ANALYSIS OF IMPACT ENERGY AS A BASIS OF COLLISION SEVERITY IN VEHICLE ACCIDENTS

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Analysis of Impact Energy as a Basis of Collision Severity in Vehicle Accidents

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DECLARATION

This thesis is my original work and has not been presented for degree award in any other university.

Signature………………………………… Date ………………………………………

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This Thesis has been submitted for examination with our approval as university supervisors.

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DEDICATION

I dedicate this research work to my late sibling and guardian Conrad Ashika Khata.

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LIST OF SYMBOLS

e	Coefficient of Restitution
μ	Coefficient of friction
®	Registered trademark
™	Trade mark

LIST OF ACRONYMS AND ABBREVIATIONS

BEV	Barrier energy velocity
CRASH	Computer Reconstruction of Automobile Speeds on the Highways
EBS	Estimated barrier speed
EES	Equivalent energy speed
GVWR	Gross Vehicle Weight Rating
IIHS	Institute of Insurance and Highway Safety
JKUAT	Jomo Kenyatta University of Agriculture and Technology
KE	Kinetic energy
KeNHA	Kenya National Highways Authority
NHTSA	National Highway and Traffic Safety Administration
NTSA	National Transport and Safety Authority
SMAC	Simulation Model of Automobile Collisions
vCRASH	Virtual Computer Reconstruction of Automobile Speeds on the Highways

ABSTRACT

The need to advance vehicle safety is a significant aspect in vehicle manufacture dynamics. This is due to many variables present during vehicle collisions accidents. As such, major concerns should focus on the effects resulting from impact energy and forces from car crash. However, existing systems in vehicle transport safety employ little techniques to limit these effects. This study is aimed at determining the effects of frontal impact energy in vehicle collision accidents and its influence on speed adaptation. A simulation method was adopted based on impulse-momentum theorem, work-energy principle and intelligence speed adaptation (ISA) techniques. Data from the methods was collected and analyzed against Crashworthiness Data presented by General Motors Group using Minitab® 17.0, SigmaPlot® 13.0 and MATLAB®-Simulink platform. This analysis was conducted based on vehicle categories namely light, medium and heavy vehicles, also referred to as compact, intermediate and full-sizes respectively. It was established that impact energy inflicts severe vehicle damages at elevated speeds relative to vehicle weights. Furthermore, the study established that the force-deflection properties can be used to estimate full frontal impact energy, from which a relation between the conserved kinetic energy (KE) and full frontal impact energy was derived. The relation was used to develop a speed adaptation algorithm for implementing an ISA system in a MATLAB®-Simulink playground. Thereafter, a MATLAB®-Simulink vehicle model was developed whose speed profiles were adapted depending on monitored weights and speeds using the modelled ISA system. The system was regulated to ensure that the gain in KE was limited to the set impact energy value relative to each sampled vehicle. It was observed that speed limits could be adjusted in real time based on estimated impact energy value so that tolerable collision severity is achieved. The Simulink model confirmed that vehicle speed adaptation is possible based on conserved KE and a set value of frontal impact energy. However, this is exclusively depended on relative vehicle weights with respect to the type of vehicles. These findings can be useful to road transport safety authorities and insurance agencies in assessing vehicle collision accidents. Besides, vehicle manufacturers can apply the proposed solutions in the design of speed control systems and frontal bumper structures.

CHAPTER ONE

INTRODUCTION

This chapter introduces the works from previous researches and highlights the problem as far as assessment of severity of crush in vehicle collision accidents is concerned with respect to frontal impact energy. The chapter also covers the problem statement, justification and research objectives.

1.1 Background

This research provides an analytic approach on the influence of impact energy in vehicle safety dynamics. In theory, vehicle damage in car crash analysis can be used to estimate the energy absorbed during frontal, rear or side impacts. According to McHenry and Ray (2014), this energy is expressed in terms of equivalent energy speed (EES) and vehicle weight. The development of this research idea was limited to frontal damage, although the techniques used are general and can be extended in both side and rear damage analysis. Data was presented relating collision severity and impact speeds for full frontal impact tests. This was used to provide an incisive review on the force-deflection characteristics on the vehicle frontal structure. Average impact energy was estimated based on work-energy principle incorporating vehicle specific crush stiffness coefficients and suggested collision severity (crush damage).

The analysed data was used to develop an algorithm used to study the influence of full frontal impact energy on speed adaptation relative to collision severity. The algorithm was anticipated to output speed limits relative to suggested collision severity based on average impact energy value. Appropriate suggestions of EES were used to ascertain the need for adapting speed profiles against conserved KE in comparison to a set impact energy value.

From the classical mechanics, mass and speed have significant role on the energy absorbed during a car crash. The total sum of the kinetic energy is crucial in analysis of collision severity suffered by the involved bodies. From first principles, this energy is given as one half of the object mass multiplied by the square of the speed. Following Khorasani-Zavareh *et al.* (2015), accident prevention activities should not only focus on vehicle speed but also the impact energy value. This ensures that the traffic safety measures set in place are met. Understanding the role of kinetic energy in accidents will help to develop measures to reduce the generation, distribution, and collision severity inflicted during road accidents. This is because collision severity suffered is a quantifiable measure of injuries on vehicle occupants. Collision severity preventive activities should necessitate application of impact energy (conserved KE) in improving vehicle safety.

In his research Afukaar mentions that conventional speed monitoring systems have limitations in ensuring efficiency in the transport sector (Afukaar, 2003). This is because these systems do not consider the impact energy variations which influences collision severity levels during a crash. The KE at the time of a crash forms the basis of collision severity and is estimated based on extent of vehicle damage (crush damage). He proposes inclusion of the crush energy magnitude when designing active safety systems.

Fleming (2001) noted that vehicle safety is an important consideration in road transport industry and can be achieved using both active and passive safety systems. Active systems prevent accidents from happening while passive systems are inbuilt within a car to protect the occupants or users against severe injuries in the event collision accident occurs. Besides active systems reduce severe injuries and fatalities in a car crash. Therefore, any advancement in active safety systems technology sees a consequential decrease in the numbers of vehicle road carnage. For example, vehicle speed governors are some of the most important devices in the active safety systems category. These devices are used to limit the top speed of vehicles to predetermined levels (Toledo *et al.,* 2007). Unfortunately, they achieve little on the effects of speed and vehicle weight as variables of average impact energy (conserved KE) in a car collision.

From first principles energy cannot be destroyed, instead it is converted from one form to another. Likewise, during a car crash the KE gained by the bullet vehicle is transferred into inflicted crush of the vehicle body structure. Under such circumstances, there is a need to assess the severity of the crush with respect to energy. This can be achieved using crush analysis methods employed in accident reconstruction methods (Kodsi, *et al.*, 2017). Not only are these methods used to analyse collision severity against crash parameters like speed, impact energy and force but also to integrate crush stiffness coefficients so as to set accurate measures of vehicle EES. This can be utilised in active safety systems by incorporating crush energy algorithms in intelligence speed adaptation.

Following analysis done by Aldona and Grazvydas, the choice of control algorithm used to develop a vehicle safety system contributes a lot to the total outcome (Aldona and Grazvydas, 2007). In their research, when evaluating any vehicle safety system, its effectiveness has to be determined at different levels of collision severity so that in the event of a collision, collision severity is one aspect which needs to be tolerated.

Studies have shown that the collision severity is proportionate to the EES at time of the crash. This translates directly to the impact energy equivalent to the work done in vehicle damage. However, these variables alone cannot quantify the total effects of collision severity i.e. quantification of the intensity of an impact. Others include, damage location, direction of principle force, nature of object struck, time duration and crush stiffness coefficients for vehicles (Crosby *et al*., 2019; Bailey *et at*., 1995).

In the research done by Heikki and Lasse, it was shown that vehicle total weights and speeds are critical factors in vehicles crash dynamics (Heikki and Lasse, 2014). Depending on manufactures specifications, the stability of a vehicle varies depending on the real time axle weights. For example, at optimal loads, the vehicle can operate at maximum possible speeds upto certain weight limits when the vehicle is unstable (Hadi *et al.*, 2016). Therefore, in the event of a collision there will be varying levels of collision severity indices depended the weights.

In road traffic accidents, vehicle loading and speed are parameters that contribute to all energy transferred at impact. These two properties are connected to impact energy

(Newton's second law of motion) and so a focus should be placed on integrating the impact energy magnitude in speed monitoring systems. This is because the crush damage magnitude is thought to be influenced by both speed and crush energy. By definition, energy is taken as a function of monitored vehicle weight which influences the gain in KE. Where gain in KE implicates the impact energy transferred at crash as a basis of collision severity.

1.2 Problem Statement

During motor vehicle collision accidents, the severity of a collision is greatly influenced by impact energy conserved from the gained kinetic energy. For instance, at elevated speeds a large amount of kinetic energy is gained and so in case of a collision, this results into a high degree of collision severity during frontal impacts. From this, there is a direct relation between vehicle speed and the kinetic energy generated. It follows therefore that frontal impact energy is a factor of both vehicle speed and gross vehicle mass (GVM). By monitoring vehicle speeds and gross vehicle mass in real-time, then the expected frontal impact energy in case of a collision can be established and corrective measures can be suggested in order to adjust the speed. This will result in admissible vehicle crush damage. However, existing methods used in the monitoring and triggering of vehicle speed limits have limitations to include gross vehicle weights in the real time speed governing control sequences as proposed in this study. This is depicted in form of severe crush damage during vehicle collision accidents. In this research, a simulation method is used to analyse full frontal impact energy as a basis of collision severity in vehicle collision accidents. The results of frontal impact energy and the degree of collision severity have been used to develop an algorithm to monitor real-time vehicle speeds relative to gross vehicle weights, from which the gain in KE is evaluated and related to recommended frontal impact energy. This is then used to adapt vehicle speeds to new limits expected to inflict tolerable degrees of collision severity relative to gross vehicle weights in real time sequences for each sampled vehicle.

1.3 Justification

The primary evidence available for estimating degrees of collision severity is vehicle crush damage given in units of length from car crash tests. These tests suggest that collision severity is influenced by impact energy among other vehicle crash dynamics suggested in Kudlich-Slibar impulse-momentum model. To come up with proper solutions to such phenomenon, a lot of scientific research should be done to advice on the need to limit vehicle speeds based on impact energy magnitudes. While there is an increase of research in vehicle crash dynamics for both rear, side and frontal collisions, most of these studies focus on the advancement of passive safety systems in vehicles. These passive safety systems are just a fraction representative of the solutions developed towards safety advancement in vehicle manufacture dynamics. The impact energy in question is proportionate to the gain in kinetic energy by a moving vehicle, given as a function of gross vehicle weight and speed (EES). Notably enough, work done to inflict damage has been related to the energy absorbed by the body during the damage process from first principles in physics. Simulation methods can therefore be employed to conduct systematic analysis to show that total work done in vehicle damage is directly related to the conserved KE evaluated through monitored speed and weight in real-time. Further analysis of the average impact energy in full frontal collisions can be investigated using the work-energy theorem. This information can be exploited to come up with a relation between conserved kinetic energy, average frontal impact energy and collision severity in car collisions. Using this relation, algorithms can be developed relative to vehicle models for generation and adaptation of speed profiles to specific limits based on monitored speed and weight (gain in kinetic energy). Therefore, the information obtained from this research will be fundamental in designing and improving of conventional active safety systems in vehicle manufacture dynamics.

1.4 Objectives

1.4.1 Main Objective

To determine the effect of frontal impact energy on collision severity and its influence on vehicle speed adaptation.

1.4.2 Specific Objectives

1. To analyse the influence of vehicle speeds on collision severity based on momentum-based impact model in vCRASH® software suite.
2. To evaluate the influence of impact energy on collision severity based on the principle of energy conservation in vCRASH® software suite.
3. To characterise force-deflection properties for frontal impact energy based on the work-energy theorem.
4. To investigate the influence of impact energy on speed adaptation in vehicles using a developed system-model in MATLAB®-Simulink platform.

CHAPTER TWO

LITERATURE REVIEW AND THEORETICAL BACKGROUND

2.1 Introduction

This chapter gives a detailed literature review, technical aspects of the research and introduces the governing equations relating to the proposed study.

2.2 Road Traffic Safety

In the research done by Meng, Kees and Rob, it is suggested that the implementation of advanced driver assistance systems is a key factor in contributing towards road traffic safety (Meng *et al.*, 2005). The authors proposed speed assistance systems with reference to collision severity. This is in-line with motor vehicle safety where occupant safety is an important consideration among performance criterion in road transportation.

According to Du Bois and Chou, vehicle collision is a sequence of circumstances subjecting the vehicle structure to several forces; They proposed that measures have to be put in place when the forces involved are considered to exceed the energy absorbing capabilities of the vehicle structure to reduce severity levels. Consequently, to achieve total vehicle safety, crash tests are done on new vehicle models to determine crashworthiness and occupant protection (Du Bois *et al.,* 2004). The results from these test are however applied on a narrow scope in real world systems e.g. air bag systems but not in real-time speed monitoring with respect to mitigating degree of collision severity levels. Du Bois and Chou argued that the borrowed understanding of vehicle to barrier impact tests can be used in the development of the necessary models and algorithms for intelligent adaptation of vehicle speed as a counter measure of collision severity.

2.3 Vehicle Safety Systems

According to Demestichas, vehicle systems have advanced to Intelligent Transportation Systems (ITS) which involve the application of technology in movement of goods and people. The pronounced potential market is in services and products which includes systems for passenger vehicles and cargo transportation (Demestichas *et al*., 2010).

Since the year 2000, electronic systems have become an integral part to the operation, safety and control of the vehicle including collision warning and avoidance, emergency communications and vision enhancement. With modern technology, driving functions have been automated to the extent of artificial intelligence incorporation in vehicles. The vehicle market has become a significant user of monolithic circuits, where preliminary data is presented on the market categories and potential volumes for advancement of road transport safety (Anders and Kullgren, 2004).

In their study Stephens, Cory and Hopton, suggested that the potential behind rapid and precise predictions of vehicle crash response can be achieved with recent advancements in computer-augmented structural analysis models, where the most essential steps required is to develop and validate inequalities and numerical simulations. This process is however impeded by the complexity in the car structures, solutions to relating equation, the extensive calculations and inadequacies in basic data. Test data indicate that no major technical advances are required (Stephens *et al.*, 1995).

