation 0.04		Superelevation 0.06	
	Rollover Speed (km/h)	Margin of Safety (km/h)	Rollover Speed (km/h)	Margin of Safety (km/h)	Rollover Speed (km/h)	Margin of Safety (km/h)	Rollover Speed (km/h)	Margin of Safety (km/h)
(a) WB - 15								
40	55	15	53	13	51	11	48	8
50	71	21	68	18	66	16	62	12
60	92	32	87	27	81	21	78	18
70	110	40	108	38	102	32	100	30
80	132	52	129	49	122	42	118	38
90	154	64	144	54	140	50	138	48
100	166	66	156	56	153	53	150	50
110	178	68	169	59	168	58	164	54
(b) WB - 20								
40	51	11	49	9	47	7	45	5
50	66	16	64	14	61	11	57	7
60	84	24	79	19	75	15	69	9
70	102	32	100	30	96	26	91	21
80	121	41	117	37	114	34	108	28
90	138	48	132	42	126	36	120	30
100	155	55	142	42	137	37	132	32
110	168	58	153	43	148	38	144	34
(c) NASHTA 3a / 2a								
40	53	13	50	10	49	9	46	6
50	69	19	67	17	63	13	59	9
60	87	27	82	22	77	17	74	14
70	106	36	102	32	98	28	94	24
80	126	46	121	41	119	39	114	34
90	146	56	139	49	138	48	134	44
100	159	59	152	52	151	51	144	44
110	172	62	166	56	163	53	155	45

Table 6.6 Rollover speeds for different design vehicles and superelevation rates ($R_2 = 150\%$ R_1)

Design Speed (km/hr)	Reverse Curve Only				Reverse Curve & Vertical Alignment			
	Superelevation 0.04		Superelevation 0.06		Superelevation 0.04		Superelevation 0.06	
	Rollover	Margin of	Rollover	Margin of	Rollover	Margin of	Rollover	Margin of
	Speed (km/h)	Safety (km/h)	Speed (km/h)	Safety (km/h)	Speed (km/h)	Safety (km/h)	Speed (km/h)	Safety (km/h)
(a) WB - 15								
40	72	32	69	29	62	22	56	16
50	88	38	81	31	77	27	72	22
60	116	56	107	47	98	38	92	32
70	149	79	133	63	122	52	112	42
80	188	108	172	92	162	82	150	70
90	218	128	206	116	211	121	201	111
100	233	133	221	121	224	124	217	117
110	251	141	238	128	241	131	232	122
(b) WB - 20								
40	66	26	62	22	57	17	52	12
50	82	32	77	27	74	24	70	20
60	105	45	98	38	92	32	88	28
70	127	57	121	51	116	46	106	36
80	151	71	144	64	148	68	132	52
90	184	94	171	81	164	74	158	68
100	197	97	188	88	182	82	171	71
110	212	102	203	93	198	88	186	76
(c) NASHTA 3a / 2a								
40	69	29	67	27	63	23	54	14
50	85	35	80	30	75	25	71	21
60	108	48	102	42	96	36	89	29
70	139	69	128	58	119	49	109	39
80	174	94	162	82	151	71	144	64
90	206	116	194	104	197	107	189	99
100	221	121	214	114	212	112	207	107
110	238	128	228	118	230	120	224	114

Table 6.7 Rollover speeds for different design vehicles and superelevation rates ($R_2 = 200\%$ R_1)

Design Speed (km/hr)	Reverse Curve Only				Reverse Curve & Vertical Alignment			
	Superelevation 0.04		Superelevation 0.06		Superelevation 0.04		Superelevation 0.06	
	Rollover Speed (km/h)	Margin of Safety (km/h)	Rollover Speed (km/h)	Margin of Safety (km/h)	Rollover Speed (km/h)	Margin of Safety (km/h)	Rollover Speed (km/h)	Margin of Safety (km/h)
(a) WB - 15								
40	107	67	102	62	96	56	91	51
50	138	88	122	72	108	58	99	49
60	155	95	144	84	122	62	109	49
70	188	118	166	96	147	77	125	55
80	228	148	204	124	186	106	169	89
90	254	164	229	139	223	133	211	121
100	274	174	254	154	248	148	232	132
110	288	178	272	162	266	156	251	141
(b) WB - 20								
40	102	62	96	56	90	50	86	46
50	126	76	112	62	101	51	97	47
60	144	84	122	62	107	47	101	41
70	162	92	144	74	126	56	116	46
80	204	124	187	107	166	86	144	64
90	231	141	221	131	204	114	188	98
100	251	151	244	144	233	133	208	108
110	269	159	259	149	251	141	231	121
(c) NASHTA 3a / 2a								
40	105	65	99	59	93	53	89	49
50	129	79	117	67	105	55	98	48
60	149	89	133	73	114	54	107	47
70	169	99	150	80	133	63	120	50
80	220	140	197	117	171	91	160	80
90	248	158	225	135	217	127	201	111
100	266	166	248	148	238	138	221	121
110	280	170	266	156	260	150	244	134

Table 6.8 Summary statistics of reverse curve model

<i>Independent Variable</i>	<i>Coefficient</i>	<i>t-Stat</i>
Intercept	-11.090	-0.995
$(V-30)^2$	0.107	117.76
$(e_{max})^{-1}$	3.693	8.006
g_1	-93.229	-2.078
r	-13.333	-2.553
Coefficient of determination, $R^2 = 0.995$		
Significance of F statistic = 0.000		

Scenario	Horizontal Alignment	Vertical Alignment
[Scenario 1] simple horizontal curve with radius R_1		
[Scenario 2] reverse horizontal curve with two opposite radii R_1 & R_2		
[Scenario 3] reverse horizontal curve with two opposite radii R_1 & R_2 superimposed by vertical alignment		

Figure 6.1. Schematic representation for different scenarios – Reverse curves

Figure 6.2 Procedures followed to quantify the effect of reverse curvature

Figure 6.3 Required increase in minimum curve radii for different values of reverse curve ratio for design vehicle WB-15 ($e = 0.04$) – reverse curvature effect only

Figure 6.4 Required increase in minimum curve radii for different values of reverse curve ratio for design vehicle WB-20 ($e = 0.04$) – reverse curvature effect only

Figure 6.5 Required increase in minimum curve radii for different values of reverse curve ratio for design vehicle 3a / 2a ($e = 0.04$) – reverse curvature effect only

Figure 6.6 Required increase in minimum curve radii for different values of reverse curve ratio for design vehicle WB-15 ($e = 0.06$) – reverse curvature effect only

Figure 6.7 Required increase in minimum curve radii for different values of reverse curve ratio for design vehicle WB-20 ($e = 0.06$) – reverse curvature effect only

Figure 6.8 Required increase in minimum curve radii for different values of reverse curve ratio for design vehicle 3a / 2a ($e = 0.06$) – reverse curvature effect only

Figure 6.9 Required increase in minimum curve radii for different values of reverse curve ratio for design vehicle WB-15 ($e = 0.04$) – reverse curvature & vertical alignment effects

Figure 6.10 Required increase in minimum curve radii for different values of reverse curve ratio for design vehicle WB-20 ($e = 0.04$) – reverse curvature & vertical alignment effects

Figure 6.11 Required increase in minimum curve radii for different values of reverse curve ratio for design vehicle 3a/2a ($e = 0.04$) – reverse curvature & vertical alignment effects

Figure 6.12 Required increase in minimum curve radii for different values of reverse curve ratio for design vehicle WB-15 ($e = 0.06$) – reverse curvature & vertical alignment effects.

Figure 6.13 Required increase in minimum curve radii for different values of reverse curve ratio for design vehicle WB-20 ($e = 0.06$) – reverse curvature & vertical alignment effects.

