**Consideration of the Effect of Vehicle Stability
Augmentation Algorithm Failure Modes
on Vehicle Behaviour**

by

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Abstract

One of the most critical driving situations is the loss of directional control. For this research, a 2003 Ford Windstar was considered as a representative vehicle and the ADAMS multibody dynamics software was used to develop a vehicle model. The mathematical model was validated by comparison with conventional analytical models. Full-scale experimental tests were run with the experimental vehicle to further validate that the ADAMS built-in driver model is representative of actual driver behaviour. A representative electronic stability control (ESC) algorithm was developed and implemented in Fortran then interfaced to ADAMS to represent the ESC system. It was then customized to represent the different potential failure modes for various sensors detecting longitudinal velocity, steering wheel angle, and yaw rate during a standard double lane-change manoeuvre. Simulation results were analyzed. From this study it was concluded that the effect of the failure modes on the vehicle behaviour was significant and could lead to large lane excursions and potential loss of control. On the other hand, the positive benefits of ESC in safely completing aggressive manoeuvres was demonstrated.

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for their contribution during all my steps and for all advise and encouragement that they
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List of Symbols

<i>Symbol</i>	Description
SSF	Static stability factor
T_w	Track width
H	Height of the centre of gravity above the ground
W_f	Front axle load
W_r	Rear axle load
l_1	The distances between the front and the centre of mass
l_2	Distances between the rear axles and the centre of mass
W	Vehicle weight
$F - R_r$	Net tractive force transmitted between the road and wheels
L	Wheelbase
$W_{fd}\%$	Percentage difference between the simulated and calculated load distributions on the front axle respectively
$W_{rd}\%$	Percentage difference between the simulated and calculated load distributions on rear axle respectively
W_{fa}	Front axle load simulated by ADAMS
W_{ra}	Rear axle load simulated by ADAMS
W_{fc}	Front axle load calculated manually
W_{rc}	Rear axle load calculated manually
ρ	Air density
C_D	Aerodynamic drag coefficient
A_f	Frontal area of the vehicle
V_r	Relative velocity between the vehicle and the wind
a	Vehicle inertial acceleration

$Gain_{acc.}$	Lateral acceleration gain
a_y	Lateral acceleration
δ_f	Ackerman steer angle of the front wheels
V	Forward vehicle speed
g	Acceleration due to gravity
K_{us}	Vehicle understeer coefficient
$Gain_{curv}$	Curvature gain
R	Radius of curvature
Ω_{zd}	Desired yaw rate
Ω_{za}	Actual yaw rate
δ_{swa}	Input steering wheel angle
ξ_s	Steering gear ratio
K_P	Proportional gain
P_b	Brake pressure applied over the brake pad/drum effective contact area
A_b	Effective contact area where the brake pressure is applied
T_b	Braking torque on the wheel
r_r	Tire's rolling radius
r_b	radius from the axis of rotation to the effective contact area where the brake pressure applies
K_{pa}	composite ESC controller gain
M	Moment about the centre of gravity of the vehicle
F_b	Braking force applied on each individual tire
P_b	Braking pressure
Y_{dis}	Lateral displacement from the vehicle centre of mass to the centre line of the middle lane
F	On-off failure mode

H	Periodic time
ω	Failure mode frequency
t	Time at which failure modes occur
A	Amplitude of the failure mode signal
K	Degraded performance failure mode
N	Sum of the individual noise signals
S_i	Noise signal component i
A_i	Amplitude of individual noise signal i
ω_i	Frequency of the i^{th} noise signal component

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Chapter 1

Introduction

1.1 Motivation

In response to the potential risks associated with loss of vehicle directional stability, automotive manufacturers have evolved vehicle safety systems (passive and active) and are upgrading them continuously. This research introduces the importance of vehicle safety from the perspective of directional control, presents vehicle safety-focused technology and describes the implementation in simulation of a representative vehicle electronic stability control (ESC) system including validation of the vehicle model mathematically, implementation of a simple ESC controller, experimentation using actual road test results to qualitatively support the vehicle model behaviour, and investigation of the effect of several ESC sensor fault modes.

1.2 Single Vehicle Accidents

Achieving a high level of vehicle safety requires that technological efforts focus on the root causes of driving hazards.

Figure 1.1: Distribution of causes of fatal vehicle accidents.

1.2.1 Statistics for Road Fatality

A report prepared by the Institute for Automotive Safety [1] categorized the root causes of fatalities in automobile accidents. It was found that 38% of fatal road accidents are due to physiological factors, 46% are due to the false assessment of the situation by the driver (including prevailing vehicle dynamics or misjudging the road and road conditions), 11% are due to unexpected behaviour, and 5% are due to technical problems. The relative proportions are highlighted in Figure 1.1.

In each of the four categories, injury occurs without any conflict with other road users or from collisions resulting from loss of control. Statistics published by one of the manufacturers of hydraulic and electronic safety and stability control systems [2] indicate that vehicle crashes remain the leading cause of death for Americans aged 6 through 33. On average, one American dies in a traffic accident every 13 minutes. While human safety is of paramount importance, the economic impact of motor vehicle accident injuries and deaths exceeds \$230 billion per year. The 2001 Ontario Road Safety Annual Report [3] provides a comprehensive perspective of road safety in Ontario over a thirty year period. The results, summarized in Figure 1.2, indicate that while the number of licensed drivers

Figure 1.2: The effectiveness of road and vehicle safety enhancements over the period from 1970 through 2001 [3].

is increasing linearly and has more than doubled since 1970, the annual number of vehicle fatalities is steadily decreasing and almost halved over the same period of time.

1.2.2 Statistics for Road Safety Due to Using ESC

In November 2002 Daimler Chrysler (DC) announced in a press release that by analyzing 1.5 million pieces of sampled accident data from 1998 to 2001, accumulated by the Federal Statistical Office in Germany, comparing DC vehicles with an ESC safety system in 2000/2001 with DC vehicles without an ESC safety system in 1999/2000, the accident rate for cars with ESC was 15% lower [4].

From analysis of the statistics of traffic accidents in Japan for Toyota passenger cars with and without an ESC system [4], it was estimated that the accident rates for vehicles with ECS showed approximately a 35% reduction for single car accidents, a 30% reduction for head-on collisions with other automobiles, and approximately 50% and 40% reductions for accidents where severe or moderate damage occurred respectively, compared to vehicles without ESC.

Although the safety enhancement due to ESC systems is already proven, only 9% of all vehicles manufactured in the U.S. are equipped with ESC systems compared to more than 30% in Europe.

As recent crash data studies from Germany and Japan have shown significant crash reductions with ESC systems, the National Highway Traffic Safety Administration (NHTSA) has an interest in evaluating this technology to determine whether these benefits might also be attainable in the U.S. [5].

NHTSA is currently conducting a comprehensive program of ESC hardware and performance testing using vehicles that are being run with and without ESC on a test track and is currently developing plans to use the National Advanced Driving Simulator (NADS) to examine how ESC affects average drivers' ability to avoid crashes. This research will assist NHTSA in understanding for which situations (i.e. event type and pavement conditions), drivers age, and vehicle types the ESC system is most helpful.

Improving vehicle safety must consider the entire human, vehicle, and environment (HVE) system [6]. The observed enhancement in safety results from improvements in both road design and new technology in vehicle safety and stability control systems. Just as clear zone widths, reduced side slope grades, introduction of rumble strips, and removal of roadside hazards have been implemented to reduce vehicle run-off-road events and generally improve road safety, technological advances such as anti-lock brake systems (ABS), traction control systems (TCS), and electronic stability control (ESC) have played a significant role in improving safety through improvements in vehicle design. Reducing vehicle accidents resulting from loss of directional stability remains a key focus in transportation safety projects.

1.3 Vehicle Safety Systems

Driver safety systems present in vehicles, in general, can be classified in two main categories: “passive” and “active”.

1.3.1 Passive Safety Systems

The purpose of passive systems is to protect the driver and occupants from injury after an accident has occurred. Passive systems include seat belts and air bags.

Seat belts

Seat belts function by holding the occupants in their seats during a collision. To achieve maximum protection, it is necessary that the vehicle occupants decelerate along with the vehicle after minimal forward movements out of their seats following an impact. This is achieved by triggering the belt tightener assembly to engage immediately upon initial impact to ensure that effective restraint for the passengers is immediately available.

Air bags

Air bags are an effective injury prevention device as they reduce the number of fatalities and mitigate major injuries and have been widely used in new motor vehicles. In fact, driver-side air bags are installed in over 90 percent of all 1994 model-year cars, and over half of these also have passenger air bags. U.S. law required all new cars to have both driver and front passenger air bags by 1998 (and trucks by 1999).

The function of air bags is to protect the driver and front passengers against head and chest injuries during collisions, deployed air bags are shown in Figure 1.3.

Upon collision, as detected by sensors, the air bags are inflated instantaneously to cushion the driver and passenger. Due to concern over potential hazards associated with air bag

Figure 1.3: Activation of air bags during collision.

deployment, a simple electronic control unit processes the input signals from peripheral acceleration sensors which are mounted at suitable points on the vehicle body to detect side impact and trigger the air bags.

1.3.2 Active Safety Systems

While passive technologies improve safety following impact, the preferred approach is to improve vehicle performance and handling such that accidents can more effectively be avoided. This is the approach underlying active vehicle safety control systems. Figure 1.4 [7] shows the ever-increasing role of control technology in vehicle systems and the associated growing dependence on electronic parts as the main components of these control systems. According to widely published information, the effect of vehicle stability on human safety and driver responses has motivated most car manufacturers to focus significant innovation and research on stability augmentation algorithms. They have designed control schemes that attempt to provide a maximum level of stability for the vehicle and, thus, a high level of safety for the driver.

Figure 1.4: Increasing role of electronic vehicle control systems [7] .