On economic analysis, capabilities can be developed through use of crash algorithm to acquire needed data for vehicle crash predictive system in every vehicle model category. With the advancement of crash test techniques through advanced experimental techniques and computer simulations, crash data can be readily retrieved and on-board computer code embedded into active safety systems for monitoring vehicle speeds, gross vehicle mass and expected collision severity in vehicles accidents (Stephens *et al.,* 1995). This sees the integration of intelligent transportation systems in road transportation systems aimed towards safety advancements.

The extensive use of embedded systems in the automotive industry has seen major changes in the architectural designs of vehicle safety systems. Where many traditional vehicle safety systems have merged to either one or more hybrid protection systems. From a safety system point of view, vehicle driving scenarios can be described into five states: normal driving state, warning state, crash avoidable state, crash unavoidable state and post event state. The first three states focus on accident avoidance while the last two focuses on damage mitigation. The crash avoidable state is described under both active and passive systems (Rohr, *et al.*, 2000). It is with this state that this research project proposes the need for intelligent adaptation of vehicle speeds using suggested average impact energy value as a basis of collision severity in vehicle collision accidents.

Considering the five states, it is apparent that vehicle safety systems development should be concerned with an integrated and intelligent system approach. This will see a mitigated approach in probability of collisions, collision severity and after crash mitigations using active and hybrid safety systems. Evidence in the effectiveness of active safety systems is seen in reduced road collision accidents severity. According to Rath and Knechtges, this has been championed towards the implementation of active and passive safety systems such as airbag systems, car crush zones and stronger body structure, and invention of side impact protection systems (Rath *et al*., 1995).

The use of intelligent transportation systems in mitigation of collision severity will improve vehicle safety. Since the systems will be focused on vehicle crashworthiness with regards to body structure manufacture dynamics. According to Aldona and Grazvydas, the use of intelligent transportation safety systems will champion not only improved safety in road transport but also vehicle safety infrastructure systems. This is in support to the idea that motor vehicle safety systems rely on vehicle dynamics as well as the vehicle kinematics for bodies in motion (Aldona *et al*., 2007).

From first principles, a moving body is described by a system of energies which is transferred during collision. An integrated system is able to analyse the influence of this energies with respect to crash variables such as stiffness coefficients, collision severity and impact speeds will see an additional advancement in intelligent transportation systems as applied to vehicle safety (Figueiredo *et al*., 2001).

2.3.1 Active and Passive Vehicle Safety Systems

Industrial strategies for vehicle safety systems has been evolving over the past decades. Earlier, individual passive systems and features such as seatbelts, airbags, knee bolsters, crush zones, etc. were incorporated in vehicles for saving lives and minimizing injuries when an accident occurred. Later advancements have seen measures such as improving visibility, headlights, windshield wipers, tire traction among others have been employed to reduce the probability of getting into an accident (Rohr *et al.*, 2000).

New dynamics in road transport require actively avoidance of accidents as well as providing maximum protection to the vehicle occupants and even pedestrians. With advancement in technology, safety systems like collision detection/warning systems, intervention systems, lane departure warning, drowsy driver detection and advanced safety interiors are gaining momentum (Rohr *et al.*, 2000). Advanced vehicle systems will see inclusion of detailed concepts of the safety state algorithms, a unified view of the safety system, and the technologies that are required to implement the systems described as active measures in vehicle safety dynamics.

2.4 Application of Crush Stiffness Coefficients in Vehicle Safety

Nystrom (2001) acknowledged that vehicles of different sizes, weights, manufacturing years and origin vary in how much they deform when involved in a crash. This is because they differ in their crush stiffness coefficients and is justifiable from conducted vehicle crash tests prior to releasing vehicles to the market. These tests are done by either car manufactures or crash simulations using accident reconstruction tools. Both approaches require that the reconstruction procedure evaluates several coefficients. These include force-deflection parameters and crush stiffness coefficients (Nystrom, 2001).

In their research Kodsi and others defined force deflection parameters as factors to represent the beginning of damage threshold which defines the maximum force per unit width that can be sustained without producing any permanent crush and linear relationship between the force and the amount of permanent crush (Kodsi *et al*., 2017).

Proper understanding of this properties can be used in implementing vehicle safety systems such as car airbags release mechanisms and vehicle crush zones.

According to Fay, crush stiffness coefficients can be used to estimate the crush energy transferred by bullet vehicle and in turn the crush energy can be used to predict the collision severity expected in a collision. Moreover, since impact speed is correspondingly used to estimate the crush inflicted, impact energy can be defined based on these coefficients. Important to note is that, when only one body is being analysed where little is known about the other body, then the change in impact speed is taken as an estimation to energy equivalent speed (EES) (Fay, 2001). Therefore, we can use averaged impact energy value to develop speed adaptation systems for advancement of vehicle safety systems.

2.5 Momentum-Based Impact Model for Car Crash Analysis

Vehicle crash simulation software use various principles for car crash analysis. For example, the momentum-based impact model relies on restitution instead of vehicle stiffness coefficients. This model is adopted for most crash simulations and was first described in Kudlich-Slibar model (Smit *et al*., 2019). In this model the user can calculate full impacts and sliding impacts. The model defines impact in two phases; compression phase and the restitution phase. At the end of compression phase, the velocities of vehicles at the impulse point are said to be identical for full impacts. The vehicles separate due to elasticity of the vehicle structures known as the restitution, *e*.

In his research Smit noted that the value of restitution from Kudlich-Slibar model can be explained as the ratio between the restitution impulse and compression impulse (Smit *et al*., 2019). This is called Poisson-restitution, which allows the restitution to be defined between -1 to 1, in vehicle crash simulation software. The negative value stands for a scenario when a collision has no common velocity when one body tears through another body. With this model, two main parameters that influence the preciseness of data collected are restitution and friction.

2.5.1 Restitution, *e*

According to Schram restitution is the amount of elastic rebound of the car after a crash. If the value of restitution is set to zero, the crush damage event (collision) is fully inelastic. Otherwise if the value of restitution is defined as 1, the crush damage is said to be fully elastic. This implies that there will be no crush damage after the crash. This value can also be defined as the ratio between the relative approach speed and the relative separation speed (Schram, 2003). From different studies, a common understanding for the value of *e* shows an inverse exponential curve between restitution and speed. This also shows that for high collision velocities, the value of *e* is between 0 to 0.1 and almost unity for low collision velocities.

2.5.2 Friction, μ

The coefficient of friction in crash simulation software is of importance when the impact is of sliding nature. An impact is said to be sliding if the impulse vector is outside of the friction cone. This is a scenario when the impulse vector makes an angle with the contact plane that is greater than 45° from both directions. Berg and others have stated that μ is taken to be the ratio between impact contact force and the normal component of crash (Schram, 2003; Berg *et al*., 1998). Little is known about vehicle to vehicle friction, and therefore a default value is difficult to state. In most crash test software, the value is taken as 0.1.

2.6 Stiffness Based Model for Car Crash Analysis

Besides momentum based impact model, vehicle crash analysis software employ the stiffness based model (Schram, 2003). In Momentum-Based Impact model, stiffness of the vehicle was not taken into account. In crash-based reconstruction, stiffness is defined by the force deflection factors as discussed in section 2.4 paragraph 2. This model allows the users of simulation tests to compare the crush profiles found in the simulation to those of the real vehicles. The model incorporates force-deflection curves in which the amount of crush is determined by the force magnitude that is applied to different nodes on the vehicle (Schram, 2003). This model as opposed to other accident analysis methods allows, the user to find different stiffness for different parts of the vehicle.

2.7 Crash Tests Analysis using Computer Simulation Algorithms

The science of crash simulation software was adopted to replace the expensive real world barrier tests conducted by manufacturers of automobile. This is because analysis of vehicle crash tests comes with a wider scope of understanding various aspects in vehicle collision accidents (Du Bois *et al*., 2004). For example, research institutes like National Highway and Traffic Safety Administration (NHTSA), Institute of Insurance and Highway Safety (IIHS) and European Accident Research and Safety Authority (EARSA) have been developing computer simulation models in order to analyse vehicle to vehicle and vehicle to pedestrian accidents (Du Bois *et al*., 2004). These Institutes been validating the models with full scale tests using Computer based algorithms and comparing the safety tests with real world accidents events. Reports given by these institutions present reconstructions of real-world accidents using the latest models and demonstrate the possibility of in-depth case studies on accident reconstruction software and analysis methods (Du Bois *et al.,* 2004).

These simulation models respond to various vehicle crash parameters from classical mechanics involving bodies in motion which include: impact speed, energy transferred, restitution and coefficient of friction and tyre forces in the case of vehicle dynamics (Burg, 1980). The analysis from these simulation corresponds to various impact configurations such as overall pedestrian behaviour, vehicle to vehicle collisions and vehicle to barrier are studied.

According to Jarašūniene and Jakubauskas, these type of tests are performed for ascertaining safety limits set in place by different state governments (Jarašūniene and Jakubauskas, 2007). For example, in the design of car air bag systems, various simulations are done on the vehicle to ascertain at what speeds the sensor can be triggered to inflate the airbag. The results from the simulations show that inflation happens when there is a collision force similar to running into a barrier at 16 to 24 km per hour. A mechanical switch is flipped when there is a mass shift that closes an electrical contact, telling the sensors that a crash has occurred. The sensors receive information from an accelerometer built into a microchip.

In their research, Prasad and Chou asserts that with the advancement of computer science technology, mathematical modelling has become part of Car Accident Reconstruction in engineering and many other areas of the physical sciences. Proposal by McHenry on using mathematical simulation models described the dynamic response of a vehicle occupant involved in a collision event (Prasad and Chou, 1993). Since then other sophisticated models have been developed for simulating occupant kinematics in crashes which has seen some increased developments in advanced passive and active vehicle safety systems.

During the past decades, a great deal of emphasis has been placed on the use of mathematical models in research and development in the field of automotive safety (Prasad and Chou, 1993). Examples of these models include: Crash Victim Simulation Computer Program, PC-Crash, CRASH3 Algorithm, CRASH Computer program, SMAC simulator and Virtual Crash suite. The listed algorithms analyse a number of vehicle control variables with respect to classical mechanics as discussed in the following sections.

2.8 Vehicle controls in Car Crash Simulations

Crash analysis software algorithms allow users to design complex and sophisticated sequences of driver inputs using the interfaces provided. This is in support to the rigid vehicle motion dynamics namely braking forces, steer effects, tire-force models, crash dynamic properties, motion sequence and braking lag amongst others as required for the subject simulation tests. A deep understanding of these parameters assists one to specify the order of defining a sequence of a simulated crash.

2.8.1 Tire force models

A moving vehicle comes with three tire force models: constant, linear and TMeasy models. The specified tire force model determines the vehicle's response to braking, acceleration, and steering inputs (Hirschberg, Rill and Weinfurter, 2007). When carrying our car crash simulation analysis, the TMeasy model finds most application as it represents semi-physical model structure between the virtual and real world events.

2.8.2 Steering inputs

This is defined at the axle. It can be thought of as the angle of the wheel heading with respect to the local for an axle with zero-track-width. The angle yields a specific turning radius which is directly dependent upon this angle and the vehicle wheelbase. The actual angles of the wheels during simulation are automatically adjusted so that the equivalent turning radius is kept for any track width assuming no sideslip. During simulation tests, the difference between the angles at each wheel is negligible. Each steering input for each wheel has an associated steering time. This is the time that controls the time interval over which the specified steering angle is set. The intermediate steering angles are determined by simple linear interpolation between the initial angle at the current simulation time-step and the final angle specified by the user (Rajamani, 2011).

2.8.3 Crash Dynamic properties

When performing a crash simulation various aspects of a vehicle design which influences the crash dynamics are grouped into drive train and braking, distribution of mass, suspension and steering, aerodynamics and tires. The simulator algorithm allows the user to input the parameters as per the crash test desired, this enables interpretation of the results with relation to real world events.

2.8.4 Motion sequences

Most car crash simulation software come with five sequences of motion which include, reaction motion allowing the user to specify a steering input and time interval over which the input is made. This sequence has options to lock wheels and modify tire-terrain coefficients of friction. Uniform motion offers similar options to reaction type. Deceleration motion allows the user to specify a steering input over an interval and specify braking options. Acceleration motion allows the user to specify a steering input over an interval and specify acceleration options. Accelerate-backward type allows the user to specify a steering input over an interval and specify acceleration options for a car in reverse direction.

2.8.5 Longitudinal tyre force

Rajamani suggests in his research that the user typically specifies the desired rate of acceleration or deceleration up to a maximum allowable based upon the adhesion value for the tire-terrain interface (Rajamani, 2011). With no steering input, the total longitudinal tire force on the vehicle is given by,

$$F_{X,Total} = \sum_{i=1}^{N} \widetilde{F}_{x'',j} = \sum_{j=1}^{N} f_{x'',j} . \mu_j . N_{Z'',j} \text{ , } \qquad \text{2.1}$$

where N describes the number of wheels undergoing braking or acceleration, μ is the value of adhesion which describes wheel tire-terrain drag factor for the tire, $N_{z'',j}$ defines the normal force at the contact path of the tire, and $f_{x'',j}$ is the total braking or acceleration on the tire expressed as a function of the maximum drag factor and defined with respect to the tire's longitudinal direction $\hat{x}''$. The variable $f_{x'',j}$ is bounded within $-1 \leq f_{x'',j} \leq 1$, where a negative value defines braking and positive value defines acceleration condition.

The longitudinal tire force is a function of acceleration input, which is regarded to be the obvious circumstance in an event of a collision. Using classical physics definitions, acceleration is defined as

$$\widetilde{a} = \frac{\sum_{j=1}^{N} \mu_j . N_{z",j}}{m} \quad \text{.......2.2}$$

$$f_{x",j} = \frac{m\widetilde{a}}{\mu \sum_{j=1}^{N} N_{z",j}} \quad \text{.......2.3}$$

It is bound with respect to lower limit and upper limit so that for each tire during simulation of vehicle speeds and mass, the total force experienced is given in Equation 2.3. Assuming that the adhesion factor on tire $j.$ is μ (Rajamani, 2011).