Figure 6.14 Required increase in minimum curve radii for different values of reverse curve ratio for design vehicle 3a/2a ($e = 0.06$) – reverse curvature & vertical alignment effects.

Figure 6.15 Margin of safety against rollover for different values of reverse curve ratio combined with vertical alignment – design vehicle (WB-20) and $e = 0.06$

Figure 6.16 Comparison between the effect of reverse curvature and the effect of vertical alignment on minimum radius for WB-15. $R_2 = 50\% R_1$ and $e = 0.04$

Figure 6.17 The effect of superelevation on vehicle stability, design vehicle WB-15 and R_2
 $= 100\% R_1$

Figure 6.18 The effect of superelevation on vehicle stability, design vehicle WB-20 and $R_2 = 100\% R_1$

Figure 6.19 The effect of superelevation on vehicle stability, design vehicle 3a/2a and $R_2 = 50\% R_1$

Figure 6.20 Required increase in minimum radius for different design vehicles, $R_2 = 100\%$

R_1 and $e_{\max} = 0.04$

Figure 6.21 Required increase in minimum radius for different design vehicles, $R_2 = 100\% R_1$ and $e_{\max} = 0.06$

Figure 6.22 Comparison for required minimum radius using both TAC and mathematical model for 3a/2a design vehicle ($R_2 = 100\% R_1$, $e_{\max} = 0.06$)

CHAPTER 7: MODEL DEVELOPEMENT - COMPOUND CURVES

7.1 Background

As was discussed earlier, the guidelines for compound curve design are brief and vague. The design guides advise the designer to use caution when using compound curves and to maintain the radius of the flatter curve to be generally not be more than 50% greater than the radius of the sharp curve. The guide also recommended a ratio of 1:1.25 to be more desirable on high-speed roads where the speeds are at or near the maximum for the curvature (TAC 1999). No particular reason or reference was given for that recommendation and no indication was found for the reason to select the ratio of 1:1.5 or 1:1.25, which seems to be selected by practice. No publications were found discussing vehicle stability on compound curves.

In using VDM to evaluate vehicle stability on compound curves, two different schemes were followed similarly to what followed before for reverse curves as shown in Figure 7.1. The first scheme was to investigate the effect of compound curvature by itself, and the second scheme was to investigate the effect of being superimposed by a vertical alignment. A description for each scheme is illustrated herein after to examine the effect of both schemes, either individually or combined together.

7.2 Effect of Compound Curvature

In order to investigate the effect of compound curvature, the procedures followed for each design vehicle were similar to those followed for reverse curves. The procedures are as shown in Figure 7.2:

- Input a flat alignment (horizontal curve with radius R_1) as a basis for comparison. This flat alignment includes simple horizontal curve and superelevation.
- Run simulation using design speed (V) for that horizontal alignment.
- Read the results (base scenario): rollover (if any) and lateral acceleration.

Scenario	Horizontal Alignment	Vertical Alignment
[Scenario 1] simple horizontal curve with radius R_1		
[Scenario 2] compound horizontal curve with two arc radii R_1 & R_2		
[Scenario 3] compound horizontal curve with two arc radii R_1 & R_2 superimposed by vertical alignment		

Figure 7.1 Schematic representation for different scenarios – Compound curves

- Introduce another horizontal curve in the same direction (with radius R_2) to form a compound curve consisting of two horizontal curves in the same directions with two different radii (R_1 & R_2).
- Read the results (scenario 1): rollover (if any) and lateral acceleration.
- Increase the horizontal curve radius (R_1) and re-read the results (scenario 2). Continue to increase the radius until the results obtained from (scenario 2) match the results obtained from base scenario.
- Record the radius R_1 from scenario 2, and compare it with the radius from scenario 1.
- Repeat previous steps with different radii of the new curve (R_2) keeping R_1 the same, so that the effect of the radius of the opposite curve on vehicle stability can be measured and quantified.
- Repeat previous steps for different values of R_1 and V .
- Repeat previous steps for a different superelevation.

Following the procedures, the required increase in sharper arc radii (R_1) corresponding to different values of flatter arc radii (R_2) are shown in Tables 7.1 and 7.2 for $e_{max} = 0.04$ and $e_{max} = 0.06$, respectively. The required increase is also shown graphically in Figures 7.3 through 7.8 for different types of design vehicles and different rates of superelevation.

Figure 7.2 Procedures followed to quantify the effect of compound curvature

The tables and figures show that the compound curvature, by itself, has some impact on truck stability. An increase in horizontal curve radius is needed to compensate for the effect of compound curvature. That increase ranges from 0% to 11.5%, depending on the radius of the flatter arc in the compound curve (R_2). More increase is generally required in sharper arc radius (R_1) where the flatter arc radius (R_2) is smaller. It should be noted that the increase required to compensate for the effect of compound curvature is generally smaller than that for reverse curvature. As shown in the tables, the required increase in radius R_1 also depends on superelevation rate. The increase for $e_{max} = 0.06$ was found to be generally greater

than that required for $e_{max} = 0.04$. This may be explained in light of the conservative rates of side friction used in the design guides as was discussed earlier in Chapter 5. It was also found that more increase is required for curves with low design speeds, which is a particular concern for freeway ramps, where horizontal compound curves are usually used along with horizontal reverse curves to minimize land use. Those ramps are usually having low design speed. It should be noted that the increase required in this stage is to compensate for the effect of the compound curvature only.

7.3 Effect of Vertical Alignment

In order to quantify the effect of vertical alignment, the compound curves used in scenario 2 were superimposed by different types of vertical alignments (upgrade, downgrade, crest curve, and sag curve) as shown in scenario 3 – Figure 7.1. The procedures are as follows:

- Superimpose the reverse alignment (obtained from scenario 2) by a specific vertical alignment (Figure 7.1).
- Re-read the results: rollover and lateral acceleration.
- Increase the horizontal curve radius (R_1) and re-read the results (scenario 3). Continue to increase the radius until the results (obtained from scenario 3) matches the results obtained from (scenario 2).
- Record the radius from scenario 3, and compare it with the radius from scenario 2.

Following the procedures shown earlier, the results for the required minimum radii and the required increase in curve radius R_1 are shown in Table 7.3 and Table 7.4 for $e_{max} = 0.04$ and $e_{max} = 0.06$, respectively. The results are also shown graphically in Figures 7.9 through 7.13. The values shown are the maximum radii values required based on all study configurations of vertical alignments (i.e. flat grade, upgrade, downgrade, crest curve, and sag curve). It was found that the most critical cases occur when the horizontal alignment is superimposed by either a sag curve or a downgrade where both the vehicle weight and the traction forces are in the same direction, which is the same like in reverse curves case.

The results show that another increase is required in the minimum radius is required to compensate for the effect of being superimposed by a vertical alignment. The overall increase in the minimum radius R_1 ranges from 5.0% to 25.5% as shown in Tables 7.3 and 7.4. It should be noted that the increase shown in Tables 7.3 and 7.4 is the overall increase to compensate for both the effect of compound curvature and the effect of being superimposed by a vertical alignment. Comparing Tables 7.1 and 7.3, and Tables 7.2 and 7.4, it is noticed that the effect of compound curvature is overpowered by the effect of vertical alignment, which indicates that the compound curvature has less effect on vehicle stability than the reverse curvature and also has less effect on vehicle stability than being superimposed by a vertical alignment.