1.4 ESC Development

The origin of ESC can be directly related to ABS and TCS. ABS systems function by using wheel-speed sensors to detect whether a wheel is showing a tendency to lock. In the case of lock-up tendency, an electronic control unit commands reduction in the braking pressure individually on the wheel concerned. This high-speed correction of the braking pressure allows the vehicle to achieve optimal braking without loss of steering ability associated with front-wheel lock-up or loss of directional stability associated with rear-wheel lock-up. ABS systems include: the wheel-speed sensors, the electronic control unit (ECU), the hydraulic modulator, and the wheel brakes. The wheel-speed sensors output their signals to the electronic control unit, which uses them to calculate the peripheral speeds of the wheels. The ECU implements controller logic that controls the solenoid valves in the hydraulic modulator that applies braking pressure. Using the solenoid valves, the hydraulic

modulator transmits the ECU's control signals and controls the wheel brake pressure such that each wheel develops approximately the ideal braking effort. The braking pressure transferred from the hydraulic modulator serves to force the brake linings against the brake drums or the brake pads against the brake discs [8]. TCS makes use of the same components as ABS and automatically controls drive torque applied at each wheel consistent with the ability of the wheels to develop frictional force at the wheel/road interfaces. If the driver demands excessive torque without TCS, one or more wheels could start to spin and the vehicle could become uncontrollable. With TCS, the system would detect the tendency to spin and adapt the torques by intervening in the engine-management system and the wheel braking system in order to achieve overall improvement in traction [8]. ABS prevents wheel-lock when the brakes are applied and TCS prevents wheel-spin during drive-off. Both ABS and TCS increase vehicle stability in the vehicle's longitudinal direction and correspondingly in cornering ability. ESC exploits the control of wheel longitudinal tractive and braking effort to increase the stability of the vehicle in the transverse or yaw direction and correspondingly reducing the possibility of skidding. Due to the fact that steering is essentially affected by using the brake system, ESC is sometimes simply referred to as a system that implements *brake steering*. ESC is beneficial for avoiding obstacles, minimizing the effect of sudden wrenching of the steering wheel and facilitating driving on varying road surfaces. An ESC system senses the vehicle's longitudinal and lateral dynamics; senses the driver's commands including steering wheel angle, brake pressure, and engine torque; and identifies critical situations, in many cases before the driver does. ESC then considers the possible ways of intervening to improve the situation including individual brake pressure application and engine management. In addition to the components that are common to ABS and TCS, ESC also requires a number of additional sensors that include a yaw rate sensor, steering wheel angle sensor, lateral acceleration sensor and brake pedal pressure sensor [9] as reflected in Figure 1.5. Each wheel, along with its

Figure 1.5: Schematic illustration of an integrated ESC control system [2].

pneumatic tire, develops a cornering or guiding force when the wheel's pointing vector is not coincident with its velocity vector; or alternatively stated, when a slip angle is present. Vehicles are described to exhibit understeer behaviour where their front end slip angle increases more rapidly than their rear end slip angle as lateral acceleration rises. The opposite condition is referred to as oversteer. With ESC, both understeer and oversteer can be controlled by applying brake force at individual wheels to achieve the compensating yaw moment required to adjust the vehicle orientation. Braking force would be applied, in the case of a left hand turn, to the left rear wheel in the case of understeer and the right front wheel in the case of oversteer as shown in Figure 1.6. Most advanced control system manufacturers integrate their system with a monitoring and diagnostic system to detect the output and performance of individual components of the system and correlation between their responses. They mainly depend on the *on-board diagnosis* (OBD) technique for detecting anomalous signals and troubleshooting. The systems include software that

Figure 1.6: ESC control of understeer and oversteer behaviour in a left turn [10].

evaluates the system performance with a continuous series of checks. In the event that a problem is detected, a warning is displayed to the driver using an indicator lamp, which alerts the driver of the problem or malfunction. The system also saves fault memory data in the form of specific error codes that will subsequently be available for root cause analysis. This data is retained even through power-off conditions. The continuous testing of sensors and actuators is primarily performed to immediately detect sensor malfunction, defective system components, or erroneous signals resulting from other sources. If a fault is detected, the system will either immediately deactivate the ESC or allow the current vehicle manoeuvre to be completed. The selected option generally depends on the specific defect and instantaneous operating condition [9]. It is conceivable that the ESC will be disabled during a manoeuvre where it is contributing significantly to the vehicle stability. Consequently, this has lead researchers to focus some attention on fault detection and failure analysis, the effect of switching to another redundant system if required, and the

potential hazards possible throughout the transient period as the system reverts to an alternative mode. Manufacturers indicate that the dominant factor controlling the vehicle at all times is physical human control. The response of the driver/vehicle system in the transient period following an ESC fault is of interest and is the focus of this research.

1.5 Literature Review

Various topics relevant to the research will be discussed in this section focusing on the electronic stability program and fault detection.

The problem of losing control that faces drivers when trying to keep the vehicle from spinning at of control during aggressive manoeuvres is discussed by Zanten [11] and significant advantages of using the ESC control system to help keep the vehicle under control and stable are cited. ESC was found to be a robust control system and maintained excellent performance during panic braking. Due to ESC, a significant reduction in vehicle stopping distance was achieved when braking while cornering compared to vehicles equipped with ABS alone (in fact, by more than 15% during braking while cornering). Regarding the concept of vehicle handling, Forster [12] focused on the average driver and human behaviour noting that extra attention compared to the other considerations related to the concept of vehicle handling is required when designing ESC systems. Since the driver is not aware of the actual instantaneous available friction capacity between the tires and the road, the driver may panic when aggressive handling is required; whereas ESC automatically implements a near optimal solution for ensuring vehicle stability. The importance of human factors in reacting properly even when an ESC is activated was discussed by Zobel, Friedrich and Becker [13]. Accidents could not be prevented using ESC if the driver reacts recklessly and depends exclusively on the system. Simulations were conducted with and without ESC and the enhancement in the stability using ESC was proven.

Figure 1.7: National Advanced Driving Simulator illustration. [14]

A study compared the driver's performance in cases of critical loss of control both with and without the ESC system. Several road test scenarios were simulated using the NADS as shown in Figure 1.7 for two different vehicles models: the Oldsmobile Intrigue and Ford Excursion. The participants for the research were from three age groups. The results show an 88% reduction in the loss of control when using ESC. An overall reduction in loss of control was observed for each of the events simulated across the age groups. From an improvement standpoint, 34% more drivers retain control with ESC than without ESC. Babala and Kempen [15] discussed the effect of various defect modes for ESC as the performance of the vehicle is directly related to the performance of the ESC and its electronic components such as sensors and actuators. Research also focused on the diagnostics that must be quick enough to maintain consistent vehicle performance despite corrupted signals sent to the control system which cause destabilizing activation of the (ESC) system. Fault detection and handling discussed by Tubb and Omerdic [16] indicate that a fault can be either a failure in a physical component or a change in system performance. Although good design practice tries to minimize the occurrence of faults and failures, recognition that such events do occur, enables system designers to develop strategies by which the

effect they exert is minimized. They also categorized the fault detection and isolation into a two-stage process: the first stage includes a condition that if the fault has occurred it must be recognized (fault detection); and the second stage includes a requirement that the nature of a fault should be determined such that appropriate remedial action may be initiated (or the problem isolated).

1.6 Overview

This chapter discussed the motivation for using electronic control systems in vehicles to improve stability and passenger safety. It referred to data collected from various statistics, introduced different types of safety systems and their effects on driving behaviours, and provided an introduction to the failure of the electronic parts of the control systems that is the focus of this research. A number of topics relevant to the research have been reviewed in electronic stability control ESC (as an advanced control system) and electronic fault detection. Chapter 2 deals with the research vehicle, its description and various components. It discusses the ADAMS simulation model used in the research, the validation process, and the controller used in this model. Chapter 3 deals with describing the on-road experimental tests and the instrumentation used and compares the results qualitatively with the ADAMS model followed by a brief discussion. Chapter 4 discusses a variety of electronic failures types such as on/off failure, degraded performance and interference of noise signals accompanied by its effect on the stability of the vehicle. Chapter 5 presents the conclusions derived from the research and future work that is recommended.

Chapter 2

Model and Controller Development

2.1 Research Vehicle Description

The vehicle considered in the current research is a 2003 Ford Windstar LX minivan shown in Figure 2.1. This choice is based on two primary considerations: such a research vehicle is available in the Faculty of Engineering at Carleton University and the vehicle characteristics make it well suited to the current research programme.

The vehicle is owned and operated by the Department of Civil and Environmental Engineering in support of a highway geometrical design research programme. It can be equipped with different instruments to measure the characteristics of road surfaces and different measures of driver behaviour. Features of the vehicle that make it attractive for the current research include the following.

- It is front wheel drive so the weight of the engine and transmission over the front driving wheels provides good traction in severe road and weather conditions that could cause rear-wheel drive vehicles to lose traction and begin skidding.
- It is equipped with four-wheel ABS brakes that use computer controlled sensors to assist the driver in the event of impending wheel lock-up, using computer control of

Figure 2.1: Ford Windstar LX research vehicle.

Table 2.1: 2003 Windstar configuration

Parameter	Value
Wheelbase	3.066 m
SSF	1.26
Front Track	1.633 m
Rear Track	1.600 m
Load at the front axle	12.49 kN
Load at the rear axle	11.84 kN

the hydraulic system to pump the brakes more quickly than human reflexes permit.

- It uses rack and pinion steering that is a simple and direct system that responds quickly to driver inputs and provides drivers some direct sense of conditions at the front wheel and tire/road interface.

Key specifications and parameters for the Windstar research vehicle that are used in subsequently-presented simulation work are provided in Table 2.1.

Note that the height above the ground of the centre of gravity of the vehicle h is calculated using the formula

$$h = T_w/2SSF$$

where SSF is the static stability factor and T_w is the average track width of the vehicle, where track widths are the distances between the centres of the left and right tires at the front and rear of the vehicle.