2.9 Principles of Car Crash Analysis

Collision analysis requires elaborate solutions to various causes of vehicle collision accident events. Through application of first principles with respect to impact dynamics described in Kudlich-Slibar model, simulations can be performed so as to establish to what extent does certain impact speeds and gross vehicle mass inflict severe injury. From such simulations appropriate regulatory models can be designed using vehicle motion principles. Adjustment of vehicle speeds and gross vehicle mass in a simulation environment will give elaborate understanding between crash parameters for viable barrier test data (Schram, 2003). This will enable in analysis of impact speed, transferred energy and stiffness coefficients towards modelling of a mathematical system for intelligent speed adaptation towards management of impact energy in vehicles prior to collision. These principles include depth of penetration, restitution, friction, impulse and equivalent energy speed, energy lost, crush damage/collision severity.

2.9.1 Friction

Frictional effect allows to specify the maximum coefficient of the friction used in a collision simulation model. This allows proper definition of impact energy modelling with respect to various crash circumstances. For instance, Schram and others suggest the coefficient of friction, μ as 0.1 (Schram, 2003).

2.9.2 Restitution

During a collision event, it is important to consider the overall effect of resultant velocity. By definition of restitution, impact energy value relies on the vehicle speed, where the coefficient of restitution

$$e = -\frac{\widetilde{v}_{\mathrm{Re}l,f} \cdot \hat{n}}{\widetilde{v}_{\mathrm{Re}l,i} \cdot \hat{n}} \quad \ldots\ldots 2.4$$

defines the nature of resultant changes in speeds. In Equation 2.4, v is velocity and e is as defined in subsection 2.5.1.

The value of e can take the range between $-1 \leq e \leq 1$, where 0 represents the situation in which all available energy is lost to inelastic effects, coefficient 1 represents the case when no available energy is lost in inelastic collision and a negative -1 refers to situations of no common velocity (Schram, 2003; Genta, 1997).

2.9.3 Depth of penetration

To quantity the collision severity in simulation algorithms, the depth of penetration is analysed. This is defined as a function of time expressed as Δt_p. It describes the impulse centroid within the volume of the vehicle. It shows that when a vehicle is engaged in a crash, then the impulse centroid will be defined from initial time and depth of penetration as $t_i + \Delta t_p$.

2.9.4 Impulse and delta-v

From first principles, the impulse vector is defined as

$$\left|\widetilde{J} \cdot \hat{n}\right| = (1+e) \cdot \widetilde{m} \cdot \left|\widetilde{v}_{\mathrm{Re}l,i} \cdot \hat{n}\right| . \quad \ldots\ldots 2.5$$

where J is the impulse, n is the normal vector, m is the object mass and v is relative velocity. Equation 2.5 defines the force delivered to the colliding vehicle at the measured impact speed. The value of impulse can also be related to change in velocity (EES) of the vehicle as defined in

$$\Delta v = \left| \frac{\tilde{J}.\hat{n}}{m} \right| = EES \quad \ldots\ldots 2.6$$

where J is the impulse, m is the object mass and Δv is velocity change. Since the study focused on the bullet vehicle whose parameters are known, therefore to simulate influence of speed on collision severity, the velocity change is equated as EES (Berg, 1998).

2.9.5 Crush damage (Collision severity)

It is related to the impact forces during collisions and EES. Other factors being equal the greater the EES, the greater the potential of crush damage. By comparing the EES to vehicle damage in terms of collision severity, we gain knowledge of vehicle crashworthiness and effectiveness of protective devices in vehicle transport safety. The value of EES is analogous to the collision severity dosage, hence proper knowledge will enable appropriate protective measures (McHenry *et al*., 2014).

Collision severity from collisions has been scientifically related to the impact speed at impact. This has been done through the use of experimental crash models and simulated impact mechanics that gives a direct relation between changes in speed and crush depth. This approach has been exploited for accident reconstruction using computer crash algorithms like Computer Reconstruction of Automobile Speeds on the Highways (CRASH) and Simulation Model of Automobile Collisions (SMAC) (McHenry *et al*., 2014).

According to research by Brach and Welsh , CRASH analysis employs vehicle equivalent mass, visual estimation and other impact mechanics. Using the principal direction of force and tangential correction factor, a relationship between crush energy to impact speed can be established. An in-depth analysis of speed in relation collision severity has benefits in advancement of vehicle safety (Brach and Welsh, 2007).

2.9.6 Energy lost, Impact Energy and Force

Energy lost is also described as the energy conserved in a collision. It is a function of vehicle mass and the impact speed. This energy inflicts damage in form of crush in the vehicle body structure. The degree of energy lost and hence collision severity will depend on the impact speeds. Therefore, if the energy is not regulated, it means critical injuries in accidents and likewise loss of lives. Poor road safety measures in place put little consideration in management of energy lost (energy absorbed) during collision. From classical definitions of rigid body dynamics, energy transferred in collisions is

$$E_{loss} = \frac{1}{2}.\left(1 - e^2\right).\tilde{m}.\left(\tilde{v}_{\mathrm{Re}l,i}.\hat{n}\right)^2 \quad \ldots\ldots 2.7$$

where $\tilde{V}_{Rel,\,i}$ is the velocity equivalent to EES that influences the collision severity and m is the gross vehicle mass reduced with respect to restitution e neglecting torque coefficients (Brach *et al*., 2007; Marine, 2005; Marine, 2002; Fonda, 1999).

In his research Vangi affirmed that the best approach to define this energy is to experimentally conduct several crash tests to simulate the accident conditions. He notes that this is expensive and time-consuming task which cannot always be undertaken. He suggests an alternative is to compare the physical damage to crash tests damage from published tests (Vangi, 2009).

Fay in his research suggests that the energy absorbed technique is appropriate method for effective estimate of the impact speeds against collision severity. This method assumes that the linear force-deflection does not vary across the width of the vehicle i.e. the center of the vehicle is not stiffer than the fender area in the frontal impact (Fay, 2001).

Conversely, impact energy has been related to changes in speed. The corresponding variation in speed produces four times the impact energy of the initial impact barrier speed. This can be very alarming when high impact speed and large vehicle mass are put into consideration (Vangi, 2009). To characterize the levels of severity in both the front and rear impacts, the concept of speed change is crucial. The EES defines the energy absorption likely to occur in a collision. The EES of the colliding vehicle defines the change in the vehicle speed and the possible direction it will take over the duration of impact. This implies that the collision severity or collision severity is quantified in terms of delta-v (McHenry *et al.*, 2014).

The equation used in the model defines force as,

$$F = a_0 + a_1 C \text{ ....} 2.8$$

where F is impact force, C is the crush damage, a_0 and a_1 are vehicle specific crush coefficients. Equation 2.8 describes the force per unit width of the frontal structure as a function of crush (Neades and Roy, 2011; Neptune, 1999; Neptune, 1998). Equation 2.8 can be applied in full frontal crash tests to define force deflection properties using

$$\frac{1}{2}mv^2 = \iint_{C,w} (a_0 + a_1 C)\,dC\,dw + k \text{ ....} 2.9$$

where C is collision severity, w is vehicle width or frontal length, a_0 and a_1 are vehicle specific crush coefficients. Equation 2.9 is justified from work energy theorem definition where energy absorbed in inelastic collision is equal to work done.

Vangi suggests that the energy absorbed (or work done) to achieve a particular degree of crush in a structure can be obtained by integrating the local force per unit area over the volume of crush damage (Vangi, 2009). If the damage is uniform over the vertical dimensions, then this integration can be eliminated to leave integration with respect to crush and width given as

$$E_i = \int F\,dC\,dw + k \text{ ....} 2.10$$

where E_i is the impact energy, F is impact force, C is collision severity, w is vehicle width and k is a constant defining some initial energy which is absorbed with no crush.

2.9.7 Equivalent Energy Speed, EES

According to Berg the equivalent energy speed is the speed which inflicts total impact energy lost in a collision (Berg, 1998). This is defined as

$$E_{loss} = \frac{1}{2}\left|\langle F.\hat{n}\rangle.C\right| = \frac{1}{2}.m.EES^2 \quad \ldots\ldots 2.11$$

where E_{loss} is the energy lost during collision, m is the gross vehicle mass, F and C is as defined in subsection 2.9.6 for Equation 2.8.

$$EES = \left(\frac{2E_{loss}}{m}\right)^{1/2} \quad \ldots\ldots 2.12$$

where E_{loss} is the energy lost during collision. This energy is then absorbed by the vehicle structure in inelastic collision.

It is important to be able to predict the EES in vehicle impacts from the available data of the vehicle to crash tests so as to advance vehicle safety. For example, the National Highway and Traffic Safety Administration (NHTSA) and McInnis Engineering Corporation in the United States of America have over one thousands staged collisions data for vehicle to barrier tests. This implies the crush coefficients for most vehicles are known (Nystrom, 2001). The data can be used for front and rear collisions analysis directly or using accident reconstruction methods for collision severity prediction.

2.10 Car Crash Tests Experiments

In their simplest form, car crash tests consist of propelling a bullet vehicle on a barrier at a known speed and measuring the collision severity as shown in Figure 1. During these tests, different forms of impact configurations are used concerning the intended research findings (Neades and Roy, 2011; Prochowski, 2010). These tests have established that speed range of 40 kmh^{-1} to 65 kmh^{-1} is termed as the band within which most injury accidents will occur.

Johnson and others in their study suggested that when analysing the impact energy and change in impact speed at point of impact, the simplest approach is to adopt head-on collisions with a solid immovable barrier (Johnson and Gabler, 2012).

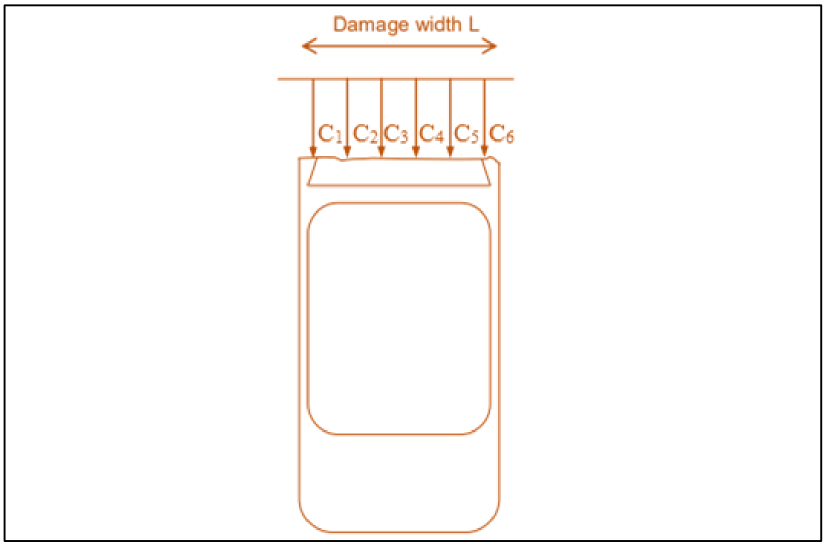

Figure 1: Crush damage suffered after a head-on impact crash test

From Figure 1, L is the original body width or length of the vehicle while the labels C_1 to C_6 represents a series of crush measurements taken across the face of the damaged automobile. Assuming head-on collisions, a simplified and uniform crush profiles are depicted for specific vehicle models. These crush profiles are available with world agencies like NHTSA in the US for research reference in accident reconstruction journals (Johnson and Gabler, 2012).

According to Neades there are crush profiles for different vehicles (Neades and Roy, 2011). In his research he used a Volvo 850 model to give a more practical review of a sample crash test data as shown in Table 1.

Table 1: Crush profile of Volvo 850 of 1994 crash test (Neades and Roy, 2011)

			Crush damage (m)						
Vehicle	Vehicle Mass	Test speed	C_1	C_2	C_3	C_4	C_5	C_6	C_{ave}
Volvo 850	1442 kg	56.3 kmh^{-1}	0.419	0.459	0.488	0.495	0.465	0.431	0.460

From the test profile, the average crush depth for the test speed of 56.3 kmh^{-1} is found to be 46.00 cm. Where threshold speed and gradient constants given as

$$v = b_0 + b_1 C \qquad \text{.......2.13}$$

where v is impact speed, C is crush damage in units of length, b_0 and b_1 are vehicle crush coefficients in units of mph and mph/m respectively.

Equation 2.13 forms the simplest description of the EES with respect to vehicle coefficients b_0 and b_1 as suggested by Campbell and McHenry (McHenry *et al.*, 2014; Campbell, 1974). Therefore, for Volvo 850 series this translates to

$$v = 1.05C + 8 \qquad \text{.......2.14}$$

where b_1 = 1.05 and b_0 = 8. Using Equation 2.14 EES for Volvo 850 under this test was estimated to be 40 kmh^{-1} with a crush damage of 30 cm. However, if the same vehicle suffered similar damage by collision with another car, it will prove impossible to calculate the initial speed. This is because little is known of the other vehicle in terms of both impact energy and the changes in velocity (Johnson and Gabler, 2012).

Therefore, the proper assumption made is that of a fixed barrier stated as

$$\Delta V_1 = \sqrt{\frac{2(E_1 + E_2)}{m_1(1 + \frac{m_1}{m_2})}} \qquad \text{.......2.15}$$

$$\Delta V_1 = \sqrt{\frac{2E_1}{m_1}} \quad \text{.........2.16}$$

where m_1 and m_2 represent the vehicle masses and the E_1 and E_2 stand for the respective crush energies of involved vehicles. Equation 2.16 is the reduced form of Equation 2.5 and it estimates impact velocity as equated in Equation 2.13. From Equation 2.16 the velocity change at crash is known as the EES, or the initial speed at which most of the kinetic energy is converted to crush energy (Neades and Roy, 2011).

One method to analyse collision severity is by finding the total crush along the region of crush damage referred to as damage profile measuring procedure. This can be arrived at using a mathematical formula of finding the area of a non-uniform region. Suggested regular shapes are used to estimate the crush over that region e.g. using trapezoidal approximations of the damage region as shown in Figure 2. The individual crush zones are summed up to give the collision severity used to estimate the crush energy (Fay, 2001).