7.4 Rollover Speeds

As was discussed earlier in Chapter 6, rollover for trucks is more likely to occur prior to skidding. The rollover speeds and margin of safety for different design vehicles (WB-15, WB-20 and 3a/2a) negotiating compound curves with R_2 / R_1 ratio equals 105%, 125% and 150% are shown in Tables 7.5, 7.6 and 7.7, respectively. It was found that for lower design speeds, rollover speeds are closer to design speeds with smaller margins of safety, even without being superimposed by a vertical alignment. However, rollover speeds for compound curves are not as critical as those for reverse curves. For example, for WB-20 truck negotiating a compound curve with curvature ratio of 105%, design speed 40 km/h and superelevation rate 6%, the rollover speed is 64 km/h with a margin of safety 24 km/h. If the same reverse curve was superimposed by a vertical alignment, the rollover speed decreases to 59 km/h with a margin of safety only 19 km/h. This is still a particular concern for freeway ramps, where horizontal compound curves superimposed by vertical alignments are commonly used. Noting that many ramps have unrealistically low design speeds in comparison with the design speed of the mainline roadway, safety problems may be encountered due to driver behaviour to overdrive ramps if their design speeds were not appropriately selected. It should be noted that the values shown in Tables 7.5, 7.6 and 7.7 are for eccentric load distribution condition, which could be a design control for some situations like trucks transporting mining rocks or heavy equipment.

7.5 Sensitivity Analysis

The values shown in Tables 7.1 through 7.4 represent conservative values based on all test alignments and configurations, where sag curves and downgrades generally controlled in most cases with eccentric cargo load distribution. Thus, these values will be more conservative for upgrades and crest curves. To determine the minimum radius required based on specific alignment configuration, which are not explicitly included in Tables 7.1 through 7.4, a mathematical model should be developed to be used for specific alignment configurations. In order to establish such proper mathematical model with logical explanation for all the variables, some sensitivity analysis should be conducted to determine the sensitivity of such proposed model towards different variables.

7.5.1 Effect of Compound Curvature vs. Vertical Alignment

Comparing Table 7.1 with Table 7.3 and Table 7.2 with table 7.4, it is noticed that the effect of compound curvature is overpowered by the effect of vertical alignment on the required increase in minimum radius. A graphical representation for this finding is shown in Figure 7.14 for vehicle type (WB-15), rate of superelevation 4% and $R_2 = 105\% R_1$. The effect of compound curvature is less than 20% the effect of the vertical alignment.

7.5.2 Effect of Superelevation

The required increase in minimum radii for $e_{max} = 0.06$ was found to be generally more than that required for $e_{max} = 0.04$, especially for lower speeds. This finding is true either for the case of compound curvature by itself, or the case of compound curvature combined with vertical alignment. This may be explained in light of conservative rates of side friction used in the design guides as was explained earlier in Chapters 5 and 6. The effect of superelevation on vehicle stability is shown in Figures 7.15 through 7.17 for design vehicles WB-15, WB-20 and 3a/2a, respectively and $R_2 = 105\% R_1$. As shown in the figures, the effect of superelevation is not as significant as other effects.

7.5.3 Effect of Design Vehicle

The required minimum curve radii for different design vehicles are shown in Figures 7.18 and 7.19 for superelevation rate 0.04 and 0.06, respectively. It was found that the increase required for WB-20 is generally greater than what is required for WB-15. As was mentioned before, WB-20 is the largest design vehicle in AASHTO tables, so that it is expected to have the largest variation in tire forces and reactions between front and rear tires, as well as between inside and outside tires. The required increase for NHTSA 3a/2a truck is generally lying between that for WB-20 (the largest design truck in AASHTO) and WB-15 (the moderate design truck running on North American roads). It should be noted that the properties of NHTSA

3a/2a heavy-truck combination were measured from the field, so that it is more appropriate to be used as a design vehicle.

7.5.4 Effect of The Ratio Between Sharper and Flatter Arcs

The effect of the ratio between sharper and flatter arcs on required increase in minimum curve radii is shown in Figures 7.3 through 7.14. This effect is affecting vehicle stability on compound curves. However, that effect is not as much as for reverse curves. That effect almost disappears for ratio greater than 150%, so that this maximum ratio was taken as an upper limit throughout experimental work.

7.6 Model Calibration

The values in Tables 7.1 through 7.4 represent conservative values based on all test alignments and configurations, where sag curves and downgrades generally controlled in most cases. Thus, these values will be more conservative for upgrades and crest curves. To determine the minimum radius requirements based on specific vertical alignment configurations, which are not explicitly included in Tables 7.1 through 7.4, a mathematical model was developed and may be used for specific alignment configurations. The model was calibrated using the simulation results obtained from VDM. The required minimum radius (dependent variable) was related to design speed and alignment data (independent variables). Many combinations of independent variables were examined and the final model was selected based on

the same criteria used before for both simple and reverse curves. A total of 540 observations were used to calibrate the mathematical model. The final model is given by

$$R_{min} = -81.644 + 0.086(V - 20)^2 + \frac{3.658}{e_{max}} - 600g_1, \quad [R^2 = 0.991] \quad (7.1)$$

where R_{min} = the required minimum radius for the first arc of a compound curve (m), V = design speed (km/h), e_{max} = maximum rate of superelevation, and g_1 is the first grade of vertical alignment (in decimal) as was explained earlier in Chapter 6. The relevant statistics are shown in Table 7.8. The coefficients of all independent variables are significantly different from zero at the 95% confidence level.

This model has logical explanation for the effect of each independent variable on R_{min} . The positive sign for the coefficient of V means that R_{min} increases with the increase in V . The positive sign for the coefficient of $1/e_{max}$ means that R_{min} decreases with the increase of e_{max} , as expected. The negative signs for the coefficient of g_1 means that R_{min} increases when g_1 is a downgrade. In this case, both vehicle weight and traction forces are in the same direction, resulting in maximum variation in the load distribution on different tires of the vehicle.

It should be noted that g_2 was rejected from the model as it was found to be insignificant, which is expected since the largest increase in the actual lateral

acceleration occurs in the first segment of the horizontal curve while the vehicle is still negotiating the first grade of the vertical alignment. The compound curve ratio was also rejected as it was also found to be insignificant. This model was calibrated using the results for 3a/2a design vehicle as that vehicle represents a majority of trucks really operated on North American roads. Since the properties of this heavy-truck combination were measured from the field, so that it is more appropriate to be used as a design vehicle as was discussed before. It should be also noted that the compound curvature ratio (r) is not as significant as it is in the reverse curve.

Table 7.1 Required increase in curve radius R_1 corresponding to different values of second arc radii R_2 (superelevation rate is 4%)

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 105% R1		R2 = 125% R1		R2 = 150% R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	60	61	1.7%	60	0.0%	60	0.0%
50	100	102	2.0%	101	1.0%	100	0.0%
60	150	154	2.7%	152	1.3%	151	0.7%
70	200	208	4.0%	203	1.5%	202	1.0%
80	280	292	4.3%	289	3.2%	284	1.4%
90	380	398	4.7%	394	3.7%	391	2.9%
100	490	514	4.9%	509	3.9%	506	3.3%
110	680	716	5.3%	710	4.4%	705	3.7%
(b) WB-20							
40	60	64	6.7%	63	5.0%	62	3.3%
50	100	108	8.0%	107	7.0%	105	5.0%
60	150	163	8.7%	162	8.0%	158	5.3%
70	200	218	9.0%	217	8.5%	211	5.5%
80	280	306	9.3%	307	9.6%	297	6.1%
90	380	417	9.7%	418	10.0%	409	7.6%
100	490	542	10.6%	542	10.6%	529	8.0%
110	680	754	10.9%	754	10.9%	738	8.5%
(c) NHTSA 3a / 2a							
40	60	62	3.1%	61	2.4%	61	1.5%
50	100	104	3.9%	104	3.7%	103	2.6%
60	150	157	4.7%	156	4.1%	155	3.4%
70	200	210	5.1%	209	4.6%	208	3.8%
80	280	295	5.4%	294	4.9%	291	4.0%
90	380	401	5.6%	400	5.2%	397	4.4%
100	490	518	5.8%	517	5.6%	514	4.8%
110	680	721	6.0%	720	5.9%	715	5.2%