2.2 ADAMS Model

The Windstar research vehicle was mathematically modelled using the Automatic Dynamic Analysis of Mechanical Systems (ADAMS), that is a multibody dynamics software package produced by the MSC software company. The company is considered one of the leading global providers of virtual product development (VPD) solutions including simulation software and services. ADAMS allows engineers to simulate, understand, and quantify the performance of mechanical systems before building expensive hardware prototypes and conducting physical tests. It also allows engineers to gain in-depth understanding of the performance of mechanical systems. Specifically, the ADAMS-Chassis module was used to provide an efficient means of implementing a fully-coupled vehicle dynamics model and appropriate models of the vehicle steering and braking systems. Implementing the vehicle model mainly involved customizing a minivan template for ADAMS-Chassis. The software provides capability for simulating a variety of driving tasks under conditions ranging from ideal to severe. Further, it was possible to use advanced control systems for the anti-lock brake system and electronic stability control system by interfacing ADAMS to user-written Fortran subroutines that implement control algorithms. Figure 2.2 provides an overview of the Windstar model. The model implemented by ADAMS-Chassis considers a sophisticated multibody diagram model that includes large numbers of 3-dimensional

Figure 2.2: ADAMS model.

linkages, joints, bushings, and connectors that form the main components of the Windstar. It consists of the main body model that includes the chassis (sprung mass) and the body mounts connected by the independent Macpherson and semi-independent suspension types respectively in the front and the rear of the vehicle. The pneumatic tires in ADAMS were modelled using the Pacejka 89 model which is similar to the Majic formula [10]. The front Macpherson suspension model and the associated legend are shown in Figure 2.3. This provides an example of the sophisticated model implemented and simulated using ADAMS chassis. The model consists of a strut connected to the chassis from the top by the top mount and located at the centre of the coil spring which in turn is guided by the upper and lower spring seats, the strut connected from the lower end to the lower control arm by the lower ball joint, the anti-roll bar connected by right and left bushings to the drop links which are directly connected with the dampers. In the Windstar model, the front suspension is of the independent Macpherson type that allows each wheel to move vertically without affecting the opposite wheel. This configuration is typical as it facilitates engine placement and distributes suspension loads widely [17]. The most unique feature of a MacPherson strut suspension is that all of the components are contained in a single assembly as shown in Figure 2.4. The typical MacPherson strut assembly can be

No.	Component	No.	Component
1	Top mount	6	Lower control arm (rear)
2	Spring	7	Anti-roll bar
3	Lower ball joint	8	Anti-roll bar bushing (right)
4	Subframe front (left)	9	Drop link bar (right)
5	Lower control arm (front)	10	Drop link external (right)

Figure 2.3: The description of the MacPherson strut components.

modelled efficiently using ADAMS-Chassis in a complete detailed design which provides successive accurate modelling tools for the rest of the vehicle. Basic manual simulation includes many simplifications and approximations in order to implement the same model that are not required using ADAMS.

The rear suspension is of the semi-independent type and uses a modified beam axle called a twist axle. When one wheel reacts to a bump, the axle flexes, thereby absorbing some of the impact instead of transmitting the full impact to the other wheel. This offers the

Figure 2.4: Different MacPherson strut modelled by ADAMS.

advantage of providing a smoother ride and improved handling control. A trailing twist axle rear suspension is widely used in front wheel drive vehicles because of the simple construction and light weight. It consists of two trailing arms interconnected by a twist beam [18] as shown in Figure 2.5. They are attached in a rigid manner in order to assure stability when vertical and lateral forces occur.

The widely-used rack and pinion steering system is favoured due to the advantages of complexity reduction and easier accommodation of drive line components in the front-wheel drive Windstar [17]. The rack and pinion gear system transmits steering wheel movement to the front wheels and is generally considered to be the ideal gear system for a car equipped with a power-steering system. Rack and pinion steering is widely used for various types of vehicles. When the rigidity of the mounting for the rack and pinion on the body increases the driver becomes able to control the vehicle accurately.

The ADAMS-Chassis brake template was used to simulate the brake system that was also

Figure 2.5: Twist axle.

integrated with the stability control system. A Fortran subroutine was implemented to simulate the effect of ABS on avoiding wheel lock-up and improving the longitudinal slip ratio. Once implemented, the completed computational ADAMS model was validated to ensure an accurate representation of quantitatively-expected results.

2.3 Model Validation

The ADAMS-Chassis vehicle model is a customized version of the minivan template provided in the ADAMS library to match the Ford Windstar experimental vehicle. However, prior to using the model it was necessary to ensure that the Windstar characteristics were correctly implemented in the ADAMS model. For this purpose, the following three tests of the model performance were completed.

2.3.1 Longitudinal Load Distribution

The model was first tested to ensure that it demonstrated the correct longitudinal weight distribution between the front and rear axles which have nominal values of 12.49 kN and 11.84 kN respectively and correct longitudinal load transfer while accelerating. This provides support for the longitudinal centre of mass placement as well as the height of the

centre of mass and models of the aerodynamic and rolling resistance forces. Under static conditions, the ADAMS model closely reproduced the axle weight distribution expected for the Windstar based on the centre of mass location obtained from Ford and the weight distribution measured on the research vehicle. The vehicle model was next run at several combinations of vehicle speed and constant acceleration level. The simulated results were compared with those obtained by hand calculations using the conventional calculations for longitudinal load transfer

$$W_f = \frac{l_2}{L}W - \frac{h}{L}(F - R_r) \quad (2.1)$$

and

$$W_r = \frac{l_1}{L}W + \frac{h}{L}(F - R_r) \quad (2.2)$$

where W_f and W_r are the front and rear axle loads respectively, l_1 and l_2 are the distances between the front and rear axles and the centre of mass respectively, L is the wheelbase, h is the height of the centre of mass, W is the vehicle weight, and $F - R_r$ is the net tractive force transmitted between the road and wheels, where F is the total tractive effort and R_r is the total rolling resistance force. Consideration of the longitudinal equation of motion for a vehicle allows the net tractive effort to be related to the sum of the inertial and aerodynamic resistances as follows

$$F - R_r = \frac{1}{2} \rho C_D A_f V_r^2 + \frac{W}{g}a$$

where ρ is the air density, C_D is the aerodynamic drag coefficient, A_f is the frontal area of the vehicle, V_r is the relative velocity between the vehicle and the wind, g is acceleration due to gravity, and a is the vehicle inertial acceleration. Using this, $F - R_r$ could be evaluated for any combination of vehicle acceleration and speed. The specific conditions tested as well as the simulation and calculation results are presented in Figure 2.6 using

Figure 2.6: Axles load distribution calculated manually and using ADAMS as a function of constant vehicle acceleration.

the following equations respectively:

$$W_{fd}\% = \frac{W_{fa} - W_{fc}}{W_{fa}} 100\% \quad (2.3)$$

and

$$W_{rd}\% = \frac{W_{ra} - W_{rc}}{W_{ra}} 100\% \quad (2.4)$$

where $W_{fd}\%$ and $W_{rd}\%$ are the percentage difference between the simulated and calculated load distributions on the front axle and rear axle respectively; W_{fa} and W_{ra} are the front and rear axle loads simulated by ADAMS and W_{fc} and W_{rc} are the front and rear axle loads calculated manually. From Equations 2.3 and 2.4 it is concluded that for different constant acceleration levels with an arbitrary selected constant longitudinal velocity, $W_{fd}\%$ and $W_{rd}\%$ didn't exceed 8.5%. The same process was repeated for a wide range of longitudinal velocities ranging from 40 to 120 km/h. Specific acceleration/velocity pairs considered for validation are identified in Figure 2.7. For each group of points at a particular speed, the longitudinal acceleration versus longitudinal axle load transfer can be plotted similarly to what was done in Figure 2.6. The conclusion that was extracted is

Figure 2.7: Longitudinal velocity and longitudinal acceleration data point pairs considered for longitudinal load transfer validation.

that the percentages $W_{fd}\%$ and $W_{rd}\%$ remain under 8.5%. It was found that the ADAMS model demonstrated longitudinal load transfer as expected though some differences existed between the calculated and simulated values. These differences are attributed to suspension and linkage effects present in the ADAMS model but not considered by conventional hand calculations. During acceleration, longitudinal load transfer results in ride height changes at the front and rear suspensions, which may alternatively be considered as a change of height at the centre of mass plus a change of pitch angle [19]. Vehicles, such as the Windstar, are designed to incorporate anti-lift for the front suspension and anti-squat for the rear suspension. Anti-lift is a suspension characteristic that uses acceleration-induced forces in the front suspension to reduce height increase of the front axle while anti-squat is a suspension characteristic that uses acceleration-induced forces in the rear suspension to reduce height decrease in the rear suspension. This causes the height of the centre of gravity for the axles to decrease or increase respectively and is consistent with the simulated result for which the ADAMS curve is higher than the manually calculated one in the case of the front axle and is lower for the rear axle as shown in Figure 2.6 and that the effect is almost negligible at low acceleration levels (less than 0.25 m/s^2).

2.3.2 Lateral Acceleration Gain

Next, the lateral acceleration gain and the curvature gain during constant steering manoeuvres were investigated as a function of vehicle speed. These tests are useful for investigating the transient handling and steady-state directional control response of the vehicle. The simulation tests began with initial conditions corresponding to the vehicle driving in a straight line at various constant speeds. Fixed steering angle inputs were commanded and the lateral acceleration levels and path curvatures of the simulated vehicle extracted. The results were used to determine the lateral acceleration gain [20] from

$$\text{Gain}_{\text{acc.}} = a_y / \delta_f$$

where a_y is the lateral acceleration measured in units of g and δ_f is the Ackerman steer angle of the front wheels in radians. This calculation was repeated for vehicle velocities in the range 0 to 120 km/h. The results were also predicted using the analytical expression [20]

$$\text{Gain}_{\text{acc.}} = V^2 / (gL + K_{us}V^2)$$

where V is the forward vehicle speed and K_{us} is the vehicle understeer coefficient that is a function of the front and rear tire cornering stiffnesses and the front and rear axle loads. The latter expression can be derived from the classical bicycle model of four-wheel vehicle handling. Figure 2.8 compares the simulated and predicted responses where smooth curves have been drawn through data points to highlight the trends. The results agree closely for vehicle speeds below 40 km/h beyond which they begin to diverge. This result is expected as the analytical expression for the lateral acceleration gain assumes a constant understeer coefficient irrespective of the lateral acceleration level. However, the understeer coefficient is known to change as the vehicle speed changes, and the corresponding lateral acceleration increases.