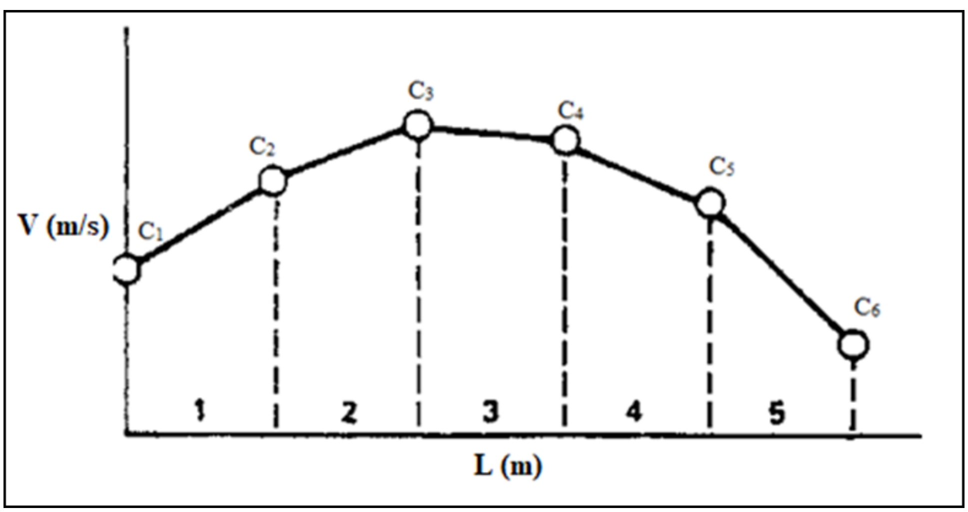

Figure 2: Trapezoidal approximation of crush damage for a six-point crush profile (Fay, 2001)

Where, V is the impact velocity during full-frontal impact that produces a certain degree of damage and L is the crush damage or collision severity due to the impact velocity.

This research justifies the need to include impact energy factor in speed adaptation systems. Its influence on intelligence monitoring of speed and gross vehicle mass in real time will see effective mitigation of collision severity in vehicle accidents relating to collisions. This is based on considerations that both speed and vehicle weights influence collision severity indirectly from the conserved KE. The study tries to qualify that the inflicted crush damage can be used as an estimate of collision severity.

Knowledge from literature review and theoretical background was used in formulating relevant relations to develop an algorithm for intelligence speed adaptation for different vehicle relative to: conserved KE, frontal damage energy, estimated crush damage and other vehicle crash dynamics constants. So as to achieve the research design, crash test analysis techniques of accident reconstruction science were studied. This offered a platform to understand and relate collision severity, vehicle speeds and gross vehicle mass in application to active road safety systems.

CHAPTER THREE
MATERIALS AND METHODOLOGY

3.1 Introduction

This chapter presents the materials and methods that were used to quantify and assess the need for including impact energy as a factor in vehicle speeds adaptation processes, acquisition of the data and the procedural work undertaken in the respective study area.

3.2 Area of study

Vehicle safety is a wide area of research that focuses on possible technologies employed in advancing safety of road users. Aldona and Grazvydas asserts that intelligent vehicle safety systems have been largely adopted to ensure superior safety on roads both in vehicle-based or infrastructure-related systems (Aldona *et al*., 2007). They suggest that safety of road users depend on various parameters such as speed, vehicle weights and impact energy in an event of a collision.

Over the last few decades, strategies in automotive safety industrial systems have been evolving with a focus on both active and passive safety systems. Initially, individualized passive devices and features like seatbelts, airbags, knee bolsters, crush zones, etc. were developed for saving lives and minimizing injuries when an accident occurs. Later on, preventive measures including improving visibility, headlights, windshield wipers, tire traction and others have been deployed to reduce the probability of getting into an accident with focus on actively avoiding accidents as well as providing maximum protection to the vehicle occupants and even pedestrians (Rohr *et al*., 2000).

This research is aimed at finding a concept for improving vehicle safety during vehicle collision accidents relative to vehicle speeds adaptation by considering gross vehicle mass values, crush damage and impact energy in frontal collisions. By doing this, vehicle engine speeds can be adapted within tolerable limits based on the gained kinetic energy (monitored gross vehicle mass and speeds). This will also ensure that the energy transferred to the vehicle body structure during a collision inflicts admissible collision severity thresholds.

A review on vehicle safety systems and models required to implement this idea was analysed from the objectives findings. Other areas of the research such as vehicle controls, accident reconstruction analysis and car crash test simulations were addressed. This aimed to familiarise with the existing relations between car crash dynamics and impact energy recorded during collisions.

3.3 Materials

A simulation method was adopted for this study with considerations on both accident reconstruction science and physical world simulation procedures. All simulations were run on the same machine which had the following specifications: -

- Operating system: Windows 10 pro 64-bit
- Processor: Intel(R) Core(TM) i3 5005U CPU @ 2.00 GHz
- Memory: 4GB RAM

For system modelling and simulation, the MATLAB® stable release R2015a Simulink playground from MathWorks developer(s) was used. It runs on an environment build using C, C++ and Java languages. For accident reconstruction analysis the Virtual CRASH® version 4.0 developed by vCRASH®, Americas, Incorporation was adopted. The software provides an interface to stage virtual accidents based on impulse – momentum principle suggested in Kudlich-Slibar model (Smit *et al*., 2019).

Data collection was done using Virtual CRASH® and Simulink software. Since they come with interfaces incorporated with data collection and displays applications i.e. report generation tools and display scopes. This provided a virtual link to monitor experimental results. Data analysis tools used were Minitab® 17.0 developed by Minitab LLC and SigmaPlot® 14.0 developed by Systat Software Incorporation.

3.4 Methodology

Simulations were conducted to qualify and analyse the influence of full frontal impact energy as a basis of collision severity in vehicle accidents relating to collision on speed adaptation. Experimental designs were modelled and reviewed in the study by applying work energy theorem and impulse-momentum principle to understand vehicle safety dynamics. According to Schram, the use of simulation methodology has become one of the most common experimental methods (Schram, 2003). This is due to their simplicity and ability to analyse real world events using virtual systems. This methodology involves the determination of characteristics relating real life events and their underlying principles. Models developed through this experimental method are known to offer the most economical techniques of assessing the real problems under study. The study was categorised into three main segments based on the specific objectives of the study: -

First, car crash simulation experiments were conducted focusing on analysing the influence of vehicle speeds on collision severity in full frontal impacts was conducted. Followed by evaluating impact energy from measured speeds, collision severity in full frontal impacts based on principle of energy conservation and damage profiling procedures in car crash analysis. Prior to these procedures, selection of test parameters for car crash simulation experiments were done.

Secondly, characterisation of force-deflection properties with a focus on defining the energy conserved in full frontal impacts from differential and integral calculus techniques based on the collected data was carried out.

Finally, an investigation on frontal impact energy influence on vehicle speed adaptation sequencing was carried out. This was procedural on three tasks where a vehicle model was designed using MATLAB®-Simulink platform to produce needed speed profiles unique for each sampled vehicle, development of speed monitoring and adaptation algorithm depended on conserved kinetic energy and frontal impact energy as the inputs and lastly design of vehicle speed adaptation control system using MATLAB®-Simulink which had input and output modules. The control system utilised speed profiles from MATLAB®-Simulink designed vehicle as an input variable and output module displayed adapted speed limit profile signals to be utilised by interfaced speed governor unit as this study suggest.

3.4.1 Selection of Test Parameters for Car Crash Simulations

This stage was concerned with initial activities involving determining the parameter inputs for use in simulation of virtual collision accidents. It is from these initial activities that the study relates the collision accidents causation factors in real world to the controlled simulations using computer programs. These simulations provide important review on data generated using accident reconstruction analysis methods and crash test data used to model needed equations. Research done by Neades and others observed that during collisions the contact forces between the involved vehicles has been found to vary over time (Neades and Roy, 2011). These forces depend on the local vehicle structure, impact velocity, the contact situation (elastic or inelastic collision) etc. The dependencies of these parameters are highly non-linear and difficult to formulate. Hence, the treatment of these parameters as integral values in simulations is more efficient as proposed in the Kudlich-Slibar impulse-based analysis model (Burg, 1980; Smit *et al*., 2019). These parameters include: tire force models, steering inputs, restitution, friction, EES, energy lost, motion sequence and longitudinal tire force.

i. Tire force model

When carrying out car crash simulation analysis, the TMeasy model find most application as it represents semi-physical model structure between the virtual and real world events. The model defines both the braking, acceleration and steering behaviour of the vehicle at the instant of car crash. The tire force parameter is assumed to have negligible impact on the suggested system model for control of energy transferred in a collision by the bullet vehicle. This parameter is defined under vehicle controls in the previous section.

ii. Steering Inputs

The steering inputs as defined in earlier section defines the initial impact angle upon which the bullet vehicle will collide with the barrier (another car or incident object). In accident analysis tests this angle has an overall effect on the formula used to define the absorbed energy. The study assumes a 0° angle as the steering input so that from Equation 2.10, the overall energy absorbed will be defined. This steering input value is defined under vehicle controls in subsection 2.8.2.

iii. Motion sequence

This parameter is discussed under crash dynamic properties in the previous chapter. It explains the type of motion described by the bullet vehicle prior to colliding with the barrier. The study adopted a linear acceleration profile, which defines the speed profile of a car from the initial time of ignition to the time when the car acquires maximum speed described by the car engine power and torque.

iv. Longitudinal types force

It is a parameter defining crash dynamic properties. It is governed by the motion sequence adopted in the study. As explain in the previous section, the longitudinal tire force is considered to have significant impact with relevance that the motion sequence chosen is a linear acceleration model.

v. Restitution

It is a variable defined under principles of crash analysis. The study assumes a scenario when the bullet vehicle collides and the effect is that of inelastic collision. In this study restitution is set to 0.100 (all energy is lost in inelastic collision). This constant defines the kinetic energy transferred to the vehicle structure. It is as defined in Equation 2.4. The parameter has significant effect in our study area as expressed in Equation 2.9.

vi. Friction

Little is known about vehicle to vehicle friction, and therefore a default value is hard to state. The study assumes a crash scenario where the value is taken as 1.000. However, for this study, it is negligible since little is known about the road surface before the time of crash. It is discussed under principles of crash analysis in the previous section.

vii. Crush damage (Collision severity), C

It is related to the impact forces during collisions and EES. Other factors being equal the greater the EES, the greater the potential of occupant injury. By comparing the EES to vehicle damage and occupant injury, the study gained knowledge of vehicle crashworthiness and effectiveness of protective mechanism in vehicle transport safety. Since the value of EES is analogous to the collision severity dosage, its proper knowledge enables appropriate modelling of impact force characteristics versus collision severity.

3.4.2 Car Crash Simulation Experiments

Staged crashes were simulated to understand the relationship between crush severities, vehicle speeds and average impact energy. This was achieved using Virtual CRASH® computer-based accident reconstruction software. The software provides a virtual environment of real world events with a user interface as shown in Figure 3.

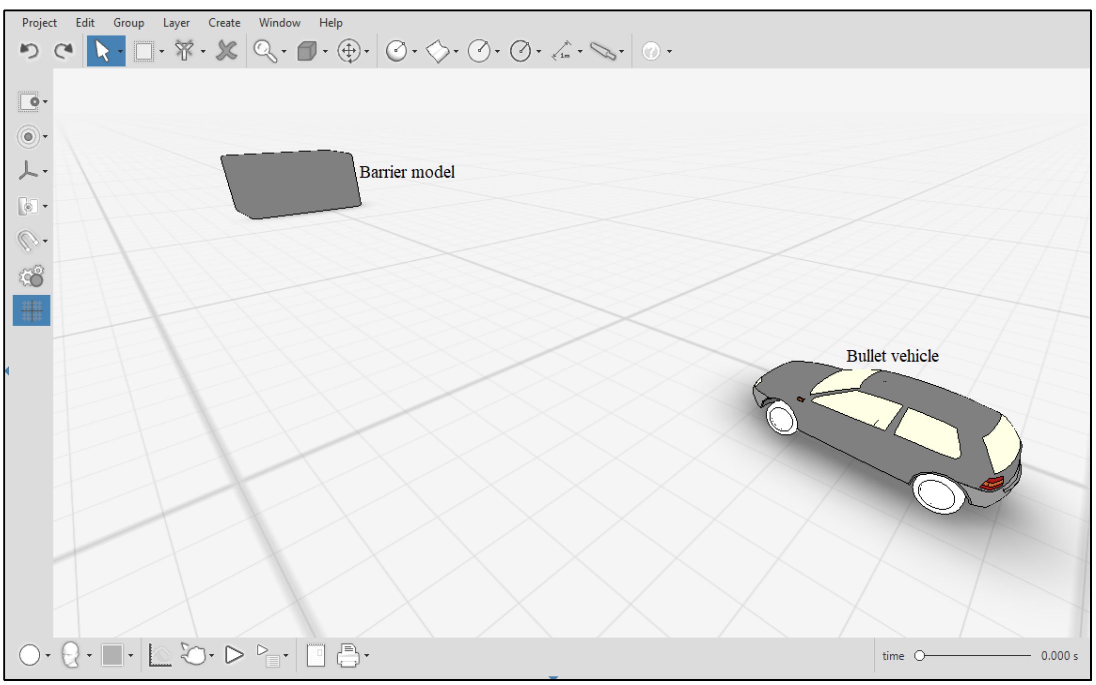

Figure 3: Virtual CRASH® 4.0 playground

The vCRASH® interface comes with a variables control panel as shown in Figure 4. It allowed modification of test conditions discussed previously with reference to impact-based model (Smit *et al*., 2019). During each crash test a different vehicle model as given in Table 2 was selected. A uniform fixed concrete crash barrier object was modelled with dimensions of (9.0 x 2.5 x 3.0) m, mass of 1 x 10^6 kg, and vehicle controls in Figure 4 were adjusted to suite Kudlich-Slibar model.

The numeric figures of the concrete barrier are important as they can help in estimating the density of the barrier in other related studies. For example, the density can be computed and related to what we have in the physical environment.

Simulation tests were done as shown in Figure 5 and data recorded at the point of crash from data panel of Figure 6.