Table 7.2 Required increase in curve radius R_1 corresponding to different values of the second arc radii R_2 (superelevation rate is 6%)

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 105% R1		R2 = 125% R1		R2 = 150% R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	55	56	1.8%	55	0.0%	55	0.0%
50	90	92	2.2%	91	1.1%	90	0.0%
60	130	134	3.1%	132	1.5%	131	0.8%
70	190	197	3.7%	193	1.6%	192	1.1%
80	250	261	4.4%	255	2.0%	253	1.2%
90	340	357	5.0%	351	3.2%	347	2.1%
100	440	464	5.5%	456	3.6%	450	2.3%
110	600	636	6.0%	625	4.2%	620	3.3%
(b) WB-20							
40	55	58	5.5%	57	3.6%	56	1.8%
50	90	96	6.7%	94	4.4%	92	2.2%
60	130	139	6.9%	136	4.6%	133	2.3%
70	190	207	8.9%	201	5.8%	197	3.7%
80	250	274	9.6%	266	6.4%	260	4.0%
90	340	373	9.7%	365	7.4%	355	4.4%
100	440	488	10.9%	474	7.7%	462	5.0%
110	600	669	11.5%	648	8.0%	633	5.5%
(c) NHTSA 3a / 2a							
40	55	57	3.6%	56	1.8%	55	0.0%
50	90	94	4.4%	92	2.2%	90	0.0%
60	130	136	4.6%	133	2.3%	131	0.8%
70	190	200	5.3%	196	3.2%	192	1.1%
80	250	264	5.6%	259	3.6%	254	1.6%
90	340	360	5.9%	354	4.1%	349	2.6%
100	440	468	6.4%	459	4.3%	455	3.4%
110	600	641	6.8%	629	4.8%	623	3.8%

Table 7.3 Recommended minimum radius for compound curves combined with vertical alignment (superelevation rate is 4%)

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 105% R1		R2 = 125% R1		R2 = 150% R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	60	65	8.3%	64	6.7%	63	5.0%
50	100	110	10.0%	108	8.0%	106	6.0%
60	150	166	10.7%	164	9.3%	163	8.7%
70	200	224	12.0%	223	11.5%	222	11.0%
80	280	326	16.4%	320	14.3%	318	13.6%
90	380	449	18.2%	440	15.8%	436	14.7%
100	490	584	19.2%	578	18.0%	572	16.7%
110	680	822	20.9%	814	19.7%	807	18.7%
(b) WB-20							
40	60	70	16.7%	68	13.3%	66	10.0%
50	100	117	17.0%	114	14.0%	111	11.0%
60	150	176	17.3%	172	14.7%	168	12.0%
70	200	236	18.0%	231	15.5%	227	13.5%
80	280	334	19.3%	329	17.5%	325	16.1%
90	380	458	20.5%	455	19.7%	448	17.9%
100	490	597	21.8%	592	20.8%	587	19.8%
110	680	840	23.5%	833	22.5%	826	21.5%
(c) NHTSA 3a / 2a							
40	60	68	13.3%	65	8.3%	64	6.7%
50	100	114	14.0%	110	10.0%	109	9.0%
60	150	168	12.0%	166	10.7%	163	8.7%
70	200	226	13.0%	224	12.0%	222	11.0%
80	280	330	17.9%	323	15.4%	320	14.3%
90	380	452	18.9%	442	16.3%	438	15.3%
100	490	595	21.4%	581	18.6%	574	17.1%
110	680	838	23.2%	821	20.7%	810	19.1%

Table 7.4 Recommended minimum radius for compound curves combined with vertical alignment (superelevation rate is 6%)

Design Speed (km/hr)	Design Minimum Radius (m)	R2 = 105% R1		R2 = 125% R1		R2 = 150% R1	
		Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)	Minimum Required Radius (m)	Required Increase (%)
(a) WB-15							
40	55	56	1.8%	56	1.8%	55	0.0%
50	90	92	2.2%	91	1.1%	90	0.0%
60	130	148	13.8%	142	9.2%	141	8.5%
70	190	221	16.3%	214	12.6%	211	11.1%
80	250	295	18.0%	289	15.6%	284	13.6%
90	340	408	20.0%	399	17.4%	389	14.4%
100	440	533	21.1%	524	19.1%	516	17.3%
110	600	731	21.8%	721	20.2%	711	18.5%
(b) WB-20							
40	55	59	7.3%	58	5.5%	57	3.6%
50	90	98	8.9%	97	7.8%	96	6.7%
60	130	156	20.0%	151	16.2%	149	14.6%
70	190	231	21.6%	224	17.9%	222	16.8%
80	250	307	22.8%	302	20.8%	297	18.8%
90	340	423	24.4%	416	22.4%	411	20.9%
100	440	550	25.0%	544	23.6%	534	21.4%
110	600	753	25.5%	748	24.7%	739	23.2%
(c) NHTSA 3a / 2a							
40	55	58	5.5%	56	1.8%	55	0.0%
50	90	96	6.7%	92	2.2%	90	0.0%
60	130	155	19.2%	151	16.2%	147	13.1%
70	190	229	20.5%	226	18.9%	220	15.8%
80	250	304	21.6%	299	19.6%	294	17.6%
90	340	418	22.9%	412	21.2%	407	19.7%
100	440	545	23.9%	537	22.0%	532	20.9%
110	600	748	24.7%	741	23.5%	730	21.7%

Table 7.5 Rollover speeds for different design vehicles and superelevation rates ($R_2 = 105\%$ R_1)

Design Speed (km/hr)	Compound Curve Only				Compound Curve & Vertical Alignment			
	Superelevation 4%		Superelevation 6%		Superelevation 4%		Superelevation 6%	
	Rollover	Margin of	Rollover	Margin of	Rollover	Margin of	Rollover	Margin of
	Speed	Safety	Speed	Safety	Speed	Safety	Speed	Safety
	(km/h)	(km/h)	(km/h)	(km/h)	(km/h)	(km/h)	(km/h)	(km/h)
(a) WB - 15								
40	68	28	66	26	64	24	61	21
50	94	44	91	41	88	38	85	35
60	109	49	105	45	99	39	92	32
70	136	66	129	59	122	52	112	42
80	155	75	147	67	141	61	139	59
90	188	98	176	86	168	78	152	62
(b) WB - 20								
40	66	26	64	24	61	21	59	19
50	91	41	88	38	83	33	81	31
60	102	42	97	37	92	32	88	28
70	128	58	119	49	114	44	105	35
80	151	71	144	64	136	56	132	52
90	180	90	153	63	148	58	142	52
(c) NASHTA 3a / 2a								
40	67	27	65	25	62	22	60	20
50	92	42	90	40	85	35	83	33
60	106	46	102	42	94	34	90	30
70	133	63	122	52	120	50	108	38
80	152	72	145	65	139	59	136	56
90	182	92	162	72	154	64	149	59

Table 7.6 Rollover speeds for different design vehicles and superelevation rates ($R_2 = 125\%$ R_1)