Figure 2.8: Lateral acceleration gain versus longitudinal velocity.

A similar comparison was performed for the curvature gain with the simulated curvature calculated from the expression

$$\text{Gain}_{\text{curv}} = \frac{1/R}{\delta_f}$$

where R is the radius of curvature extracted from the simulation and δ_f is the steer angle. The results were compared with predicted values for the curvature gain [20] calculated from

$$\text{Gain}_{\text{curv}} = \frac{1}{L + K_{us} V^2 / g}$$

Similar results were obtained that support the previous conclusion.

2.3.3 ABS Performance

The third test examined the effectiveness of ABS on reducing the longitudinal slip ratio. The ADAMS model, including the ABS control system, was used to simulate an open-loop braking scenario with and without the ABS system enabled. When full braking effort was applied without ABS the results showed that the longitudinal slip ratio reached 100% indicating wheel lock-up. When the same scenario was run with the ABS system enabled,

Figure 2.9: Effect of ABS in reducing longitudinal slip in the case of severe braking.

the longitudinal slip ratio decreased significantly and was limited to 4%. A comparison of the longitudinal slip response as a function of time for both conditions is shown in Figure 2.9.

2.3.4 Summary

While only very specific validation tests were performed, comparison of results from the ADAMS model of the Windstar with analytical calculations and verification that the simulated vehicle response was as expected, provided confidence in the model's ability to produce representative vehicle behaviour.

2.4 Representation of the ESC Controller

2.4.1 Simple Stability Augmentation Algorithm

To investigate the effect of stability augmentation system failure, it was necessary to implement a simple stability augmentation controller. This was accomplished by coding

a Fortran subroutine that is called by ADAMS. The resulting subroutine requires four inputs: time, steering wheel angle, vehicle speed, and yaw rate. It provides the front right, front left, rear right, and rear left brake line pressures as outputs. Control is achieved by determining the instantaneous desired yaw rate Ω_{zd} from the vehicle behaviour. The required moment about the centre of gravity then could be calculated using a proportional controller based on a function of the error between actual and desired yaw rate. This in turn leads to an equation for the output braking pressures. A detailed development follows.

The controller development begins with the classical handling equation [20]

$$\delta_f = \frac{L}{R} + K_{us} \frac{a_y}{g} \quad (2.5)$$

where

$$R = \frac{V}{\Omega_{zd}} \quad \text{and} \quad a_y = \frac{V^2}{R}.$$

Then,

$$\frac{a_y}{g} = \frac{\Omega_{zd} V}{g}.$$

Substituting into Equation 2.5, we obtain

$$\delta_f = \left(\frac{L}{V} \Omega_{zd} \right) + K_{us} \left(\frac{\Omega_{zd} V}{g} \right) \quad (2.6)$$

Then solving for Ω_{zd} we obtain

$$\Omega_{zd} = \frac{\delta_f}{\left(\frac{L}{V} \right) + K_{us} \left(\frac{V}{g} \right)} \quad (2.7)$$

where

$$\delta_f = \frac{\delta_{swa}}{\xi_s}$$

and δ_{swa} is the input steering wheel angle in radians and ξ_s is the steering gear ratio. Upon substitution, the final expression for Ω_{zd} is

$$\Omega_{zd} = \frac{\delta_{swa}}{((\frac{L}{V}) + K_{us}(\frac{V}{g}))\xi_s} \quad (2.8)$$

which is a function of the input steering wheel angle, the forward vehicle speed, g , and characteristics of the vehicle (K_{us} , ξ_s and L).

The moment M about the centre of gravity of the vehicle required to ensure that the actual and the desired vehicle yaw rates approach each other and could be expressed as

$$M = K_p(\Omega_{zd} - \Omega_{za}) \quad (2.9)$$

where Ω_{za} is the actual yaw rate input to the subroutine from the ADAMS simulation and K_p is an as-yet-undetermined proportional gain for the system.

As illustrated in Figure 2.10, the total moment results from the braking force developed at a single tire ground interface such that

$$M = F_b \frac{T_w}{2} \quad (2.10)$$

where F_b is the single-wheel braking force applied by the ground to the tire and T_w is the track width. Brake pressure P_b applied over the brake pad/drum effective contact area A_b , and at a radius r_b from the axis of rotation results in a wheel braking torque T_b such that

$$T_b = P_b A_b r_b \quad (2.11)$$

Figure 2.11 illustrates that F_b is related to braking torque by the wheel's rolling radius r_r

Figure 2.10: Moment due to the braking force.

Figure 2.11: Tire/ground braking force.

such that

$$T_b = F_b r_r \quad (2.12)$$

Combining Equations 2.11 and 2.12

$$F_b = P_b A_b \frac{r_b}{r_r} \quad (2.13)$$

Substituting the previous equation into Equation 2.10 leads to

$$M = P_b A_b \frac{r_b}{r_r} \frac{T_w}{2} \quad (2.14)$$

This can now be substituted into Equation 2.9

$$P_b A_b \frac{r_b}{r_r} \frac{T_w}{2} = K_p (\Omega_{zd} - \Omega_{za}) \quad (2.15)$$

This expression can now be solved for the brake pressure

$$P_b = \frac{2K_p r_r}{A_b r_b T_w} (\Omega_{zd} - \Omega_{za}) \quad (2.16)$$

A new coefficient K_{pa} can be defined for convenience as

$$K_{pa} = \frac{K_p r_r}{A_b r_b} \quad (2.17)$$

such that

$$P_b = \frac{K_{pa}}{\frac{T_w}{2}} (\Omega_{zd} - \Omega_{za}) \quad (2.18)$$

where K_{pa} is the composite ESC controller gain that must be determined. From the values of the actual yaw angular rate input Ω_{za} and the different between desired and

actual yaw rates ($\Omega_{zd} - \Omega_{za}$), the controller will evaluate the vehicle's directions (right or left turn) according to sign of the angular yaw rate and the stability conditions (oversteer or understeer) according to the sign of ($\Omega_{zd} - \Omega_{za}$). Then it will apply the braking pressure to the proper tire to control the vehicle's stability. In the case of oversteer the front tires will be controlled to regain stability where the rear slip angles are greater than the front ones. If the vehicle is in a left turn the front right tire will be controlled and the front left tire in the case of a right turn. In the case of understeer the rear tires will be controlled to regain stability where the front slip angles are greater than the rear ones. If the vehicle is in a left turn the rear left tire will be controlled and the rear right tire will be controlled in the case of a right turn.

2.4.2 Effectiveness of Stability Augmentation

To assess that the implemented stability augmentation controller achieves the desired objective of improving the vehicle stability, an aggressive double lane change manoeuvre was simulated. The vehicle began the manoeuvre in the middle lane having straight-line motion and a forward speed of 120 km/h. The vehicle then completed lane change into the right lane, followed immediately by a lane change back into the middle lane. The lateral distance between the centrelines of the lanes was 3.5 meters. The manoeuvre was simulated repeatedly to determine the optimum values of K_{us} and K_{pa} taking the significant factor for comparison to be the maximum lateral displacement of the vehicle centre of mass from the centreline of the middle lane from which the vehicle starts the simulation (Y_{dis}). Then the optimum values of K_{us} and K_{pa} were selected as the lowest points on Figures 2.12 and 2.13 respectively. The optimum values of K_{us} and K_p were selected to be 1.5 (1/rad) and 8.2 (Ns/m) respectively.

The manoeuvre was simulated for both the case where the controller was enabled and the case where it was disabled. Figures 2.14 and 2.15 show the reduction in the lateral dis-

Figure 2.12: Optimum value for K_{us} to minimize lateral displacement Y_{dis} .

Figure 2.13: Optimum value for K_{pa} to minimize lateral displacement Y_{dis} .

Figure 2.14: Comparison of vehicle lateral displacement response during a double lane change manoeuvre with and without active stability control enabled.

placement (Y_{dis}) and the yaw rate respectively. It should be noted that the lane width is 3.5 m which is the ideal target lateral centre of mass displacement during the lane change. Also Figures 2.16 and 2.17 show the changes in the steering wheel angle and the lateral acceleration due to intervention of the controller.

The previous simulation was repeated with different driving speeds 90 km/h and 150 km/h and the lateral displacement results are shown in Figures 2.18 and 2.19. All results show the enhancement in the lateral displacement due to activating the controller during the double lane change manoeuvre. By comparing the enhancements in lateral displacement for the three driving speeds, it is concluded that the improvements due to the controller are significant in a wide range of driving speeds not only for a moderate speed but also for a high speed for which the performance improvement is even greater. The sequence of animation frames included in Figures 2.20 and 2.21 show the second lane change for the case without active control and with active control respectively. The arrows indicate

Figure 2.15: Comparison of vehicle yaw rate response during a double lane change manoeuvre with and without active stability control enabled.

Figure 2.16: Comparison of vehicle steering wheel angle response during a double lane change manoeuvre with and without active stability control enabled.

Figure 2.17: Comparison of vehicle lateral acceleration response during a double lane change manoeuvre with and without active stability control enabled.

Figure 2.18: Comparison of vehicle lateral displacement response during a double lane change manoeuvre with and without active stability control enabled at vehicle longitudinal speed (90 km/h).

Figure 2.19: Comparison of vehicle yaw rate response during a double lane change manoeuvre with and without active stability control enabled at vehicle longitudinal speed (150 km/h).

the magnitude and direction of individual wheel loads. It is apparent that stability augmentation produces a desirable effect and it is inferred that the simple controller that was implemented can serve the intended purpose of providing preliminary indication of the potential hazard associated with sensor failure during a manoeuvre.

Figure 2.20: Animation frames for the vehicle without active stability control completing a double lane change manoeuvre.

Figure 2.21: Animation frames for the vehicle with active stability control completing a double lane change manoeuvre.