Table 2: Vehicle model specifications

	Full-size	**Intermediate**	**Compact**
	Chevrolet crew cab Silverado-2003-7	Chevrolet blazer LS 2000	Chevrolet corvette C6-Z06
Curb weight (kg)	2485.00	1825.00	1420.00
GVWR (kg)	4173.00	2426.00	1598.00
Payload (kg)	1687.00	601.00	169.00
Width (m)	2.00	1.71	1.84

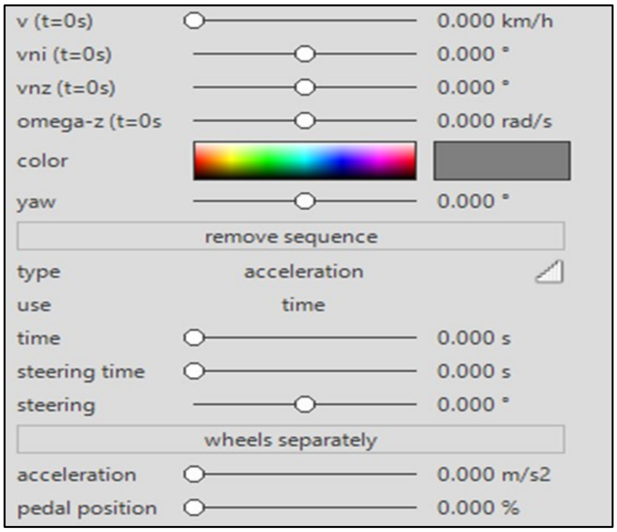

Figure 4: Vehicle controls interface in vCRASH® suite

For each simulated crash, data for impact speed, impact energy and crush damage was recorded. This data was analysed and recorded as shown in Appendix I. These data were compared to the previous works done on similar tests for different vehicle categories from GM automobiles as given in Appendix C. Using Minitab® 17.0 data analysis tool, graphs for each vehicle model were plotted for speed and impact energy versus collision severity.

Model equations defining the relationship between impact speeds and collision severity were determined from the plotted graphs. Impact energy versus collision severity was analysed against crash dynamic parameters. This gave us equations defining impact energy in terms of specific crush constants and collision severity. The crush constants related to impact speeds were later used to characterise force-deflection properties in full frontal impacts to achieve the third objective of this study.

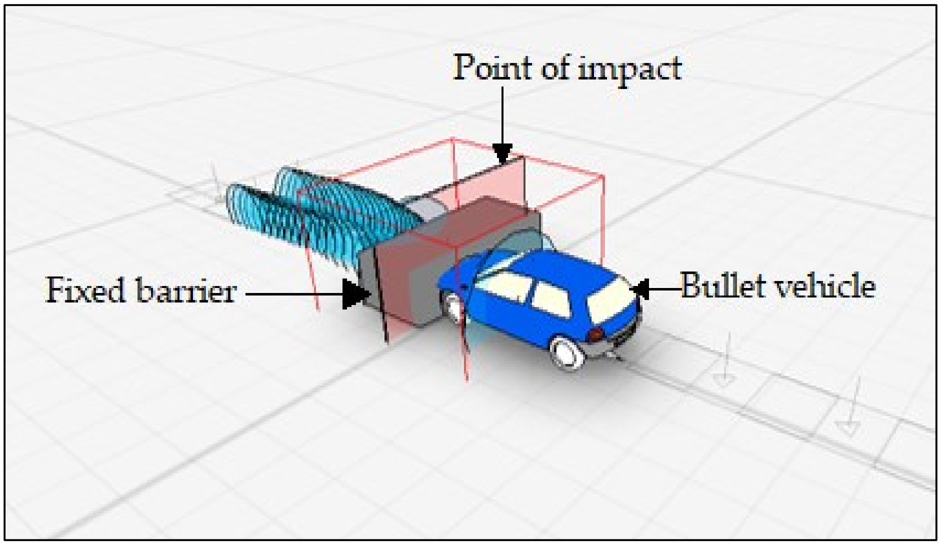

Figure 5: Crash simulation in vCRASH® 4.0 accident reconstruction software

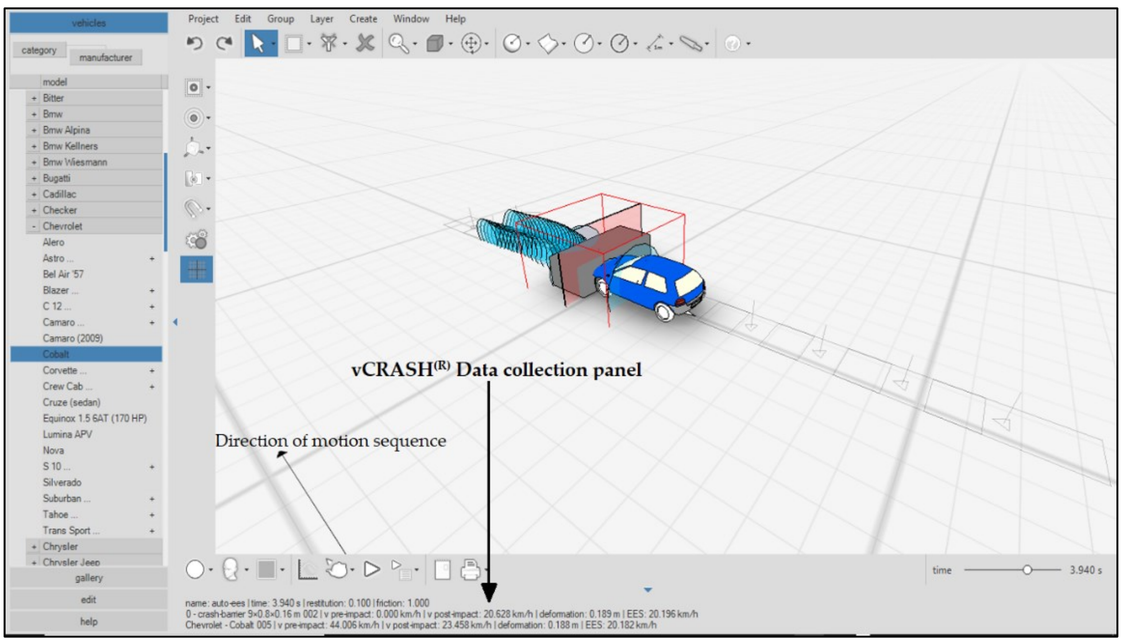

Figure 6: vCRASH® data collection panel

3.4.3 Characterisation of Force-Deflection Properties for Frontal Impact Energy

To best formulate the average impact energy respective of each vehicle, the study classified the sampled vehicles into various categories based on force-properties which was denoted as k_0, k_1 and G. The constants k_0 and k_1 were derived from the analysed crash tests data collected using the work-energy theorem defined as

$$w = FsCos\theta \quad \ldots 3.1$$

$$F = ma \quad \ldots 3.2$$

where Θ is the impact angle (steering input), a is the acceleration (motion sequence), m is the gross vehicle mass and F is the impact force as defined from first principles. These constants are related in the final equation of average impact energy as crush stiffness coefficients, k_0, and k_1. From Equation 3.2 acceleration was defined using differential calculus as

$$a = \frac{d}{dt} \bullet \frac{dx}{dt} = \frac{d^2x}{dt^2} = v\left(\frac{d}{dx} \bullet \frac{dx}{dt}\right) = v\frac{dv}{dx}. \quad \ldots 3.3$$

Thus acceleration was expressed as a function of both velocity vector and velocity differential term with respect to crush damage segment d/dx as

$$a = v \cdot \frac{dv}{dx} \quad \ldots 3.4$$

where dv is the change in impact speed, b_0 and b_1 are vehicle specific crush coefficient constants and v is the impact speed obtained from crash tests simulations experiments. The term in the derivative was used to express acceleration with respect to crash analysis parameters as $dv/dx = dv/dC = b_1$ form $v = b_0 + b_1C$. By substituting these two terms expressions in Equation 3.4, we defined acceleration in terms of crush stiffness coefficients and collision severity as

$$a = v \cdot b_1 \quad \ldots 3.5$$

$$a = (b_o + b_1C)b_1 \quad \ldots 3.6$$

Rewriting Equation 3.1, work-done in joules or simply the energy transferred during collision was defined using acceleration term from Equation 3.6 as

$$w = m(b_0 + b_1C)b_1 \cdot s \cdot Cos\theta \quad \ldots 3.23$$

where the term $m(b_0 + b_1)b_1$ defines the impact force of the impulse centroid in Figure 5 during collisions. Therefore, the force model equation was defined as

$$F = mb_0b_1 + mb_1^2C \quad \text{.......... 3.7}$$

where the term mb_0b_1 and mb_1^2 defines force-deflection properties without inclusion of vehicle frontal width.

The study assumed a uniform collision severity across the vehicle width defined as w_0 to be inflicted during a collision. Thus impact force in Equation 3.7 was expressed as force per unit width to given as

$$F = \frac{m}{w_0}(b_0b_1 + b_1^2C) \quad \text{.......... 3.9}$$

Using Equation 3.9 a new set of data relating impact force and collision severity was collected for the respective crash tests performed on the three vehicle sampled. The data was recorded as shown in Appendix II. The data was analysed and plotted using SigmaPlot® 14.0 data analysis tool to give the coefficient terms k_0 and k_1. These terms define the evaluated force-deflection properties for impact force in full frontal impacts characterised based on vehicle mass (m), full frontal width (w_0) and vehicle specific crush stiffness coefficients (b_0 and b_1) as shown in Table 3.

Table 3: Characterisation of Vehicle specific Force-deflection Properties

Property	SI unit	Characteristics
k_0 coefficient	N/m	$k_0 = \frac{m}{w_0} b_0 b_1$
k_1 coefficient	N/m^2	$k_1 = \frac{m}{w_0} b_1^2$
G coefficient	N	$G = \frac{{k_0}^2}{2k_1}$

The properties k_0 and k_1 were found to be vehicle specific from the analysis done. The graphs obtained were used to evaluate average impact energy with respect to force deflection properties and collision severity as

$$E_{average} = w_0 (k_0 C + \frac{k_1 C^2}{2} + G) \quad \text{……….............……………………………………3.10}$$

where $E_{average}$ is the average estimate of full frontal impact energy value derived from physics first principles, w_0 is vehicle full frontal length, k_0 and k_1 are the defined force deflection properties and G is a constant of integration related to properties k_0 and k_1.

3.4.4 Investigating the Influence of Impact Energy on Vehicle Speed Adaptation using MATLAB®-Simulink Platform

The final stage of the study focused on investigating the influence of impact energy values on vehicle speed adaptation using MATLAB®-Simulink platform. The software provides a virtual interface that can be used study occurrences of real world events. This section was based on three procedures as discussed below: -

3.4.4.1 Vehicle Design using MATLAB®-Simulink Playground

The preliminary stage aimed at designing a vehicle model using Simulink™ playground given in Figure 7. The vehicle model had all major parts of a real world automobile namely generic spark ignition engine, driver inputs, tires utilising TMeasy configuration, gear module, gear differential modules and the vehicle body. With the vehicle model, a normal speed profile of a car was generated. The normal speed profile was referred in the entire study as unregulated speed with reference to the proposed system for speed adaptation model. From the ignition time, the vehicle had a motion sequence defined the function

$$v = u(t) \qquad (3.11)$$

where v is the final velocity and u is the initial velocity at time t. Equation 3.11 is a linear acceleration sequence state.

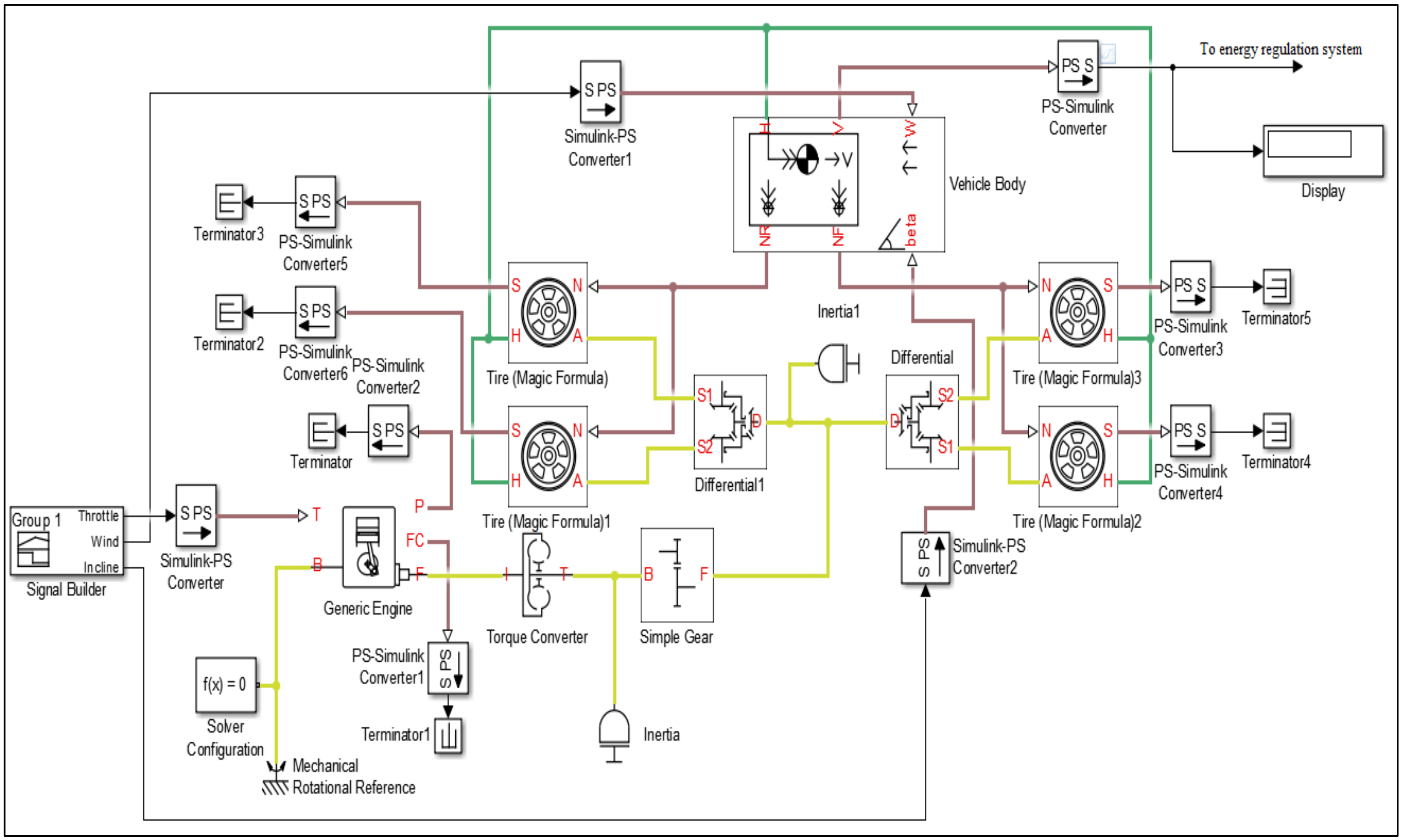

Figure 7: Simulink designed vehicle model

Figure 7 shows a MATLAB® vehicle model which was modelled on the basics of a generic gasoline engine. It had a unity throttle input from the signal builder. This ensured acceleration motion state sequence. Simulink-PS converter was used to change the dimensionless input signals to physical signals that can be used by blocks that take physical signals as inputs. PS-Simulink converter changes the input physical signal to a dimensionless output signal. Generic engine block represents a system-level model of a spark ignition engine. In this type of engines, the air-fuel mixture in the combustion chamber is ignited by a spark generated by the spark plug. The throttle input signal lies between zero and one and defines the torque demanded from the engine as a fraction of the maximum possible torque.