Design Speed (km/hr)	Compound Curve Only				Compound Curve & Vertical Alignment			
	Superelevation 4%		Superelevation 6%		Superelevation 4%		Superelevation 6%	
	Rollover Speed (km/h)	Margin of Safety (km/h)	Rollover Speed (km/h)	Margin of Safety (km/h)	Rollover Speed (km/h)	Margin of Safety (km/h)	Rollover Speed (km/h)	Margin of Safety (km/h)
(a) WB - 15								
40	94	54	91	51	88	48	87	47
50	116	66	112	62	109	59	106	56
60	144	84	136	76	132	72	128	68
70	166	96	154	84	144	74	141	71
80	223	143	214	134	201	121	198	118
90	322	232	314	224	302	212	292	202
(b) WB - 20								
40	91	51	89	49	85	45	81	41
50	108	58	104	54	99	49	92	42
60	132	72	122	62	112	52	109	49
70	155	85	141	71	132	62	128	58
80	210	130	201	121	192	112	187	107
90	311	221	302	212	293	203	289	199
(c) NASHTA 3a / 2a								
40	92	52	90	50	86	46	83	43
50	113	63	108	58	102	52	94	44
60	140	80	128	68	122	62	114	54
70	160	90	147	77	136	66	132	62
80	218	138	210	130	198	118	190	110
90	319	229	309	219	299	209	290	200

Table 7.7 Rollover speeds for different design vehicles and superelevation rates ($R_2 = 150\%$ R_1)

Design Speed (km/hr)	Compound Curve Only				Compound Curve & Vertical Alignment			
	Superelevation 4%		Superelevation 6%		Superelevation 4%		Superelevation 6%	
	Rollover	Margin of	Rollover	Margin of	Rollover	Margin of	Rollover	Margin of
	Speed	Safety	Speed	Safety	Speed	Safety	Speed	Safety
	(km/h)	(km/h)	(km/h)	(km/h)	(km/h)	(km/h)	(km/h)	(km/h)
(a) WB - 15								
40	107	67	102	62	96	56	91	51
50	138	88	122	72	108	58	99	49
60	155	95	144	84	122	62	109	49
70	188	118	166	96	147	77	125	55
80	228	148	204	124	186	106	169	89
90	277	187	255	165	223	133	211	121
(b) WB - 20								
40	104	64	100	60	91	51	88	48
50	132	82	112	62	102	52	95	45
60	144	84	136	76	118	58	101	41
70	166	96	154	84	141	71	116	46
80	215	135	189	109	177	97	155	75
90	265	175	214	124	211	121	192	102
(c) NASHTA $3a / 2a$								
40	106	66	101	61	94	54	89	49
50	136	86	118	68	105	55	98	48
60	148	88	139	79	120	60	107	47
70	174	104	158	88	145	75	120	50
80	214	134	192	112	179	99	160	80
90	270	180	232	142	226	136	205	115

Table 7.8 Summary statistics of compound curve model

<i>Independent Variable</i>	<i>Coefficient</i>	<i>t-Stat</i>
Intercept	-81.644	-5.486
$(V-30)^2$	0.086	88.469
$(e_{max})^{-1}$	3.658	6.139
g_1	-600	-5.919
Coefficient of determination, $R^2 = 0.995$		
Significance of F statistic = 0.000		

Figure 7.3 Required increase in minimum curve radii for different values of compound curve ratio for design vehicle WB-15 ($e = 0.04$) – compound curvature effect only.

Figure 7.4 Required increase in minimum curve radii for different values of compound curve ratio for design vehicle WB-20 ($e = 0.04$) – compound curvature effect only.

Figure 7.5 Required increase in minimum curve radii for different values of compound curve ratio for design vehicle 3a / 2a ($e = 0.04$) – compound curvature effect only.

Figure 7.6 Required increase in minimum curve radii for different values of compound curve ratio for design vehicle WB-15 ($e = 0.06$) – compound curvature effect only.

Figure 7.7 Required increase in minimum curve radii for different values of compound curve ratio for design vehicle WB-20 ($e = 0.06$) – compound curvature effect only.

Figure 7.8 Required increase in minimum curve radii for different values of compound curve ratio for design vehicle 3a / 2a ($e = 0.06$) – compound curvature effect only.

Figure 7.9 Required increase in minimum curve radii for different values of compound curve ratio for design vehicle WB-15 ($e = 0.04$) – compound curvature & vertical alignment effects.

Figure 7.10 Required increase in minimum curve radii for different values of compound curve ratio for design vehicle WB-20 ($e = 0.04$) – compound curvature & vertical alignment effects.

Figure 7.11 Required increase in minimum curve radii for different values of compound curve ratio for design vehicle 3a/2a ($e = 0.04$) – compound curvature & vertical alignment effects.

Figure 7.12 Required increase in minimum curve radii for different values of compound curve ratio for design vehicle WB-15 ($e = 0.06$) – compound curvature & vertical alignment effects.

Figure 7.13 Required increase in minimum curve radii for different values of compound curve ratio for design vehicle 3a/2a ($e = 0.06$) – compound curvature & vertical alignment effects.

Figure 7.14 Comparison between the effect of compound curvature and the effect of vertical alignment on minimum radius for WB-15. $R_2 = 105\% R_1$ and $e = 0.04$

Figure 7.15 The effect of superelevation on vehicle stability, design vehicle WB-15 and $R_2 = 105\% R_1$

Figure 7.16 The effect of superelevation on vehicle stability, design vehicle WB-20 and $R_2 = 105\% R_1$

Figure 7.17 The effect of superelevation on vehicle stability, design vehicle 3a/2a and $R_2 = 105\% R_1$

Figure 7.18 Required increase in minimum radius for different design vehicles, R_2
= 105% R_1 and $e = 0.04$

Figure 7.19 Required increase in minimum radius for different design vehicles,
 $R_2 = 50\% R_1$ and $e = 0.06$

CHAPTER 8: SUMMARY AND CONCLUSIONS

8.1 Summary

The combination of horizontal curves with vertical alignments is commonly used in different classifications of highways either on highway mainstream or on highway interchange ramps. The horizontal curves, combined with vertical alignments, may include single, compound or reverse horizontal alignments. The current geometric design guides do not adequately investigate vehicle stability on such 3D alignments. Computer software that simulates vehicle behaviour on different geometrical alignments was utilized to investigate vehicle stability on such 3D alignments. Since the conventional design guidelines are mainly based on flat simple horizontal curves, so that the main objective of the research was to compare vehicle lateral acceleration when negotiating such complex horizontal curves combined with vertical alignments with lateral acceleration when a vehicle negotiates a flat simple horizontal curve.

VDM RoAD (Vehicle Dynamic Models Roadway Analysis and Design) is a simulation program developed at the University of Michigan Transportation Research Institute (UMTRI). It accurately simulates a vehicle traveling through a user-defined alignment, taking into account vehicle characteristics such as body roll, pitch, yaw, and lateral weight distribution. VDM RoAD can analyze lateral acceleration,

directional control, roll stability, and stopping sight distance. VDM RoAD was used to evaluate vehicle dynamics and stability on different 3D alignment combinations.

Since the VDM RoAD has a comprehensive vehicle library, so that three different design vehicles were simulated. The first vehicle was (WB-15) design vehicle as it represents the moderate truck on North American roads. The second vehicle was (WB-20) design vehicle as it is the largest design truck in AASHTO. The last design vehicle to be simulated was the heavy truck combination (3a / 2a) combo, as it is a real truck on North American roads with its properties were measured from the field to support research at the National Highway Traffic Safety Administration (NHTSA). Although Sports Utilities Vehicle (SUV) has higher centre of gravity, which gives it potential risk for rollover, neither the vehicle library in VDM RoAD nor the design vehicle libraries in design guides include SUV as a design vehicle.

VDM RoAD was validated, along with VDANL and NADSdyna (Sayers 1999) by measuring actual lateral acceleration and other vehicle dynamics and comparing them with what was predicted by VDM RoAD. However, it should be noted that the validation was mainly based on flat alignments. The author is not aware of any validation for vehicle stability on 3D alignments.