Chapter 3

Vehicle Experimental Testing

The previous verification and validation of the ADAMS vehicle model was based on comparison with simple analytical hand calculations. While these lend credibility to the model, some comparison with data from an actual vehicle and driver was considered beneficial to validate that the ADAMS built-in driver model is in fact representative of an actual driver. For this reason a small experimental investigation was undertaken.

For the experiment, a 2003 Ford Windstar LX minivan was used for performing a group of experimental tests in the field. The measured results were compared qualitatively to the simulated results obtained using the ADAMS software. Three manoeuvre scenarios were investigated: acceleration from constant throttle; deceleration from constant throttle; and a lane change. These manoeuvres were selected to exercise the vehicle's significant parameters by producing longitudinal acceleration within both acceleration and deceleration scenarios, and lateral acceleration within the lane change scenario.

3.1 Developing the Experimental Test Plan

In developing the test plan, the detailed scenarios were implemented in the way that the same scenarios could be run successfully in the computer simulation. A step by step pro-

cedure was customized to suite the driver behaviour. Safety was considered an important factor in designing the procedure to make sure that the driver could complete the manoeuvre safely. A few trials were run by the driver before the actual test run to make sure that the manoeuvres could be completed as planned, that the driver was comfortable, and that useful data would be collected.

3.1.1 Acceleration from Constant Throttle Manoeuvre

The acceleration from constant throttle manoeuvre consisted of the following steps.

1. The driver starts from rest accelerating until an arbitrary significant speed level is achieved by controlling the throttle pedal.
2. The driver holds the throttle pedal for few seconds to achieve the constant throttle level.
3. The driver moves the throttle pedal gradually (pushing) to achieve the required positive ramp increment in the throttle signal.

From the simulation, the constant longitudinal acceleration followed by the positive acceleration ramp was achieved.

3.1.2 Deceleration from Constant Throttle Manoeuvre

The deceleration from constant throttle manoeuvre consisted of the following steps.

1. The driver starts from rest, then accelerates to an arbitrary significant speed achieved by controlling the throttle pedal.
2. The driver holds the throttle pedal for few seconds to achieve the constant throttle level.

Figure 3.1: Track geometry for ISO standard lane-change manoeuvre[21].

3. The driver releases the throttle pedal gradually to achieve the required negative ramp increment in the throttle signal.

From the simulation, the constant longitudinal acceleration followed by the negative acceleration ramp (deceleration) was achieved.

3.1.3 Double Lane Change Manoeuvre

The double lane change manoeuvre consisted of the following steps.

1. The driver starts from rest, then accelerates to an arbitrary significant speed achieved by using the throttle pedal.
2. When the constant speed level is achieved, the driver starts the double lane change scenario according to the standard ISO specification of trajectory as shown in Figure 3.1, maintaining a constant speed during the manoeuvre.
3. The driver tries to keep the speed level as constant as possible during the manoeuvre. The lateral acceleration was measured during the manoeuvre.

Figure 3.2: Trajectory constructed on the experiment test field.

3.1.4 Setup of the Experiment Field

The experiment field was selected to be at Parking Lot 6 in Carleton University in the summer time. The double lane change trajectory was implemented in the designated area according to the ISO standard double lane change test. During the setup procedure, it was taken into consideration the efficient use of the available area. Safety cones, masking tape, and construction wire were used for setting up the trajectory as shown in Figure 3.2. An arbitrary reference point was selected and used as the GPS system datum so that a practical reference point was available in the parking lot.

3.2 Instrumentation

The experimental research vehicle was equipped with a variety of data collection instrumentation. Key data collection equipment will be described briefly.

3.2.1 Corsa Instrument System

The Corsa equipment is a complete system of software and hardware capable of collecting various types of inputs, recording the data, and then saving them in a familiar format after the experiment is finished ready for subsequent post-processing.

Corsa Hardware components

The Corsa system consists of the following components.

- Data acquisition box, also known as the data logger/sensor box, equipped with builtin accelerometers [22].
- Driver control box (DCB).
- RPM/speed interface box.
- Throttle, brake, and steering wheel angle sensors.
- Set of cables : 2-serial cables and driver control box cable.

Corsa Software Components

The Corsa instrument data acquisition system comes with PC compatible software, which enables communication between the PC and the data acquisition system, retrieval of stored or real time data from the data acquisition box, and the ability to screen the data to ensure that it falls within preset parameters by configuring the data acquisition box. Recorded

data could be displayed, manipulated, sent to proper database software tools, and printed easily using the Corsa software.

Setting Up the Corsa Instrument

The data acquisition box has two accelerometers which are used to measure both longitudinal and lateral acceleration oriented at right angles to each other. The box receives power from either a 12V motorcycle battery or small gel-cell battery. The box is connected to various hardware components with the suitable cable for each component and the connection is as follows. The box is connected to the 'DCB' using a 15 pin (high density DB15) cable attached to the terminal labeled 'DCB', then it is connected to the PC by a serial interface cable at the terminal labeled 'serial'. Then it is also connected to the RPM interface box by a 9 pin (DB9) cable to the terminal labeled 'speed in' and finally the analog sensors (throttle, brake, and steering wheel angle) are connected at the terminal labeled 'analog A'.

The wheel speed sensor is connected to the RPM/speed interface box by a 9 pin (DB9) cable which is in turn wired to the 'speed in' port on the face of the data acquisition system. The DCB has an on/off switch in addition to a stop/run switch. Figure 3.3 shows the set up connections for the Corsa instrument (laptop, DCB box, and RPM/speed interface box) and speed sensor cable (DB9) which connect to the speed interface box. Figure 3.4 shows the data acquisition control box with the proper setup to both the DCB box and the speed interface box. Figure 3.5 shows the setup for the various CORSA sensors (throttle, braking, and steering wheel angle) that are connected to the data acquisition control box. Figure 3.6 shows the set up for the wheel speed sensor and its cable (DB9) to the speed interface box.

Figure 3.3: Corsa instrument components and setup.

Figure 3.4: Data acquisition control box with the input and output connections.

Figure 3.5: Throttle, braking, and steering wheel angle sensor setup.

Figure 3.6: Wheel speed sensor.

3.2.2 Leica GPS Instrument

Leica GPS System 500 Components

The Leica GPS system 500 comprises the GPS receiver hardware and PC based software for GPS surveying and related applications [23]. The main components are:

- GPS receiver which receives the satellite signals (antenna and sensor);
- GPS terminal keyboard and display device to interface with the receiver; and
- post-processing software used to process GPS data.

Leica GPS System 500 setup

For surveying the pre-set trajectory, a pair of System 500 GPS units was needed. One serves as a reference over the arbitrary fixed reference point and the other serves as a roving surveyor for the trajectory. Each unit consists of an SR model sensor connected to the antenna using the antenna cable. The GPS TR500 terminal (keyboard and display device) was needed to start the operation easily, manipulate the sensor parameters, and check the system status. Two standard camcorder batteries were needed as a source of power for the sensor and were plugged into the back side of the sensor. The TR500 could be either attached to the sensor directly or through a connection cable that plugged into the TERMINAL port on the sensor.

For achieving successful surveys, the GPS system requires undisturbed satellite signal reception by setting up the GPS sensor in locations which are free of obstructions. The line between the GPS antenna and GPS satellites should not be blocked by buildings or trees, especially for the sensor which serves as the reference unit.

The reference GPS unit is considered as a static surveyor so that the antenna must be kept perfectly steady at all time. The antenna was mounted on a tripod after centring

Figure 3.7: Set up of the GPS sensor, TR500 terminal and antenna over the tripod.

and leveling it precisely to ensure stability. The sensor controller was hung on one of the tripod legs as shown in Figure 3.7.

The second GPS unit is considered as a roving surveyor which is able to provide the trajectory of the moving antenna. The antenna was fixed at the top of the vehicle using a fixture and it had a magnetic base to ensure stability during the manoeuvre. The sensor and the terminal were fixed by a secure set up inside the vehicle as shown in Figure 3.9. The System TR500 is able to give the resultant relative trajectory by knowing and occupying the fixed reference point. The arbitrary reference point was selected at reasonable distance (less than 15 m) from the trajectory and in an unobstructed area. The relative trajectory was followed and the points were registered quickly and efficiently. Figure 3.9 shows the GPS setup over the reference fixed point. The GPS output result for the trajectory is shown in Figure 3.10. Figure 3.11 shows the trajectory plotted using CAD software.

Figure 3.8: GPS setup inside the vehicle.

Figure 3.9: GPS over the fixed point.

Figure 3.10: Trajectory path extracted from the GPS receiver.

Figure 3.11: Trajectory path extracted from the GPS receiver and implemented by CAD software tools (scale = 1 : 15.41 m).

3.3 Developing the Simulation Test Plan

The experimental test was conducted with one driver under ideal conditions. Roadway conditions consisted of dry asphalt. The manoeuvre was repeated three times for each test type to ensure that the process and the results were satisfactory. Due to the limitation of space in Parking Lot 6 and taking into consideration safety as a priority factor, the speed was limited to 30 km/h. The experiments were simulated by customizing the road test scenarios using ADAMS TeimOrbit File Format [24]. It consists of a structured file format containing various blocks of information used to describe attributes, such as file version, units data, and data description. The TeimOrbit file format is an ASCII format, which is used in MSC.ADAMS applications to write and customize the text file named the driver control files (.dcf). ADAMS chassis through one of its facilities within the drive handling link uses driver control files to describe the manoeuvres that the Driving Machine performs. The Driving Machine drives the virtual vehicle according to its instructions much like a test driver would drive an actual vehicle. The Driving Machine can input a constant throttle signal followed by a step or ramp signal, it could be used also for generating steering, braking, gear change, and clutch signal inputs to the virtual vehicle. Driver control files essentially describe how the vehicle will be driven during the experiment using the Driving Machine. Driver control files organize complex experiments into a set of smaller, simpler steps called mini-manoevres. For implementing the acceleration and the deceleration from constant throttle manoeuvres, two mini-manoevres were linked for constant then ramp throttle scenarios at different levels of throttle (same levels obtained from the experimental tests). The double lane change scenario was simulated using the change lane mini-manoevre after customizing the trajectory to match the trajectory that was set up on the experiment field and the vehicle speed. The throttle is controlled by the automated driver to maintain constant speed during the event. The steering wheel angle

Figure 3.12: Trajectory view during the lane change lane manoeuvre.

was applied to match the trajectory. The actual driver uses more sophisticated control thereby resulting in some slight differences in the simulated and measured trajectories. Figure 3.12 shows the trajectory view during the on-field double lane change experiment.