The engine assumed specific characteristics settings as each simulation was vehicle specific with different maximum engine speed outputs. The settings depended on the values of Maximum Power, Engine Speed at Maximum Power, Maximum Engine Speed, Inertia, Initial Engine Speed, Stall Speed, Speed Threshold, Torque transmission time constant, Speed Ratio Vector and Torque ratio vector. The following were the settings chosen for one of the sampled vehicle: -

- Maximum Power: 302 HP
- Engine Speed at Maximum Power: 5300 RPM
- Maximum Engine Speed: 8500 RPM
- Inertia: 0.2 $kg.m^2$
- Initial Engine Speed: 1500 RPM
- Stall Speed: 500 RPM
- Speed Threshold: 100 RPM

The Maximum Power signifies the maximum possible power that can be generated by the engine prior to the transmission of torque across the driveline. The Engine Speed at Maximum Power signifies the rotary speed of the engine crankshaft at the Maximum Power. The simulation fails if the engine speed exceeds the Maximum Engine Speed setting. Since the engine has rotating components, there exists resistance to changes in rotational speeds. As the engine speed varies, an inertia is experienced. The Initial Engine Speed signifies the speed of the engine at the start of the simulation. The Stall Speed signifies that, when the engine speed generated by the model falls below its value, the torque produced is blended to 0 Nm. The Speed Threshold signifies the minimum step change needed in the engine speed, in order to generate torque. Both the Stall Speed and Speed Threshold settings, yield 0 Nm of torque if violated, however, it helps distinguish between possible discrepancies in the simulation.

Mechanical rotational reference was used to connect mechanical rotational ports that are rigidly affixed to the vehicle frame. The torque converter is responsible for separating the torque from the generic engine and multiplying it to be sent through the driveline by a fluid coupling mechanism. The torque transmission time constant signifies the fixed time period taken to convert the input power to the impeller as output torque from the turbine. Simple gear box represents an ideal, non-planetary, fixed gear ratio gear box. The block generates torque in positive direction if a positive torque is applied to the input shaft and the gear ratio is assigned a positive value. It was adjusted to offer the maximum speed profiles for the test vehicles used. Differential block provides the bevel gear effects for the TMeasy tire system adopted. The vehicle body represents a two-axle vehicle body in longitudinal motion. It accounts for body mass, aerodynamic drag, road incline, and weight distribution between axles due to acceleration and road profile. The settings for mass, frontal area and friction coefficient were adjusted for each vehicle model as described in previous sections. The solver configuration block solves for acceleration at every time step in order to compute the speed versus time response graph.

From the simulated speed profiles, gain in kinetic energy against maximum engine speeds was studied and recorded in real-time. This phenomena was earlier defined in Equation 2.11. The gain in kinetic energy predicts the energy to be conserved during collision without the proposed speed adaptation model.

3.4.4.2 Development of Speed Monitoring and Adaptation Algorithm

An algorithm was developed to monitor two possible scenarios of how energy transforms with respect to gross vehicle mass and engine speeds. First, a focus was made on the problem statement stated by the inequality between gain in KE and average impact energy defined as $KE_C \leq$ Average Impact Energy. It was expected from this inequality that no gain in KE should exceed the set threshold for the average impact energy. This was achieved using to control equations for linear speed {v = u(t)} and gradual speed decay {dv/dt = -k(v-u)}. A nested IF-else statement was applied to related the two control equation to achieve the desired scenarios. Thereafter the algorithm was developed as illustrated:

u1 = kinetic energy, KE_C

u2 = total absorbed energy, $E_{average}$

1. If (u1 < u2)
2. The system outputs a linear motion sequence governed by v = u(t)
3. Else If (u1 = = u2)
4. The system outputs a constant motion sequence from the last speed reached by previous condition monitored by dv/dt = -k(v-u).
5. End

The algorithm was embedded in the speed monitoring and adaptation system to monitor the speeds against the inequality condition. Two outputs: adapted speed profile and gain in kinetic energy against the adapted speed limits are realized from the algorithm which are dependable on vehicle loading and monitored engine speeds in real-time.

3.4.4.3 Design of Speed Adaptation System using MATLAB®-Simulink platform

The next design of the proposed system was the speed monitoring and adaptation module given in Figure 8. It was to adapt speeds against time interval based on analysis of gained KE and average estimated impact energy value. The module had two main inputs namely: in kinetic energy defined in Equation 2.11 (variable parameter depended on monitored speed and weights) and average impact energy value defined in Equation (depended on force-deflection properties) as given in Equation 3.9.

For the system to simulate as required, the developed algorithm was executed using an IF Statement block modelling. The Statement utilised the variables KE_C and $E_{average}$ as inputs. The resultant output at the IF system block was fed to IF statement Subsystems blocks as shown in Figure 8. The IF subsystem blocks were also fed by respective inputs defined by Equation 3.11 and

$$\frac{dv}{dt} = -k(v-u) \qquad 3.12$$

where the k is a constant of proportionality to ensure an instantaneous decrease in speed to a new speed state. Equation 3.12 defines a motion sequence that gradually decays and maintains a constant sequence to infinity. The module monitors speed based on analysed kinetic energy and estimated average impact energy value. Such that when set condition is not met, a linear acceleration state is executed and when the set condition is met as a constant acceleration is executed. The final output was fed to a summation point since the designed model output two instances but only one has to be utilised at any given time.

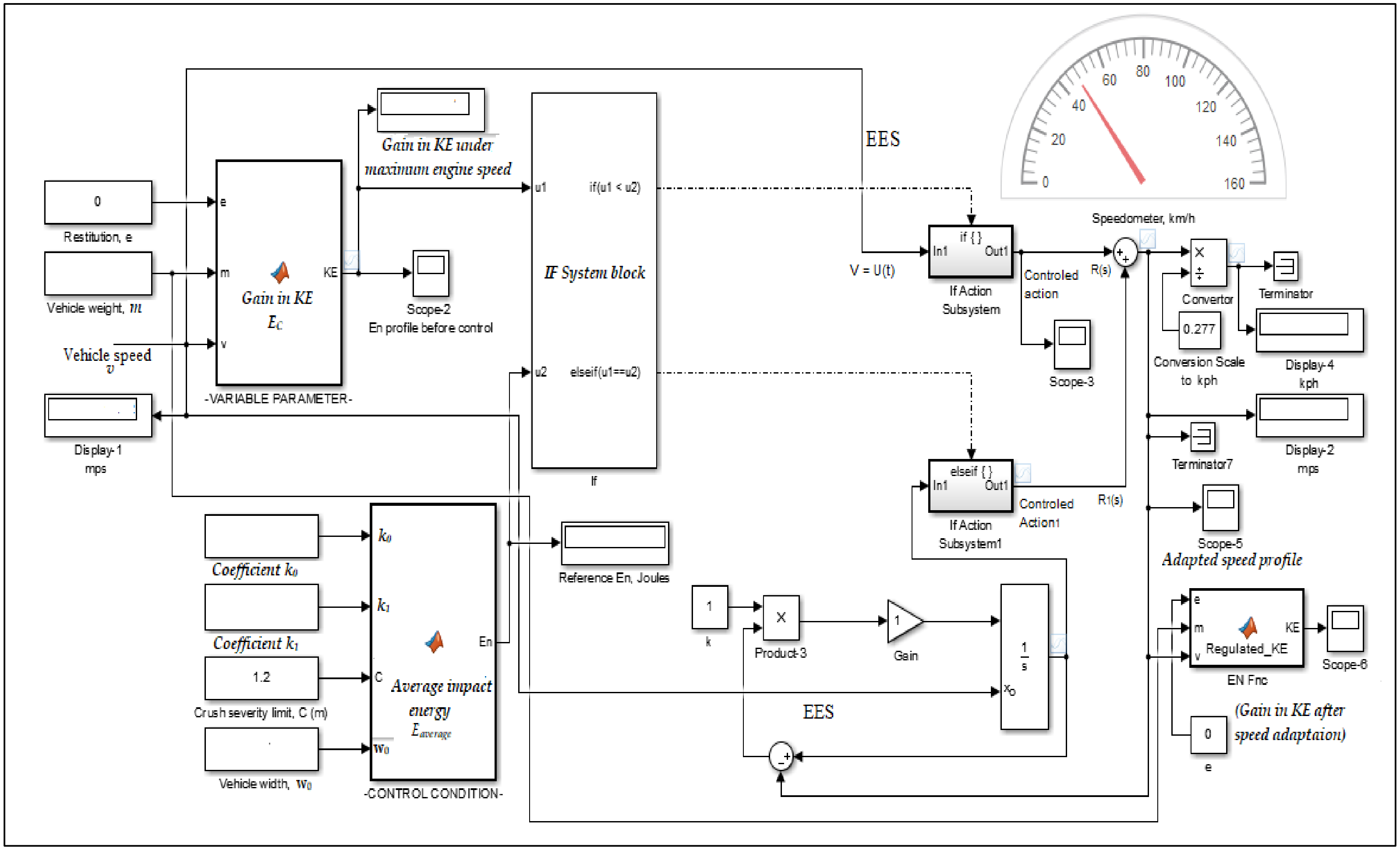

Figure 8: A Simulink speed adaptation system based on impact energy value

The Simulink system model in Figure 8, was designed to investigate the influence of impact energy values on vehicle speed adaptation. The model utilised input variable blocks for restitution, e, gross vehicle mass, m, collision severity, C, vehicle width, w_0 and crush stiffness coefficients k_0 and k_1. MATLAB® functional blocks were used to compute gain in kinetic energy and average impact energy value as given in Equation .

The gain in kinetic energy is a variable function of speed profiles and vehicle weights. The average impact energy value is a constant magnitude acting as a control reference that is vehicle specific. It is evaluated based on the force deflection properties, collision severity and vehicle width. The designed system model works using the computer algorithm described in the earlier section.

The If statement subsystem block instantaneously interprets the two inputs and the possible outputs. The output is fed to a summing point to give the net adapted speed limit based on the analysed impact energy. This new speed limit is the one to be fed in the convention speed governors so that the fuel injector valves or carburettor chambers are regulated to the new speed limit profile.

CHAPTER FOUR

RESULTS ANALYSIS AND DISCUSSION

4.1 Introduction

This chapter provides a descriptive review of the findings on the experimental work done and analysis of the data for the specific objectives. It outlines the final suggested solution to the problem statement with regards to the analysis of the collected data.

4.2 Analysing the Influence of Speed on Collision severity in Frontal impacts

Crash test data collected for the three vehicle models was presented in Table 4, Table 5 and Table 6.

Table 4: Barrier test data for Chevrolet Blazer LS 2000

Crash Test	1.	2.	3.	4.	5.	6.	7.	8.
Impact Speed (m/s)	5.510	6.796	8.063	9.558	10.633	12.324	13.497	14.087
Collision severity(m)	0.170	0.175	0.236	0.278	0.307	0.344	0.426	0.431

Table 5: Barrier test data for Chevrolet Corvette C6 Z06

Crash Test	1.	2.	3.	4.	5.	6.	7.	8.
Impact Speed (m/s)	6.153	6.790	8.060	9.558	10.632	12.324	13.495	14.085
Collision severity(m)	0.174	0.179	0.240	0.292	0.311	0.352	0.435	0.440

Table 6: Barrier test data for Chevrolet Crew cab Silverado 2003-7

Crash Test	1.	2.	3.	4.	5.	6.	7.	8.
Impact Speed (m/s)	6.140	6.788	8.055	9.544	10.624	12.316	13.487	14.077
Collision severity(m)	0.210	0.250	0.310	0.311	0.381	0.422	0.505	0.510

From these data, the study concluded the overall relationship between impact speeds and collision severity to be of linear form. This is depicted from the graphs in Figure 9, Figure 10 and Figure 11.

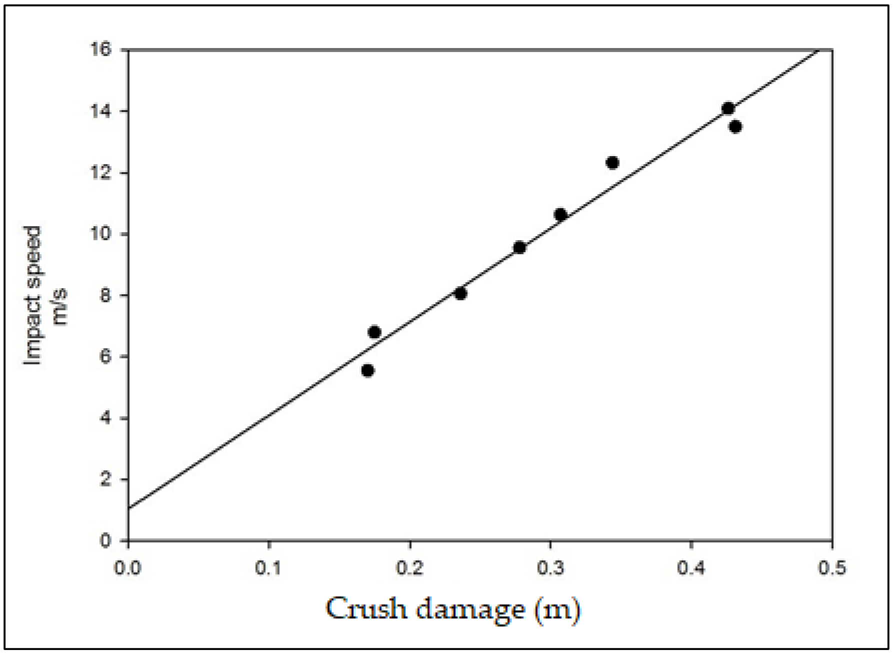

Figure 9: Speed versus Crush damage-Chevrolet Blazer LS 2000

The model equation for Figure 9 is a straight line given as

$$y = 30.48x + 1.05 \qquad \text{4.1}$$

where b_0 = 1.05, b_1 = 30.48 and x is the collision severity defined as crush damage. Equation 4.1 suggests a linear relationship between speed and collision severity. The coefficient 1.05 has units in m/s and defines the impact speed which produce no crush damage, while 30.48 has units in mps/m and is the slope of the graph representing statistical correlation of the data for the sample vehicle tested.