The evaluation procedures include the evaluation for the effect of reverse or compound curvature on vehicle stability as well as the effect of being superimposed

by a vertical alignment. Flat, minimum radius, simple horizontal curves superelevated at 4% and 6% were used as a base scenario.

No transitional spiral curves were used in the geometric configuration as it was previously found that the effect of using transitional spiral curves has been found to be relatively minimal. Evaluating vehicle stability on two identical alignments, one with spiral curve and the other one without spiral curve validated this previous finding. It was found that the lateral acceleration values in both cases were almost identical, which confirms the previous finding that spiral curves have minimal effect on vehicle stability on horizontal curves.

Vehicle dynamics were recorded for the base simple 2D alignments in order to facilitate a comparison with subsequent complex horizontal curves combined with vertical alignments. The vertical alignments include upgrade, downgrade, crest curves and sag curves. The maximum grades of +6% and -6% were selected since this grade is the controlling limit used on Canadian rural highways. Eccentric cargo load distribution was used in the case of simple horizontal curves combined with vertical alignments as it was previously found to be the most critical case and resulting in the most conservative results. For the case of complex horizontal curves combined with vertical alignment, using eccentric cargo-load condition was found to result in unrealistic results. Hence, uniform cargo-load distribution condition was used.

8.2 Findings

It was found that for simple horizontal curves combined with vertical alignments, an increase in the minimum radius is required to compensate for the effect of the being superimposed by a vertical alignment and to maintain the same level of lateral acceleration. The increase ranges from 7.3% to 18.8%, depending on the design vehicle, superelevation and design speed.

For the case of reverse horizontal curves combined with vertical alignments, an increase in the minimum radius is required to compensate for the effect of reverse curvature. That increase ranges from 1.7% to 24.8%, depending on the design vehicle, superelevation, design speed and the ratio between the radii of the sharper and flatter arcs in the reverse curve. That ratio was found to be an important factor in vehicle stability on reverse curves, as more increase in the minimum radius is generally required where that ratio is smaller. It was also found that for reverse curves another increase in the minimum radius is required to compensate for the effect of being superimposed by a vertical alignment. The overall increase required for the minimum radius ranges from 3.3% to 27.3% to compensate for both the effect of reverse curvature and the effect of vertical alignment.

For the case of compound horizontal curves combined with vertical alignments, an increase in the minimum radius is required to compensate for the effect of compound curvature. That increase ranges from 0.0% to 11.5%, depending on the

same factors as for reverse curves. It was also found that another increase in the minimum radius is required to compensate for the effect of being superimposed by a vertical alignment. The overall increase required for the minimum radius ranges from 5.0% to 25.5% in order to compensate for both the effect of compound curvature and the effect of vertical alignment.

Another important measure for vehicle stability is the rollover speed. Rollover for trucks is more likely to occur prior to skidding due to suspension characteristics and higher mass centres. The rollover speeds for reverse curves were found to be closer to design speeds with smaller margin-of-safety, especially for lower design speeds. For compound curves, rollover speeds for lower design speeds were also found to be closer to design speeds, but not as critical as those for reverse curves. This is a particular concern for freeway ramps, where horizontal reverse or compound curves superimposed by vertical alignments are commonly used. Noting that many ramps have unrealistically low design speeds in comparison with the design speed of the mainline roadway, safety problems may be encountered due to driver behaviour to overdrive ramps if their design speeds were not appropriately selected.

Design aids were developed to determine the required minimum radius for different types of horizontal curves combined with vertical alignments. These design aids are based on the most conservative alignments tested, and therefore would be conservative for some other favourable alignments. To explicitly address the type of vertical alignment as well all other factors, mathematical models were developed to

relate the minimum radius requirements to design speed and different alignment configurations.

8.3 Recommendations

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

- Geometric design consistency should be promoted in North American design guidelines, either between horizontal and vertical alignment elements or among horizontal alignment elements by themselves. This consistency should reflect new design guidelines for the design of complex horizontal alignments as well as the coordination between those horizontal alignments and vertical alignments.
- For the case of simple horizontal curves combined with vertical alignments, an increase in the minimum radius requirements of TAC and AASHTO ranging from 7.3% to 18.8% is required on such 3D alignments to maintain the same level of lateral acceleration. The increase depends on the design vehicle, superelevation, and design speed.
- For the case of reverse horizontal curves combined with vertical alignments, an increase in the minimum radius is required to compensate for the effect of reverse curvature. That increase ranges from 1.7% to 24.8%, depending on the design vehicle, superelevation, design speed and the ratio between the radii of the sharper and flatter arcs in the reverse curve. That ratio was found to be an

important factor in vehicle stability on reverse curves, as more increase in the minimum radius is generally required where that ratio is smaller. It was also found that for reverse curves combined with vertical alignments, another increase in the minimum radius is required to compensate for the effect of being superimposed by a vertical alignment. The overall increase required for the minimum radius ranges from 3.3% to 27.3% to compensate for both the effect of reverse curvature and the effect of vertical alignment.

- For the case of compound horizontal curves combined with vertical alignments, an increase in the minimum radius is required to compensate for the effect of compound curvature. That increase ranges from 0.0% to 11.5%, depending on the same factors as for reverse curves. It was also found that another increase in the minimum radius is required to compensate for the effect of being superimposed by a vertical alignment. The overall increase required for the minimum radius ranges from 5.0% to 25.5% to compensate for both the effect of compound curvature and the effect of vertical alignment.
- The required increase in minimum radius presented in this research is based on current design values of side friction for freeways. These values were developed for passenger cars many years ago and should be revised to account for the characteristics of modern passenger cars and trucks. In addition, the new approach for truck design typically tends to provide optimum power/mass ratio to carry heavier cargo. Since such design generally results in higher mass centres, further research is required to establish safety margins against rollover.

- There is a need for establishing rollover as a design control for trucks in addition to the driver comfort that is currently being used in the design guides for all types of vehicles.
- The findings of this research were based on side friction factors for freeways, which is different from those for urban streets or for interchange ramps. Further research is required to interpret the findings from this research to be used for interchange ramps or urban streets.
- Additional research should be conducted to address human factors, including driver workload and perception-reaction time. Other situations where driver behaviour should be investigated should also be covered in further research, such as braking as a reaction to increasing lateral acceleration when it exceeds driver comfort limit, or steering as a reaction to increasing roll angle where a rollover is likely to occur.

REFERENCES

- American Association of State Highway and Transportation Officials. (2001). "A Policy On Geometric Design Of Highways And Streets". AASHTO, Washington, D.C.
- Al-Masaeid, H.R., Hamed, M., Aboul-Ela, M and Ghannam, A.G. (1995). "Consistency of Horizontal Alignment for Different Vehicle Classes". Transportation Research Record 1500, TRB, National Research Council, Washington, D.C., pp. 178 - 183.
- Baker, Derek, Bushman, Rob, and Berthelot, Curtis (2001). "Effectiveness of Truck Rollover Warning Systems". Transportation Research Record 1779, TRB, National Research Council, Washington, D.C., pp. 134 - 140.
- Barnett, J. (1936). "Safe Side Friction Factors And Superelevation Design". Highway Research Board, Vol. 16, 69-80.
- Battelle Team (1995). "Comprehensive Truck Size and Weight Study, Phase 1-Synthesis". Federal Highway Administration (FHWA), U. S. Department of Transportation, Working Paper 5 (from the Internet website).
- Bester, C.J. (2000). "Truck Speed Profiles". Transportation Research Record 1701, TRB, National Research Council, Washington, D.C., pp. 111-115.
- Blue, D. W. and Kulakowski, B.T. (1991). "Effects of Horizontal-Curve Transition Design On Truck Roll Stability". Journal of Transportation Engineering, ASCE, Vol. 117, No. 1, pp. 91-102.