3.4 Post Processing the Results

The results extracted from the experimental test and the simulated tests were post-processed and the comparisons were plotted.

3.4.1 Acceleration from Constant Throttle Manoeuvre

The results for both the experimental and simulated tests were plotted in Figures 3.13 and 3.14. They show that the constant throttle level was achieved at 15%. The results were matched specially after the throttle signal stabilized after 4 sec and the longitudinal

Figure 3.13: Input throttle (15%) versus time for the experimental and simulated tests.

acceleration difference between the experimental and the simulated result was less than 0.1 g. Figure 3.15 shows the longitudinal acceleration after the throttle signal stabilized with overshooting excluded.

3.4.2 Deceleration from Constant Throttle Manoeuvre

The results were plotted for both the experimental and the simulated tests and as shown in Figure 3.16 by having the throttle level at 45.7% the results match after $t = 1.3$ sec. In Figure 3.17 it is shown that the difference between the longitudinal acceleration for the experimental result and that from the simulated one never exceed 0.15g after the stable level of throttle is achieved.

Figure 3.14: Longitudinal acceleration versus time for the experimental and simulated tests.

Figure 3.15: Longitudinal acceleration versus time for the experimental and simulated tests after 4 sec.

Figure 3.16: Input throttle (45.7%) versus time for the experimental and simulated tests.

Figure 3.17: Longitudinal acceleration versus time for the experimental and simulated tests.

3.4.3 Lane Change Manoeuvre

The simulations were run for two different speed levels: 20 km/h and 30 km/h. Figures 3.18 and 3.19 represent the steering wheel angle and the lateral accelerations respectively at a speed level of 20 km/h. In the plotted figures the transient period from both signals is excluded (during which the driver usually accelerated before starting the double lane change trajectory) and the starting point were matched. The differences in the amplitudes for the simulated tests and the experimental ones were due to the driver response factor which is different in the actual driver than the driver controller in the simulated tests. In the experiment, the driver manages the lane change smoothly and not aggressively in order to manage the trajectory safely. At a speed of 20 km/h both tests reached the end of the lane change at the same time, 17.75 sec. At the higher driving speed, the driver response was affected. Lateral accelerations in both speed levels oscillate due to the suspension movement during the test. As shown in Figures 3.20 and 3.21, the steering wheel angle and the lateral accelerations respectively were plotted at the speed level of 30 km/h.

3.5 Summary

For the acceleration from constant throttle test the longitudinal acceleration difference between the experimental and the simulated result was less than 0.1 g and didn't exceed 0.15 g for the deceleration from constant throttle. For both the simulated and experimental tests during the double lane change manoeuvre the driver was able to complete the right lane change over similar time intervals. The difference in the amplitude is due to the difference between the automated driver model and the actual driver. The driver was successfully able to manage the trajectory but the actual driver was actually under some normal stress due to having to complete the test following the procedure required. In addition to the fact that the limited experiment area in Parking Lot 6 compared to the

Figure 3.18: Input steering wheel angle versus time for the experimental and simulated tests at a speed level of 20 km/h.

actual highway increases focus on safety factors while driving the vehicle.

Figure 3.19: Lateral acceleration versus time for the experimented and simulated tests at a speed level of 20 km/h.

Figure 3.20: Steering wheel angle versus time for the experimental and simulated tests at a speed level of 30 km/h.

Figure 3.21: Lateral acceleration versus time for for the experimental and simulated tests at a speed level of 30 km/h.

Chapter 4

Effect of Sensor Failure

ESC controllers, including the representative controller developed in Chapter 2, rely on a variety of input sensors including vehicle speed, steering wheel angle, and yaw rate for calculating appropriate vehicle control signals. In the event of sensor failure, assuming the stability system is not disabled or a backup system is not immediately invoked, erroneous input signals would be used in calculating the control signal which would lead to improper behaviour. A preliminary failure study was completed for the longitudinal velocity sensor during the double lane change manoeuvre for a vehicle speed of 120 km/h. The failure study was then expanded to various input control sensors (longitudinal velocity, yaw rate, and steering wheel angle) at different values of longitudinal velocity (90, 120, and 150 km/h) for the double lane change manoeuvre.

4.1 Sensor Failures and Defects

Several types of failure modes could affect the ESC sensory system which could lead to undesired responses. These modes could be related to defects affecting the internal physical parts, the harnesses and connectors, or interference resulting in undesired signal noise.

- The first type of failure is related to physical components. It includes defects, deterioration, incomplete contact between components, dust or water contamination, and inaccurate sensor adjustment resulting from poor maintenance (gaps and clearances).
- The second type of failure is related to the harness and the connectors. It is a result of total discontinuity in the internal wiring partial discontinuity, or unstable connections (on/off). These defects are due to vibration of the harness over extended periods of time, undesired harness bending, exceeding the useful cable life, and dust or water contamination.
- The third type of failure is the interference from undesired noise signals resulting from improper vibration of loose parts, moving close to an external source of noise, or failure of other electromechanical components resulting in electromagnetic interference (EMI).

4.2 Sensor Failure Modes

Several types of failure modes could affect the ESC sensory system leading to an undesired response. They are categorized as follows.

- Sudden shutoff mode: this mode represents a complete sensor defect, total discontinuity in the harness, or separation between contacting parts.
- On/off failure mode: this mode represents dust or water contamination, exceeding the useful cable life, and unstable connection of the harness.
- Sensor degraded performance mode: this mode represents deterioration, exceeding the useful cable life, incomplete contact between parts, and partial discontinuity of the harness.

Figure 4.1: Shutoff failure mode starting at time = 5.0 sec.

- Noise interference mode: this mode represents the effects of undesired noise signals or an external source of noise.

4.3 Modelling of Failures

Mathematical modelling of failure modes that affect the sensors can be achieved by multiplying the failure mode signal by the input signal as described in the following sections.

4.3.1 Sudden Shutoff Mode

This failure mode, representative of a complete sensor electrical failure, results in the sensor signal becoming zero immediately following the failure as shown in Figure 4.1.

Figure 4.2: On/off failure mode with $H = 1.8$ sec.

4.3.2 On/Off Failure Mode

In this case the input signal becomes zero for particular intervals of time but nominal values exist at the other intervals as shown in Figure 4.2. The following formula is used

$$F = \cos(\omega t)$$

where $\omega = \frac{2\pi}{H}$, F is the failure mode value, H is the periodic time in sec, and t is the time in sec. A subsequent step replaces the negative values with the single value of zero and replaces the positive values with the single value of one.

4.3.3 Sensor Degraded Performance Mode

This is similar to the on/off mode which leads an input signal to become zero but the failure occurs gradually and cyclically, representing the wrong input data to the ESC system. As a result, unexpected output signals will be sent to the actuators, resulting in

Figure 4.3: Sensor degraded performance mode with $A = 50$.

an undesired response potentially leading to instability. The formula used for evaluating the sensor signal effect is

$$K = A(1 + \cos(\omega t))$$

where A is the amplitude of the failure effect and K is the degraded performance failure mode as shown in Figure 4.3.

4.3.4 Noise Interference Mode

This mode is modelled by a failure which leads to corruption of the input signal as shown in Figure 4.5. The formula used for evaluating the sensor signal effect is

$$N = \sum S_i \tag{4.1}$$

where N is the noise signal, i is the number of an individual noise signal component, and S_i is the noise signal component i . The noise signal component is calculated from the

Figure 4.4: Individual signals that affect the sensor with various amplitudes (50, 100, 150).

Figure 4.5: Resultant noise of the individual signal.

following formula:

$$S_i = A_i \cos(\omega_i t)$$

where A_i is the amplitude of individual noise signal i and ω_i is the frequency of the i^{th} noise signal component. Figure 4.4 and Figure 4.5 represent the individual signals and the resultant noise that affects the sensor for the case where three noise components are included.

4.4 Preliminary Failure Study Results

A preliminary study was implemented by repeating the double lane change manoeuvre after customizing the Fortran routine to represent the several modes of the sensor failures. The simulation was run at a longitudinal speed of 120 km/h then a loss of the longitudinal velocity signal (sudden shutoff failure mode) was simulated to occur at a range of times during the manoeuvre. The results plotted in Figure 4.6, represent the lateral displacement increase. This increase was significant after the failure occurred at 3.0 sec and exceeds 4.0 m into the right lane. The vehicle experiences undesirable diagonal movement across the left lane particularly after 7.0 sec. In Figure 4.7, the lateral acceleration increase due to the failure mode and its maximum value at the left lane exceeded twice the maximum value in the case of the sensory system functioning properly. In Figure 4.8, results show the effects of the failure mode on the yaw rate compared with the events when the sensor functioned properly.

The simulation results were post-processed to determine the maximum lateral displacement that would have occurred in the events when the sudden shutoff mode occurs at different time periods ranging from $t = 1.0$ sec to 6.0 sec. The results are shown in Figure 4.9 and the maximum lateral displacement is observed to occur for $t = 1.8$ sec. The previous procedures were repeated to simulate the on/off failure mode that affects the longitudinal

Figure 4.6: Effect of the sudden shutoff failure mode on system sensory input at 3.0 seconds during the double lane change manoeuvre on vehicle lateral displacement.

Figure 4.7: Effect of the sudden shutoff failure mode on system sensory input at 3.0 seconds during the double lane change manoeuvre on vehicle lateral acceleration.

Figure 4.8: Effect of the sudden shutoff failure mode on system sensory input at 3.0 seconds during the double lane change manoeuvre on vehicle yaw rate.

Figure 4.9: Maximum lateral displacement with various sudden shutoff mode occur.