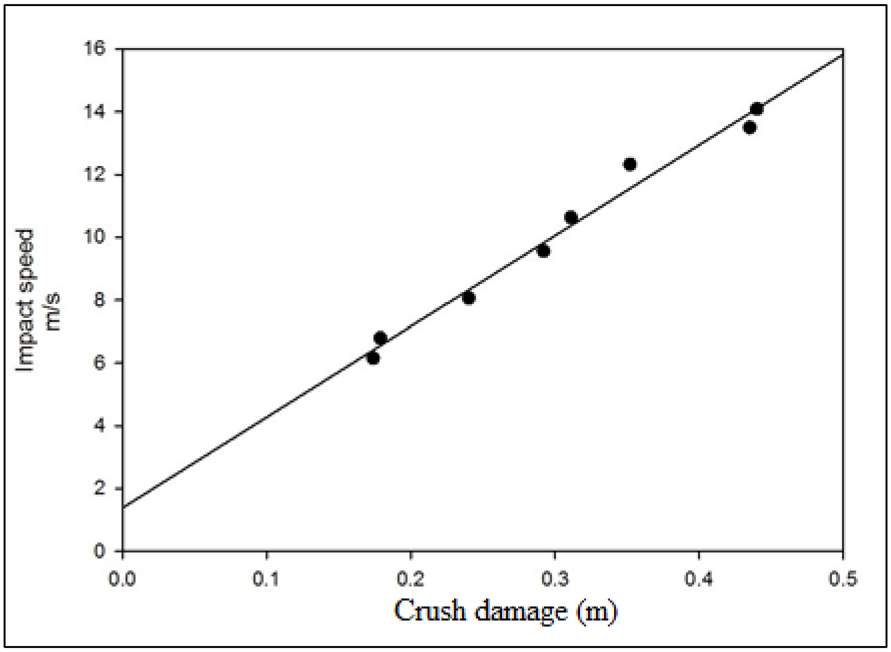

Figure 10: Seed versus Crush damage-Chevrolet Corvette C6-Z06

The model equation for the graph of Figure 10 was derived as

$$y = 28.90x + 1.36 \quad \text{.......} 4.2$$

where the coefficients b_1 = 28.90 mps/m and b_0 = 1.36 m/s are vehicle specific. Equation 4.2 takes a linear model as depicted in Equation 4.1. This indicates that impact speeds recorded for Chevrolet corvette C6-Z06 were directly proportionate to collision severity or crush damage registered for each crash test. The physical significance for coefficients 28.90 and 1.36 are similar to those depicted for Chevrolet Blazer LS 2000.

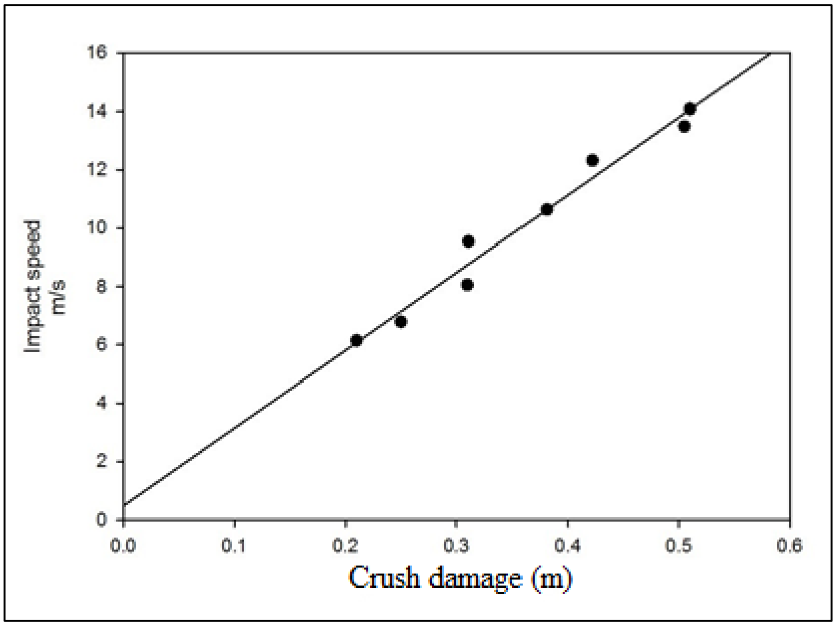

Figure 11: Speed versus Crush damage-Chevrolet Crew cab-Silverado 2003-7

The model equation for Figure 11 is a straight line of the form

$$y = 26.62x + 0.48 \qquad (4.3)$$

where 26.62 = b_1 and 0.48 = b_0. The units and physical significance depicted by the coefficients in Equation 4.3 are same as defined in Equations 4.1 and 4.2 for both Chevrolet Blazer LS 2000 and Corvette C6-Z06 models.

The graphs of Figure 9 - 11 are in clear agreement with the data presented from GM automobile crash tests. This data depicted that impact speed is linearly related to crush damage (collision severity). A general comparison of these crush coefficient constants for tested vehicle samples is provided in Table 7.

Table 7: Crush Constants from crash test simulations

Vehicle model	b_0 (ms^{-1})	b_1 (ms^{-1}/m)	C (m)
Chevrolet Blazer LS 2000	1.05	30.48	Varied with vehicle sample
Chevrolet Corvette C6-Z06	1.36	28.90	
Chevrolet Crew cab- Silverado 2003-7	0.48	26.62	

The coefficients b_1 and b_0 are seen to be vehicle specific and vary with vehicle categories, models and year of manufacture. It is therefore clear from these findings that collision severity can be used as a measure of accident severity levels. This is because crush damage levels increases with increase in impact speeds. From Equations 4.1, 4.2 and 4.3, the data for Figure 9, Figure 10 and Figure 11 can be summarised by a linear model equation of the form $v = b_0 + b_1C$ where v is impact speed in m/s, C is the collision severity in m, b_0 is the y-intercept in ms^{-1} and b_1 is the slope of the graphs in ms^{-1}/m.

The intercept b_0 is taken as the vehicle speed which produces no collision severity. From the test conducted, there was no data included at speeds below the y-intercept point. The values are obtained using graph extrapolation feature of the analysis software used. The slope, b_1 is taken to represent the data as precisely as possible over the range of data. From Figure 9, Figure 10 and Figure 11, it is clear that the linear approximation is valid over the range of data obtained for each vehicle model.

Data of this kind relating vehicle speed and collision severity in frontal crash tests can be described using crush stiffness coefficients b_0 and b_1 and the vehicle weights at which the coefficients were determined. Applying this knowledge in vehicle safety systems will help advance implementation of active safety devices (Figueiredo *et al.,* 2001). These findings concluded the first specific objective of the study.

4.3 Evaluating the Influence of Impact energy on Collision severity using the Principle of Energy Conservation

The second aim of this study, focused on establishing a relationship between impact energy and collision severity. To do so it was necessary to examine the relationship between kinetic energy and energy spent on instantaneous crush depth. In its simplest form impact energy is defined from kinetic energy. This is conserved as the energy lost during collision called instantaneous crush energy factor, $E^* = d_0 + d_1 C$ related to the collision severity as suggested by (Brach *et al.*, 2007) where d_0 and d_1 experimentally defines crush stiffness coefficients and C is collision severity. This energy factor can also be defined as a square root in terms of conserved KE and vehicle width as

$$E^* = \sqrt{\frac{2E_i}{w}} \quad \text{.......................................} 4.4$$

where E_i is the impact energy expended by the damage process equivalent to conserved KE_c, w is the width of the crush and E^* is the energy factor root describing the energy lost to instantaneous crush damage. From Equation 4.4, tabular analysis was done on the data obtained for impact energy expended in each crash test experiments. This is given in Table 8, Table 9 and Table 10.

Table 8: Energy versus Crush damage-Chevrolet Blazer LS 2000

Crash test	Kinetic energy, E_C (J)	Crush energy, E* (J)	Crush damage (m)
1.	40782.00	206.649	0.170
2.	61134.66	253.013	0.175
3.	86038.14	300.154	0.236
4.	120863.07	355.750	0.278
5.	149571.65	395.752	0.307
6.	200949.92	458.714	0.344
7.	240917.54	502.265	0.426
8.	263447.01	525.224	0.431

Table 9: Energy versus Crush damage-Chevrolet Corvette C6 Z06

Crash test	Kinetic energy, E_C (J)	Crush energy, E* (J)	Crush damage (m)
1.	50102.73	233.336	0.174
2.	61133.90	257.779	0.179
3.	86037.11	305.808	0.240
4.	120861.64	362.452	0.292
5.	149569.90	403.207	0.311
6.	200947.58	467.356	0.352
7.	240914.77	511.726	0.435
8.	262447.00	543.105	0.440

Table 10: Energy versus Crush damage-Chevrolet Crew cab Silverado 2003-7

Crash test	Kinetic energy, E_C (J)	Crush energy, E* (J)	Crush damage (m)
1.	49967.81	226.383	0.210
2.	61054.55	250.240	0.250
3.	85962.53	296.929	0.310
4.	120644.61	357.764	0.311
5.	149478.88	391.550	0.381
6.	200846.62	453.868	0.422
7.	240846.25	497.013	0.505
8.	262373.82	518.750	0.510

The data from Table 8, Table 9 and Table 10 was interpreted using graphs of E^* versus C as depicted in the Figure 12, Figure 13 and Figure 14.

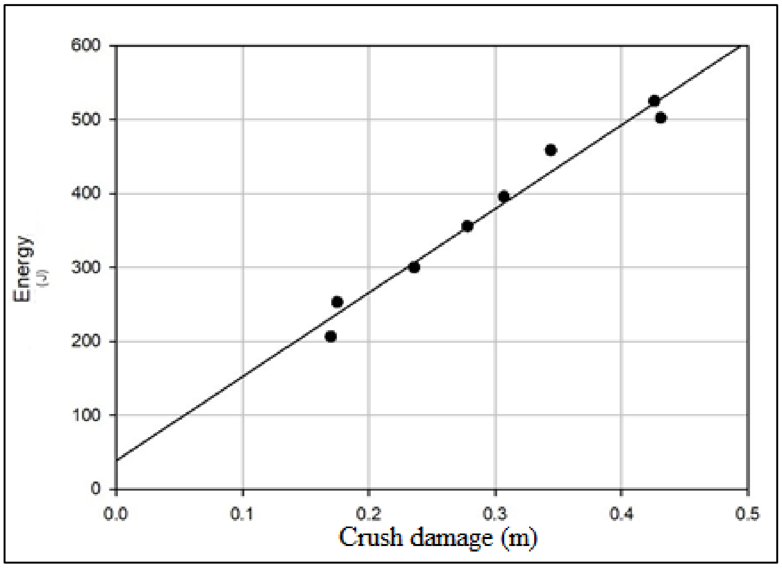

Figure 12: Energy versus Crush damage-Chevrolet Blazer LS 2000

The graph obtained in Figure 12 takes linear model of the form y = 1135.78x + 38.63 where y is the energy factor root (E*), x is the collision severity along x-axis, m and c are stiffness coefficient constants representing d_1 and d_0 respectively for the vehicle under test.

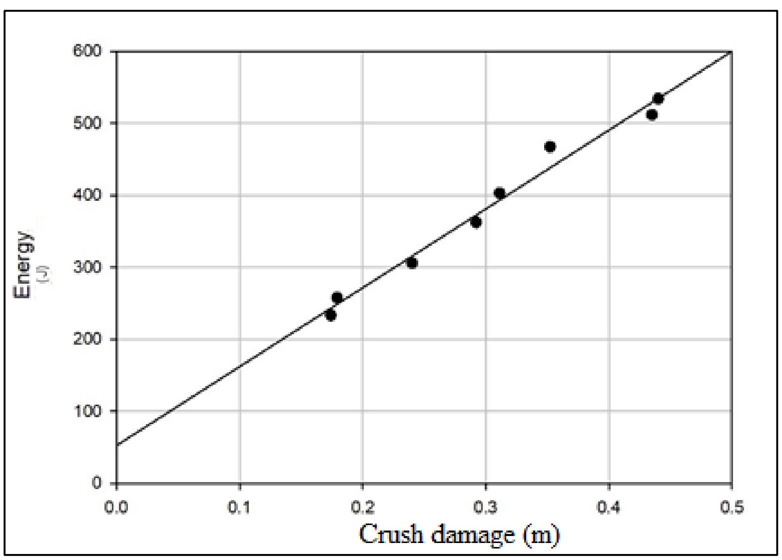

Figure 13: Energy versus Crush damage-Chevrolet Corvette C6 Z06

The graph obtained in Figure 13 takes linear model of the form: $y = 1095.56x + 52.66$ where y is the energy factor root (E^*), x is the collision severity along x-axis, m and c are stiffness coefficient constants equivalent to those obtained for Chevrolet Blazer LS 2000.