- Bonneson, James A. (1999). "Side Friction and Speed as Controls for Horizontal Curve Design". *Journal of Transportation Engineering, ASCE*. Vol. 125, No. 6, pp. 473 – 480.
- Bonneson, James A. (2001). "Controls for Horizontal Curve Design". *Transportation Research Record 1751, TRB, National Research Council, Washington, D.C.*, pp. 82 - 89.
- Carlson, P.J. and Mason, J.M. (1999). "Relationship Between Ball bank Indicator Readings, Lateral Acceleration Rates and Vehicular Body-Roll Rates". *Transportation Research Record 1658, TRB, National Research Council, Washington, D.C.*, pp. 34-42.
- Chang, Tang-Hsien (2001). "Effect of Vehicle's Suspension on Highway Horizontal Curve Design". *Journal of Transportation Engineering, ASCE*. Vol. 127, No. 1, pp. 89 – 91.
- Chrstos, J. P. and Grygier, P. A., " Experimental Testing of a 1994 Ford Taurus for NADSdyna Validation," SAE Paper No. 970563, NHTSA, February 1997.
- Website http://www.nrd.nhtsa.dot.gov/vrtc/ca/capubs/taurus_test.pdf
- Cleveland, D.E., Kostyniuk, L.P. and Ting, K. (1984). "Geometric Design Element Groups and High-Volume Two-Lane Rural Highway Safety". *Transportation Research Record 960, TRB, National Research Council, Washington, D.C.* pp. 1-13.
- Dabbour, E., Easa, S.M. and Abd-El-Halim, A.O. (2003). "Design Guidelines for Horizontal Reverse Curves Combined with Vertical Alignments". *Proceedings, CSCE 31st Annual Conference, Moncton, NB.*

- Dabbour, E., Easa, S.M. and Abd-El-Halim, A.O. (2003). "Radius Requirements for Reverse Horizontal Curves on 3D Alignments". ASCE Journal of Transportation Engineering [under review].
- Dunlap, D.F., Francher, P.S, Scott, R.E., MacAdam, C.C. and Segal, L. (1978). "Influence of Combined Highway Grade and Horizontal Alignment on Skidding". NCHRP Report No. 184, TRB, National Research Council, Washington, D.C.
- Donnell, Eric T., Ni, Yingwei, Adolini, Michelle, and Elefteriadou, Lily (2001). "Speed Prediction Models for Trucks on Two-Lane Rural Highways". Transportation Research Record 1751, TRB, National Research Council, Washington, D.C. pp. 44 – 55.
- Easa, S.M. and Dabbour, E. (2003). "Design Radius Requirements for Simple Horizontal Curves on 3D Alignments". Canadian Journal of Civil Engineering [in press].
- Easa, S.M. (2002). Geometric Design. In Civil Engineering Handbook, W.F. Chen and J.Y.Liew eds., CRC Press, Boca Raton, FL, Chapter 63, 2002, pp. 1-39.
- Easa, S.M. (1993). "Unified Design of Horizontal Circular Curves". Journal of Transportation Engineering, ASCE. Vol. 119, No. 1. pp. 94-123.
- Easa, S.M. (1991) "Sight Distance Models for Unsymmetrical Sag Curves". Transportation Research Record No. 1303, TRB, National Research Council, Washington, D.C., pp. 51-62
- Easa, S.M. (1994). "Design Considerations for Highway Reverse Curves". Transportation Research Record No. 1445, TRB, National Research Council, Washington, D.C., pp. 1-11.

- Easa, S.M., (1994). "New and Improved Unsymmetrical Vertical Curve for Highways". Transportation Research Record No. 1445, TRB, National Research Council, Washington, D.C., pp. 94-100
- Easa, S.M., (1993). "Lateral Clearance Needs on Compound Horizontal Curves". Journal of Transportation Engineering, Vol. 119, No. 1, pp. 111-123.
- Easa, S.M. (1998). "Three-Arc Vertical Curve for Constrained Highway Alignments". Journal of Transportation Engineering. Vol. 124, No. 2, pp. 163-171.
- Easa, S.M. and Hassan, Y. (1998). "Design Requirements of Equal-Arc Unsymmetrical Vertical Curves". Journal of Transportation Engineering, Vol. 124, No. 5, pp. 404-410.
- Easa, S.M. and Hassan, Y. (1997). "Analysis of Headlight Sight Distance on Separate Highway Alignments: A New Approach." Canadian Journal of Civil Engineering, Vol. 24, No. 6, pp. 1007-1018.
- Easa, S.M. and Hassan, Y. "Transitioned vertical curves: I properties." Transp. Res., 34A(6), 2000, pp. 481-496.
- Easa, S.M. and Hassan, Y. "Transitioned vertical curves: II sight distance." Easa, S.M. Geometric design. In Civil Engineering Handbook, W.F. Chen and J.Y.Liew eds., CRC Press, Boca Raton, FL, Chapter 63, 2002, 1-39. Transp. Res., 34A(7), 2000, pp. 565-584.
- Federal Highway Administration (2002). "The Interactive Highway Safety Design Model (IHSDM): Making Safety A Priority In Roadway Design". Office of Safety

Research & Development, FHWA, McLean, VA. Internet:
www.tfhrc.gov/safety/ihsdm/brief.htm.

Federal Highway Administration (2001). "Recent Geometric Design Research For Improved Safety And Operations". FHWA, U.S. Department of Transportation, Washington, D.C.

Federal Highway Administration (2000). "The Capability and Enhancement of VDANL and TWOPAS for Analyzing Vehicle Performance on Upgrades and Downgrades Within IHSDM". Federal Highway Administration, FHWA, Report # FHWA-RD-00-078. Website <http://www.tfhrc.gov/safety/00-078.pdf>.

Felipe, E. and Navin, F. (1999). "Automobiles on Horizontal Curves: Experiments and Observations". Transportation Research Record No. 1628, TRB, National Research Council, Washington, D.C., pp. 50-56.

Fitzpatrick, K. (1994). "Horizontal Curve Design: An Exercise in Comfort and Appearance". Transportation Research Record No. 1445, TRB, National Research Council, Washington, D.C., pp. 47-53.

Fitzpatrick, Kay, Krammes, Raymond A., and Fambro, Daniel B. (1997). "Design Speed, Operating Speed and Posted Speed Relationships". ITE Journal. Vol. 67, No. 2, pp. 52 – 59.

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

- Furtado, G. (2002). "Vehicle Stability on Combined Horizontal and Vertical Alignments". Thesis submitted to Department of Civil and Environmental Engineering, Carlton University, Ottawa, Canada.
- Garcia, L.O., Wilson, F.R. and Innes, D. (2003). "Heavy Truck Dynamic Rollover: Effect of Load Distribution, Cargo Type and Road Design Characteristics". Annual Meeting of Transportation Research Board, Washington, D.C.
- Gibreel, G.M., Easa, S.M. and El-Dimeery, I.A. (2001). "Prediction Of Operating Speed On Three-Dimensional Highway Alignments". Journal of Transportation Engineering, ASCE, Vol. 127, No. 1, 21-30.
- Glauz, W.D. and Harwood, D.W. (1991). "Superelevation and Body Roll Effects on Offtracking of Large Trucks". Transportation Research Record No. 1303, TRB, National Research Council, Washington, D.C., pp. 1-10
- Harwood, D. W., J. M. Mason, J.M., Glauz, W.D., Kulakowski, B.T., and Fitzpatrick. K. (1990). "Truck Characteristics For Use In Highway Design And Operation". Reports FHWA-RD-89-226 and FHWA-RD-89-227, US Department of Transportation, Washington, D.C.
- Harwood, D.W. and Mason, J.M. (1993). "Ramp/Mainline Speed Relationships And Design Considerations". Transportation Research Record 1385, TRB, National Research Council, Washington, D.C., pp. 121-125.
- Harwood, D.W. and Mason, J.M. (1994). "Horizontal Curve Design For Passenger Cars And Trucks". Transportation Research Record 1445, TRB, National Research Council, Washington, D.C., pp. 22-33.