Figure 4.10: Effect of the on/off failure mode on the control system sensory input during a double lane change manoeuvre on vehicle lateral displacement as a function of the time the failure occurred ($H = 3.0$ sec).

velocity input signal. The results are shown in Figures 4.10 and 4.11 which show the lateral displacement and lateral acceleration resulting from the failure mode compared to the case when the sensor functioned properly. Additional simulation cases were run and the results were post-processed to determine the maximum lateral displacement that would have occurred for the on/off failure mode, changing the value of H from 1 sec to 6 sec with increments of 1 sec and substituting in the failure mode formula to investigate the different on/off failure mode effects upon the input signal. The results are represented by Figure 4.12 and the maximum lateral displacement occurred for $H = 5.0$ sec. To represent the effect of the degraded performance failure mode on the longitudinal velocity input signal, simulations were run using that failure mode formula. The results are shown in Figures 4.13 and 4.14. They represent the lateral displacement and lateral acceleration that were affected by the degraded performance failure mode. This can be compared to the

Figure 4.11: Effect of the on/off failure mode on the control system sensory input during a double lane change manoeuvre on vehicle lateral acceleration as a function of the time the failure occurred ($H = 3.0$ sec).

Figure 4.12: Maximum lateral displacement at different time periods for the on/off failure mode.

Figure 4.13: Effect of the degraded performance failure mode on the control system sensory input during a double lane change manoeuvre on vehicle lateral displacement as a function of time the failure occurred ($H = 3.0$ sec and $A = 50$).

event when the sensor remained functional at $H = 3$ sec. Post-processing of the simulation results and changing the value of H from 1 sec to 6 sec, the maximum lateral displacement that would have occurred was calculated. Results are presented in Figure 4.15 and the maximum lateral displacement is found to occur at $H = 3.25$ sec. Though the results appear relatively insensitive to H .

Figure 4.16 shows the longitudinal speed behaviour when various failure modes affect the input longitudinal velocity compared to when the sensor is working properly. It should be noted that the simple representative ESC controller implemented for this work does not include engine management features thereby leading to the increasing velocity following the manoeuvre. The noise interference mode that affects the input signal was simulated using Equation 4.1 substituting the arbitrary values of the amplitudes A_1 , A_2 , and A_3 as 50, 100, and 150 respectively and the periodic times as 1.5, 5.0, and 3.25 sec respectively which

Figure 4.14: Effect of the degraded performance failure mode on the control system sensory input taking into account double lane change manoeuvre on vehicle lateral acceleration as a function of time the failure occurred ($H = 3.0$ sec and $A = 50$).

Figure 4.15: Maximum lateral displacement at different time periods for the degraded performance failure mode.

Figure 4.16: Effect of various failure modes on the longitudinal speed.

are the values corresponding to the maximum lateral displacement for the previous failure modes. The results represented by Figures 4.17 and 4.18 show the lateral displacement and the lateral acceleration when this noise signal interferes and is premultiplying the input sensor signal. Figures 4.19 and 4.20 show the effect of the interference when the noise signal is simply added to the input sensor signal.

4.5 Failure Study Results

After the preliminary study for the failure modes considering only the longitudinal speed sensor, a brief study was implemented including the additional speeds of 90 km/h and 150 km/h to span a full range of highway speeds for lane change scenarios. Simulations were also run for two more groups considering a failure of either the steering wheel angle sensor or the yaw rate sensor.

Figure 4.17: Effect of multiplying the noise interference failure mode on the control system sensory input taking into account a double lane change manoeuvre on vehicle lateral displacement as a function of time.

Figure 4.18: Effect of multiplying noise interference failure mode on the control system sensory input taking into account a double lane change manoeuvre on vehicle lateral acceleration as a function of time.

Figure 4.19: Effect of adding the noise interference failure mode on the control system sensory input taking into account a double lane change manoeuvre on vehicle lateral displacement as a function of time.

Figure 4.20: Effect of adding the noise interference failure mode on the control system sensory input taking into account double lane change manoeuvre on vehicle lateral acceleration as a function of time.

Figure 4.21: Maximum lateral displacement versus sudden shutoff failure mode start time at various speed levels.

4.5.1 Failure of the Longitudinal Speed Sensor

When affecting the input signal of the longitudinal speed sensor by the sudden shutoff mode across the three velocity levels 90, 120, and 150 km/h, the results plotted in Figure 4.21 represent the results that were extracted for the maximum lateral displacements at the various speed levels. Applying the on/off failure mode and the degraded performance mode that affect the input signal for the different speed levels, the results for the maximum lateral displacement are shown in Figures 4.22 and 4.23 respectively.

The interference noise signals were both multiplied and added to the input sensory signal for different speed levels and the results are represented by Figures 4.24 and 4.25 respectively.

Figure 4.22: Maximum lateral displacement versus on/off failure mode time period at various speed levels.

Figure 4.23: Maximum lateral displacement versus degraded performance failure mode time period at various speed levels.

Figure 4.24: Maximum lateral displacement versus the multiplicative noise interference failure mode time period at various speed levels.

Figure 4.25: Maximum lateral displacement versus the additive noise interference failure mode time period at various speed levels.

4.5.2 Failure of the Steering Wheel Angle Sensor

From Equation 2.7 and dividing the numerator and the denominator by the longitudinal velocity V the equation becomes.

$$\Omega_{zd} = \frac{\delta_{swa} V}{((L) + K_{us}(\frac{V^2}{g}))\xi_s} \quad (4.2)$$

From this, it is observed that the longitudinal speed and the steering wheel angle variables are present and multiply each other in the numerator. This leads to the fact that the result will be the same if the input longitudinal speed equals zero or the input steering wheel angle equals zero or both equal zero. Accordingly, the same results were achieved when considering the cases of the sudden shutoff and on/off steering sensor failures as when considering velocity sensor failures. Because the degraded performance failure mode has the amplitude A added at all times, it does not have the value of zero except at certain specific times, so that the output results are different and the mode could be simulated iteratively to find the most significant effect for the value $A = 0.5$. Comparison was made for the steering wheel angle signals that were affected by the degraded performance failure mode for various speed levels and the results are represented in Figure 4.26 which represents very high values for the signals compared to the case when the sensor works properly shown in Figure 2.16. The noise interference mode that affects the input steering wheel angle signal was simulated using Equation 4.1 substituting the arbitrary values of the amplitudes A_1 , A_2 , and A_3 as 0.3, 0.5, and 0.7 respectively. The noise signal is both multiplied and added to the input sensory signal for different speed levels and the results are presented in Figures 4.27 and 4.28 respectively.

Figure 4.26: Input steering wheel angle as affected by degraded performance failure mode versus time at time period = 3sec for various speed levels.

Figure 4.27: Effect of multiplying the noise interference failure mode signal by the steering wheel angle input signal taking into account a double lane change manoeuvre on vehicle lateral displacement as a function of time.

Figure 4.28: Effect of adding the noise interference failure mode signal by the steering wheel angle input signal taking into account a double lane change manoeuvre on vehicle lateral displacement as a function of time.

4.5.3 Failure of the Yaw Rate Sensor

Referring to Equation 2.11 and considering the fact that the yaw rate input signal Ω_{za} has a zero value when affected by either the sudden failure mode, on/off failure mode or degraded performance failure mode, as soon as $\Omega_{za} = 0$ the moment will increase to higher values that cause the mechanical parts to experience large deflections and as a result the simulation no longer ends successfully. This can be interpreted loosely as indications hazardous vehicle behaviour. The noise interference mode affecting the input yaw rate signal was successfully simulated to study the effect of either multiplying or adding the noise signal to the input sensory signal for different speed levels. Using Equation 4.1 and substituting the arbitrary dimensionless values of the amplitudes A_1 , A_2 , and A_3 as 50, 100, and 150 respectively. The results are represented by the Figures 4.29 and 4.30 respectively.

Figure 4.29: Maximum lateral displacement versus noise interference failure mode time period at various speed levels when the noise signal multiplied the yaw rate signal.

Figure 4.30: Maximum lateral displacement versus noise interference failure mode time period at various speed levels when the noise signal added to the yaw rate signal.

Table 4.1: Simulaton lateral displacement results [m] sorted by the type of sensor.

		Failure types					
		Sensor work	Sudden off	On/off	Degraded	Noise add	Noise mult
Speeds (km/h)	Sensor type						
90	Longitudinal velocity	4.05	5.18	8.68	5.2	6.52	5.13
120		3.77	5.58	9.95	5.49	5.87	5.59
150		3.37	5.64	9.29	5.49	4.61	5.52
90	Steering wheel angle	4.05	5.18	8.68	6.75	3.79	5.02
120		3.77	5.58	9.95	7.07	3.55	10.4
150		3.37	5.64	9.29	7.41	3.14	10.7
90	Yaw rate	4.05	N/A	N/A	N/A	5.97	5.14
120		3.77	N/A	N/A	N/A	6.23	6.75
150		3.37	N/A	N/A	N/A	3.1	5.43

4.6 Analysis

Table 4.1 shows the results sorted by the type of sensor and Table 4.2 shows the results sorted by the various longitudinal velocities. The significant results from the preliminary study were categorized into three groups each representing the effect of various failure modes for each of the input sensors. The results were plotted for different longitudinal speed levels to allow comparisons which assist in screening the vehicle behaviour. The results in the first group which represent the failure in the longitudinal speed sensor were plotted.

- In the case of the sudden shutoff failure mode occurring, from Figure 4.21 it is observed that when failure occurs within the first 3 sec after initiating the manoeuvre the maximum lateral displacements have higher values for each speed level and that continues until they reach their peak values which are varying from 5.2 m to 5.7 m. The higher the speed level from 90 km/h to 150 km/h the higher the maximum lateral displacements. Within the interval 3 sec to 4.5 sec when the vehicle starts to change lanes at time = 3 sec the speed levels change their orders. The reason for

Table 4.2: Simulaton lateral displacement results [m] results sorted by longitudinal velocity.

		Sensor work	Failure types				
			Sudden off	On/off	Degraded	Noise_add	Noise_mult
Speeds (km/h)	Sensor type						
90	Longitudinal velocity	4.05	5.18	8.68	5.2	6.52	5.13
	Steering wheel angle	4.05	5.18	8.68	6.75	3.79	5.02
	Yaw rate	4.05	N/A	N/A	N/A	5.97	5.14
120	Longitudinal velocity	3.77	5.58	9.95	5.49	5.87	5.59
	Steering wheel angle	3.77	5.58	9.95	7.07	3.55	10.4
	Yaw rate	3.77	N/A	N/A	N/A	6.23	6.75
150	Longitudinal velocity	3.37	5.64	9.29	5.49	4.61	5.52
	Steering wheel angle	3.37	5.64	9.29	7.41	3.14	10.7
	Yaw rate	3.37	N/A	N/A	N/A	3.1	5.43

this behaviour is due to completing this part of the trajectory faster in the case of driving with the higher speed. As a result, the diagonal movement toward the left lane at the moderate speed or the swinging effect at high speed occurs as shown in Figures 4.6 and 4.31 where the vehicle, instead of returning to the middle lane, starts to swing too far into the left lane with lateral displacement close to that of the right lane excursion then swings to the right lane again for the second time which causes hazardous driving.

After 4.5 sec the maximum lateral displacements have lower values for each speed level until the simulation ends. That is because the failure occurred after the vehicle returned back from the right lane.

- In the case of on/off failure the results were plotted in Figure 4.22 and indicates that as the time period H increases the maximum lateral displacement increases for each speed level. This is due to the fact that the failure mode signal has the value of zero for longer periods of time when H has high values. The higher the speed level the

Figure 4.31: Swing effect as a result of the sudden shutoff failure mode.

earlier the peak value of the maximum lateral displacement occurs, so the peak value for high speed occurs at a lower value of H than for of the moderate or slow speeds.

- When the degraded performance failure mode affects the input signal as shown in Figure 4.23 and due to the fact that the duration for the signal to have the value of zero is small, the peak point of the different levels of speed are achieved at moderate values of H . The reason for this behaviour could be explained by knowing that at small H the failure signal is not significant and at large H longer time was needed for the signal to reach the zero value which is longer than the time needed for completion of the lane change.
- Results from affecting (multiplying or adding) the input signal with various noise interference signals for different speed levels presented in Figures 4.24 and 4.25 respectively. In the case of a multiplicative noise signal the peak value of the maximum lateral displacement occurs earlier at higher speed than those for the moderate and

low speed cases. In the case the noise signal was added, the peak value for the maximum lateral displacement was higher for lower speed than for moderate and high speed due to the longer period of time that the input signal was exposed to the noise signal.

The results for the second group which represent the failure in the steering wheel angle sensor are the same as that of first group for the sudden shutoff failure mode and the on/off failure mode but for the degraded performance mode it is concluded from the graphs that the higher the speed level the larger the swing in the left lane (magnitude and duration). When multiplying the noise signal by the input signal, the higher the speed level the larger the maximum lateral displacement. In the case of adding the input signal and the noise signal the higher the speed level the earlier the peak of the maximum lateral displacement takes place, but the peak value is the lowest as the duration for the input signal to be affected by the noise signal is the smallest of the three speed levels. The results for the third group which represents failure in the yaw rate sensor reflect the high moment that affects the vehicle causing instability. The results due to the effect of the noise signal which could be multiplied or added by the input signal are plotted in Figures 4.29 and 4.30 respectively. For both multiplicative and additive noise signals, the higher the speed level the earlier the maximum lateral displacement peak value occurs. However, the peak value at 120 km/h is higher than the other levels of speed. Undesired vehicle behaviour in terms of both the diagonal movement and the swing effect occurs for both types of noise.

4.7 Summary

It is concluded that affecting the input sensory system signals by various failure modes leads to undesired situations and causes degraded performance and potentially unsafe driving. The vehicle is affected significantly when the input signals (longitudinal velocity,

steering wheel angle, or yaw rate) are affected by the failure mode signal. Particularly, the noise interference mode, either when adding or multiplying the signal, and degrades performance. As a result of the sensor failures, the vehicle experiences large lane change excursions with the magnitude depending both on the vehicle speed and the time period over which the ESC controller is exposed to erroneous input signals. The mechanical parts of the vehicle's chassis experienced large deflection which could lead to system failure and potentially catastrophic results.

Chapter 5

Discussion and Conclusion

This project has achieved the following objectives.

- A vehicle model was developed using ADAMS software tools to represent the Ford Windstar experimental vehicle model.
- Computational validation was done to confirm validity of the vehicle model.
- Fortran code was implemented to represent a generic ESC system; and its function was demonstrated to improve vehicle performance in terms of directional vehicle control.
- An experimental test was conducted using the Ford Windstar as an experimental vehicle to validate the automated driver model present in the ADAMS-chassis model compared to an actual driver reasonably well.
- The Fortran code was customized to investigate the effect of different failure modes over various sensors used by typical ESC system.

Of particular interest, data was collected from available literature concerning fatality statistics due to vehicle loss of control and the statistics focused on the effect of safety system innovations specifically for passive safety systems and the benefits gained by vehicles

equipped with an electronic stability control system. A complete ADAMS model was built using customized templates from the ADAMS library to match the Ford Windstar experimental vehicle.

The validation process was undertaken to gain confidence in modelling activity when upgrading the model by implementing the control system. The mathematical model was used to complete three tests of the model performance. First, for longitudinal weight distribution it was concluded that $W_{fd}\%$ and $W_{rd}\%$ didn't exceed 8.5% for different constant acceleration levels with an arbitrarily selected constant longitudinal velocity. The test was repeated for a wide range of longitudinal velocities spanning the range 40 to 120 km/h and the results remained under 8.5%. Difference between the mathematical and the actual models were due to the suspension and the linkage effects represented in the ADAMS model and not represented in the mathematical model due to the completeness of the model implemented and simulated using ADAMS chassis. For the lateral acceleration gain test for vehicle speeds below 40 km/h the results agree closely. Beyond 40 km/h results change their behaviour due to the fact that the analytical expression for the lateral acceleration gain assumes a constant understeer coefficient irrespective of the lateral acceleration level. From the ABS performance test it was concluded that the longitudinal slip ratio reduced significantly and was limited to 4%. Accordingly confidence was gained in the ADAMS model in terms of the implemented control system subroutine. Fortran code was implemented to represent the ESC and the optimum values for the main factors were calculated in order to achieve the most efficient controller. The significant parameter that was selected for evaluating the stability regained during the manoeuvre was the maximum lateral displacement during the right lane change within the double lane change manoeuvre. The vehicle stability was proven and significant results were obtained when using the Fortran ESC code.

The Fortran code was customized to investigate the various failure modes that affect the

different sensors available on the ESC-equipped vehicle (longitudinal velocity, steering wheel angle, and yaw rate sensors). The failure modes reflect both the internal physical failures and the failure that could occur to the harnesses and the connectors. The failure modes investigated were the sudden shut off mode, on/off failure mode, degraded failure mode, and the noise interference mode. A preliminary failure study was done for the failure modes that affect the longitudinal velocity sensor at a speed of 120 km/h during the double lane change manoeuvre. The effect of the sensor failure was significant in all the results extracted and as expected the increase in the maximum vehicle lateral displacements was achieved which reflects the severe situation that the driver could experience on the road. A study was done to investigate the effect of the failure modes for different working speeds for the various sensors. The results reflect the severe situations which the driver could experience which vary from diagonal vehicle movement to the swing effect into the left lane specially at a higher speed 150 km/h when the longitudinal velocity input sensor is affected by the failure modes. It is concluded that as the speed increases the maximum lateral displacement peak value will occur earlier than at lower speeds and that due to the vehicle handling for the trajectory is much faster. In the case of the failure modes affecting the steering wheel angle sensor the most significant effect is the swinging effect that will occur with a high lateral displacement amplitude which will negatively affect the suspension linkage and the steering system in addition to the hazardous conditions on the road. The failure modes that affect the yaw angle sensor result in a high value of the moment that affects the various chassis parts and the vehicle mechanism which could lead as a consequence to vehicle damage and loss of control.

Experimental testing was conducted to validate that the automated driver model is representative of actual driver behaviour. The experiment was setup in Carleton University Parking Lot 6 using For Windstar experimental vehicle and data collection instruments including the data acquisition system (Corsa) and the GPS equipment (Leica). Three

main tests were run: the acceleration from constant throttle; the deceleration from constant throttle; and the double lane change. These manoeuvres were selected to exercise the vehicle's significant parameters. From the results it is concluded that the longitudinal acceleration difference between the experimental and the simulated result was less than 0.1 g for the acceleration from constant throttle and didn't exceed 0.15 g for the deceleration from constant throttle. For the double lane change test the right lane change ends for both simulated and experimental test simultaneously and the amplitude difference is due to the difference between the automated driver model compared to the actual driver. The actual driver is expected to be under normal stress while driving to manage the trajectory successfully and complete the test following the required procedure. In addition, the actual driver must be constantly aware of the safety implications associated with performing the tests within the confines of space available in the experimental area. Generally the results of the research reflect the objectives and accomplished the preliminary study of the effect of the ESC controller on the vehicle stability and the consequences of particular defects occurring in the control system.

The future work recommended can be summarized as follows. It is recommended that the experimental test be run by different drivers of different ages and different driving skills to obtain a wide comparison between their behaviours. It is also recommended for the test to be run for different road designs, various road conditions, both dry and wet. This would contribute to better understanding of real driver behaviour while ESC is in use. This is likely the most significant factor that affects his performance. A wide experimental location will contribute more to the research specially if it includes various types of trajectories to help investigate the effect of the road design on the vehicle stability when failure modes intervene. It is advised to use an experimental vehicle that includes ESC to study the effect of ESC practically in increasing the stability. For the software used it is important to use software that can handle the sophisticated model; but on the other

hand it is more convenient to work with an open source tool that gives more flexibility in customizing the code and working transparently with various parts of it.

Overall it can be summarized that this research establishes a firm basis for a more comprehensive investigation of transient failure hazards associated with vehicular stability augmentation systems.

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