- Hassan, Y., Gibreel, G., and Easa, S. M. "Evaluation of Highway Consistency and Safety: Practical Application". ASCE Journal of Transportation Engineering, Vol. 126 No. 3, pp. 193 – 201.
- Haug, E. J., et. al. (1990), "Feasibility Study and Conceptual Design of a National Advanced Driving Simulator", NHTSA Contract DTNH22-89-07352, Report No. DOT-HS-807-597, March 1990.
- Hirsh, M., Prashker, J.N. and Ben-Akiva, M, (1986). "New Approach to Geometric Design of Highways". Transportation Research Record 1100. TRB, National Research Council, Washington, D.C., pp. 50-56.
- Hunter, Michael, Machemehl, Randy, and Tsyganov, Alexei (2001). "Operational Evaluation of Freeway ramp Design". Transportation Research Record 1751, TRB, National Research Council, Washington, D.C., pp. 90 - 100.
- Kontaratos, Marios, Psarianos, Basil, and Yotis, Apostolos (1994). "Minimum Horizontal Curve Radius as Function of Grade Incurred by Vehicle Motion in Driving Mode". Transportation Research Record 1445, TRB, National Research Council, Washington, D.C., pp. 86 - 93.
- Krammes, R. A., Brackett, R. Q., Shafer, M. A., Ottesen, J. L., Anderson, I. B., Fink, K. L., Collins, K. M., Pendleton, O. J. and Messer, C. J. *Horizontal Alignment Design Consistency for Rural Two-Lane Highways*. Report No. FHWA-RD-94-034, Federal Highway Administration, Washington, D.C., January 1995.
- Lamm, R. (1984). "Driving Dynamic Considerations: A Comparison of German and American Friction Coefficients for Highway Design".

- Transportation Research Record 960, TRB, National Research Council, Washington, D.C., pp. 13-26.
- Lamm, R., Choueiri, E.M., Goyal, P.B. and Mailaender, T. (1990). "Design Friction Factors of Different Countries Versus Actual Pavement Friction Inventories". Transportation Research Record No. 1260, TRB, National Research Council, Washington, D.C. pp. 135-145.
- Lamm, R., Choueiri, E.M, Hayward, J.C. and Paluri, A. (1988). "Possible Design Procedure to Promote Design Consistency in Highway Geometric Design on Two-Lane Rural Roads". Transportation Research Record 1195, TRB, National Research Council. Washington, D.C., pp. 111-122.
- Lamm, R., Choueiri, E.M. and Mailaender, T. (1991). "Side Friction Demand Versus Side Friction Assumed for Curve Design on Two-Lane Rural Highways". Transportation Research Record 1303, TRB, National Research Council. Washington, D.C., pp. 11-21.
- Lamm, R. and Smith, B.L. (1994). "Curvilinear Alinement: An Important Issue for More Consistent and Safer Road Characteristics". Transportation Research Record 1445, TRB, National Research Council, Washington, D.C., pp. 12-21.
- Liapis, Evangelos D., Psarianos, Basil, and Kasapi, Eva (2001). "Speed Behaviour Analysis at Curved Ramp Sections of Minor Interchanges". Transportation Research Record 1751, TRB, National Research Council, Washington, D.C. pp. 35 – 43.

- MacAdam, C.C., Fancher, P.S. and Segel, L. (1985). "Side Friction For Superelevation On Horizontal Curves". Transportation Research Institute, University of Michigan, UMTRI – 72895, Michigan.
- Mannering, F.L. and Kilareski, W.P. (1998). "Principles Of Highway Engineering And Traffic Analysis". Wiley and Sons, New York, N.Y., pp. 7-79.
- McGee, H. W. (1981). "Synthesis Of Large Truck Safety Research". Wagner-McGee Associates, Alexandria, Va.
- Meritt, D.R. (1988). "Safe Speed on Curves: A Historical Perspective of the Ball bank Indicator". ITE Journal, vol. 58, No. 9, pp. 15-19.
- Morrall, J. F. and Talarico, R. J. (1994). "Side Friction Demanded and Margin of Safety on Horizontal Curves". Transportation Research Record 1435, TRB, National Research Council, Washington, D.C., pp. 145 - 152.
- Moyer, R.A. and Berry, D.S. (1940). "Marking Highway Curves With Safe Speed Indications. Proceedings", Highway Research Board, Vol. 20, 399-428.
- NCHRP (2001). "Synthesis 299: Recent Geometric Design research for Improved Safety and Operations". Transportation research Board, TRB, National Research Council, Washington, D.C.
- Paniati, J.F. and True, J. (1996), "Interactive Highway Safety Design Model (IHSDM): Designing Highways with Safety in Mind," Transportation Research Circular No. 453, February 1996.
- Sayers, M.W. (1999). "Vehicle Dynamics Programs For Roadway And Roadside Studies". University of Michigan Transportation Research Institute, UMTRI-98-20-1, Michigan.

- Smith, B.L. and Lamm, R. (1994). "Coordination of Horizontal and Vertical Alignment with Regard to Highway Aesthetics". Transportation Research Record 1445, TRB, National Research Council, Washington, D.C. pp. 73-85.
- Stonex, K.A. and Noble, C.M. (1940). "Curve Design and Tests on The Pennsylvania Turnpike". Highway Research Board, Vol. 20, pp. 429-451.
- The University of Iowa (1995). "NADS Vehicle Dynamics Software - Release 4", Centre for Computer Aided Design.
- Transportation Association of Canada (1999). "Geometric Design Guide For Canadian Roads". Transportation Association of Canada (TAC), Ottawa, ON, Canada.
- Tsai, F. F. and Haug, E. J. (1989), "Automated Methods for High Speed Simulation for Multibody Dynamic Systems", Technical Report R047, Center for Computer Aided Design, The University of Iowa.
- University of Michigan Transportation Research Institute (1998). "Vehicle Dynamics Programs For Roadway And Roadside Studies". Final report. University of Michigan Transportation Research Institute. UMTRI-98-20-1.
- Wilson, J. (2002). "The National Highway Traffic Safety Administration's Rating System For Rollover Resistance: An Assessment". Special Report 265, National research Council, Transportation Research Board, Washington, D.C.
- Wong, J.Y. (1978). "Theory of Ground Vehicles". Wiley and Sons Inc. New York.

APPENDIX: NOTATION

The following symbols were used in the thesis: -

R	= Horizontal curve radius.
R_{min}	= Minimum radius for a horizontal curve.
PC	= Point of curvature for a horizontal curve.
I	= Deflection angle for a horizontal curve.
T	= Tangent distance for a horizontal curve.
E	= External distance for a horizontal curve.
M	= Middle ordinate for a horizontal curve.
C	= Length of chord for a horizontal curve.
L	= Length of a horizontal curve.
$W \& Q$	= Vehicle weight.
E	= Superelevation rate.
e_{max}	= Maximum superelevation rate.
V	= Vehicle speed.
G	= Acceleration due to gravity.
BBI	= Ball bank indicator reading.
ULA	= Unbalanced lateral acceleration or side friction factor.
f	= Side friction factor.
f_{max}	= Maximum side friction factor.
r_v	= Maximum rate of change for vertical curves.
K	= Vertical curve length corresponding to a change in grade of 1%.
r	= Ratio between flatter and sharper arcs in horizontal complex curves.
g_1	= First grade of a vertical curve.
g_2	= Second grade of a vertical curve.