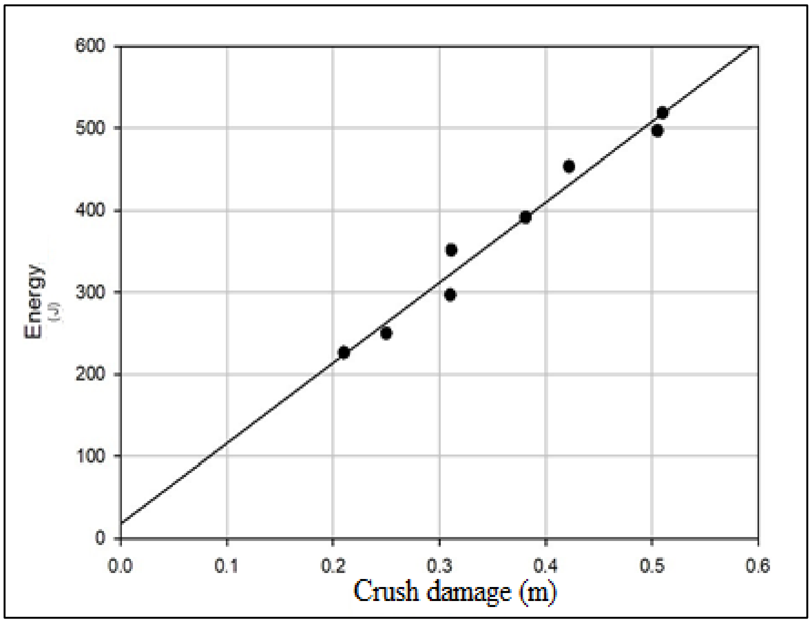

Figure 14: Energy versus Crush damage-Chevrolet Crew cab Silverado 2003-7

The graph obtained in Figure 14 takes linear model of the form $y = 980.70x + 17.93$ where y is the crush energy (E^*), x is the collision severity along x-axis, m and c are crush constants as defined for Chevrolet Blazer and corvette models.

The graphs defined in Figure 12, Figure 13 and Figure 14 obeys a straight line equation. Of which y is square root of crush energy per width, c is the y-intercept denoting the energy with no collision severity defined as d_0 and m is the slope of the graphs depicting statistical fitness over the data range collected defined as d_1. The coefficients c and m from the data clearly indicates they are vehicle specific and depend on the specific region of crush like front, rear or side, year of manufacture, country of origin and vehicle body structure.

This graphs clearly indicates impact energy expended and collision severity is directly proportional. This implies that a slight change in KE will have proportionate change in collision severity. This is fatal in case the energy change is highly pronounced and therefore calls for an action in terms of energy regulation prior to collisions (Khorasani-Zavareh *et al.*, 2015).

From Equation 2.7, conserved KE was defined with respect to crash dynamic property called restitution, e as

$$KE_C = \frac{1}{2}(1-e^2)\cdot m\cdot v^2 \quad \text{...}4.5$$

where KE_c is the impact energy E_i which defines the energy loss of the bullet vehicle in motion, m is gross vehicle mass and v is vehicle speed.

4.4 Characterising of force-deflection properties in frontal impact energy based on work-energy theorem

Form first principles, utilising work-energy principle, the study was able to deduce a model equation that relates the influence of impact force on average full frontal impact energy. This took into consideration the collision severity resulting due to impact energy. A linear relationship is observed to exist for the impact force (F) and collision severity defined as $F = k_0 + k_1C$. This implies that a slight increase in the impact force has direct linear consequences on collision severity. This is due to the fact that absorbed energy and force are directly related as suggested from the work-energy principle.

Therefore, taking Equation as the reference model, a solution can be devised towards monitoring vehicle speeds based on the crush energy, E_C and average frontal impact energy, $E_{average}$. This average impact energy varies depending on vehicle structure as justified by the analysis given in Figure 15, Figure 16 and Figure 17.

Table 11: Force versus Crush damage-Chevrolet Blazer LS 2000

Crash test	Impact force, F (N)	Crush damage (m)
1.	120827.60	0.170
2.	123784.93	0.175
3.	159864.38	0.236
4.	184705.98	0.278
5.	201858.50	0.307
6.	223742.76	0.344
7.	275200.35	0.426
8.	272243.01	0.431

Table 12: Force versus Crush damage-Chevrolet Corvette C6 Z06

Crash test	Impact force, F (N)	Crush damage (m)
1.	147576.71	0.174
2.	150914.59	0.179
3.	191636.69	0.240
4.	226350.61	0.292
5.	239034.55	0.311
6.	266405.14	0.352
7.	321813.90	0.435
8.	325151.78	0.440

Table 13: Force versus Crush damage-Chevrolet Crew cab Silverado 2003-7

Crash test	Impact force, F (N)	Crush damage(m)
1.	256884.64	0.210
2.	301945.88	0.250
3.	369537.75	0.310
4.	370664.28	0.311
5.	449521.46	0.381
6.	495709.23	0.422
7.	589211.31	0.505
8.	594843.97	0.510

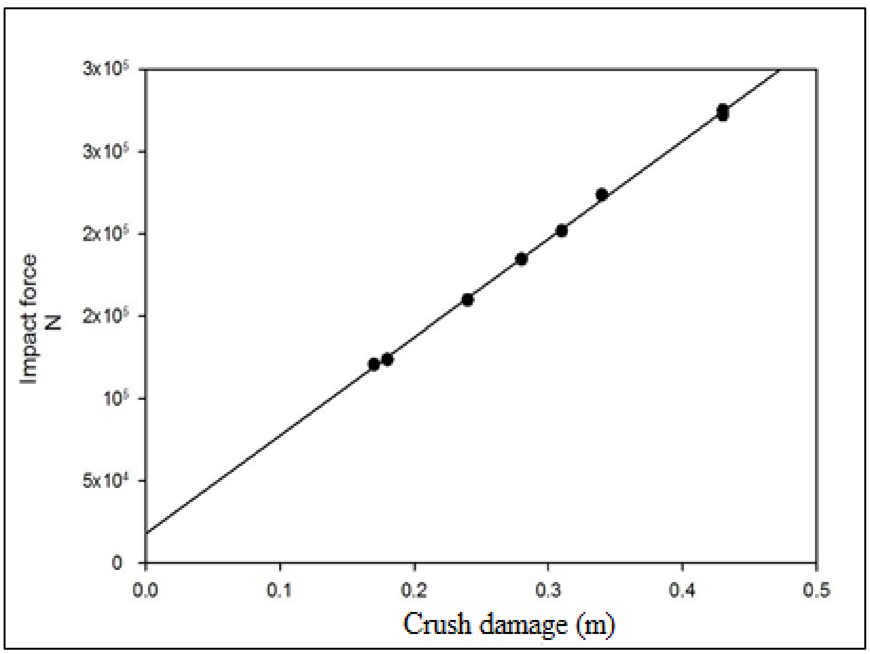

Figure 15: Force versus Crush damage-Chevrolet Blazer LS 2000

The model equation for the graph in Figure 15 is linear taking the form of

$$F = k_0 + k_1 C \text{ ....} 4.6$$

where k_0 is the impact force per width in N/m which produces no collision severity, k_1 is the slope of the graph which quantifies the statistical validity of the data over the sampled data in N/m^2 and C is collision severity in meters. The two coefficients are defined as force-deflection properties and are vehicle specific. This indicates that impact force and hence force-deflection properties directly influence the collision severity.

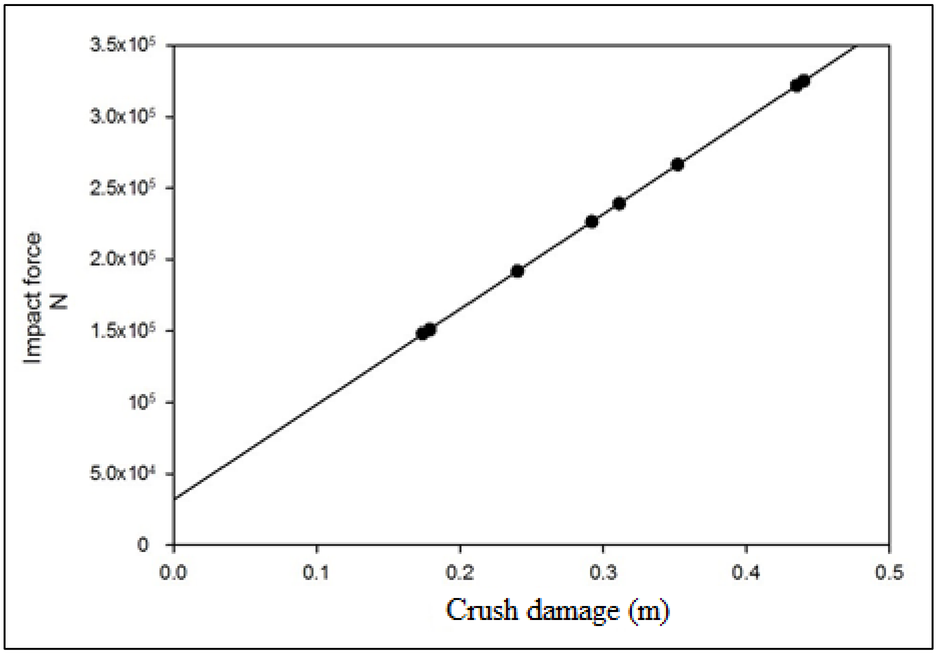

Figure 16: Force versus Crush damage-Chevrolet Corvette C6 Z06

The model equation for the graph in Figure 16 is linear taking the form of Equation 4.6. This similarity maybe attributed to use of same vehicle brand or general vehicle crash dynamic properties.

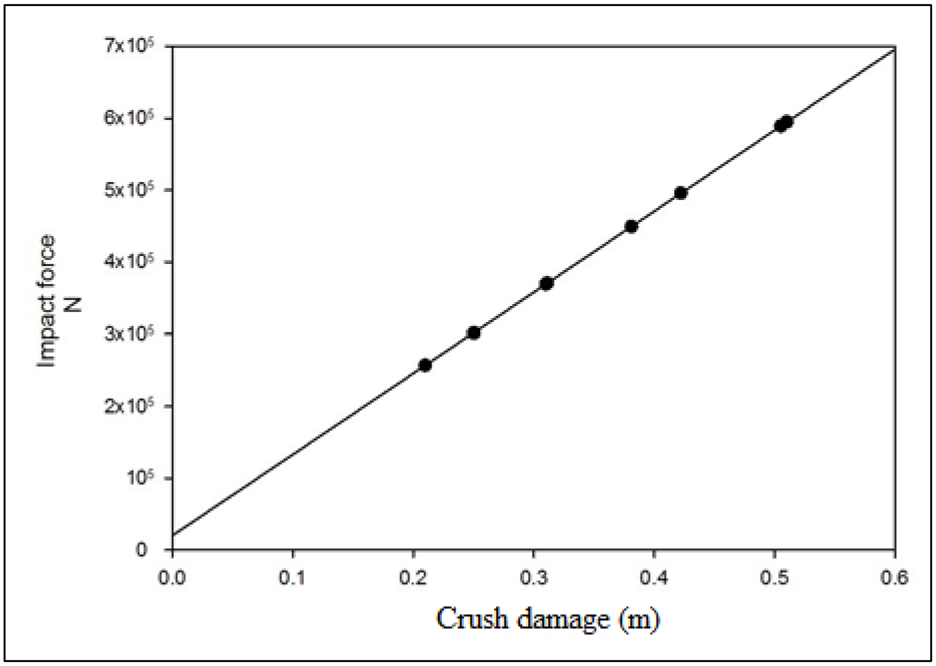

Figure 17: Force versus Crush damage-Chevrolet Crew cab Silverado 2003-7

The model equation for the graph in Figure 17 is linear taking the form of Equation 4.6. The similarity of the linear model is depicted by coefficients k_0 and k_1 which are varying for each vehicle sample used and that all experiments were performed under same conditions. Finding the area under the graphs of Figure 15, Figure 16 and Figure 17, it was deduced that the average impact energy in full frontal impacts can be defined with respect to collision severity and force-deflection properties. This is shown in Equation which gives the value of KE transferred during collisions from Equation 4.5 since energy is conserved in its physical state. Using both Equation 3.1 and 4.5, an inequality relating gain in KE at maximum engine speed and average impact energy was defined as

$$K.E \leq E_{average} \quad \ldots\ldots 4.7$$

where the parameters in the inequality are as defined in previous sections for respective factors. From Equation 4.7 the study focused on achieving our final objective as discussed in section 4.5.

4.5 Investigating the Influence of Impact Energy on Intelligent Speed Adaptation using MATLAB®-Simulink platform

The study was able to establish a relationship between vehicle weights, vehicle speeds, crush stiffness coefficients and collision severity as provided in Equations 3.1, 4.5 and 4.7. Using this knowledge, the study went on to investigate the influence of frontal impact energy as a basis of collision severity on intelligent speed adaptation. A Simulink® system model was designed using the derived model equations to show the effects of using reference vehicle weights, collision severity and crush coefficients on speed monitoring.

The designed system model is provided in Figure 8 as well as a sample vehicle model in Figure 7 to provide the input speed profiles. By varying the engine power to suite the engine specifications for each vehicle model the results in Table 14, Table 15 and Table 16 were obtained.

Table 14: Speed adaptation profiles for Chevrolet crew cab Silverado 2003-7

Vehicle weights (kg)		Maximum engine speed (ms^{-1})	Actual gain in KE (J)	Adapted speed limit (ms^{-1})	Reference KE (J)
Curb weight (CW)	2485	43.889	2405480	32.16	1285024.01
¼ Load + CW	2907	43.889	2813980	29.73	1285024.01
½ Load + CW	3324	43.889	3217630	27.78	1285024.01
¾ Load + CW	3751	43.889	3630970	26.17	1285024.01
Full load + CW	4173	43.889	4039460	24.82	1285024.01

Table 15: Speed adaptation profiles for Chevrolet Corvette C6 Z06

Vehicle weights (kg)		Maximum engine speed (ms^{-1})	Actual gain in KE (J)	Adapted speed limit (ms^{-1})	Reference KE (J)
Curb weight (CW)	1420	92.53	6078880	31.29	695298.078
Full load + CW	1589	92.53	6802350	29.58	695298.078

Table 16: Speed adaptation profiles for Chevrolet Blazer LS 2000

Vehicle weights (kg)		Maximum engine speed (ms^{-1})	Actual gain in KE (J)	Adapted speed limit (ms^{-1})	Reference KE (J)
Curb weight (CW)	1825	44.44	1823260	28.97	765916.10
¼ Load + CW	1975	44.44	1973110	27.85	765916.10
½ Load + CW	2125	44.44	2122970	26.84	765916.10
¾ Load + CW	2275	44.44	2272830	25.95	765916.10
Full load + CW	2425	44.44	2422680	25.13	765916.10

Graphical analysis was made on energy and speed profiles for different vehicle samples at different weights. From Table 14, Table 15 and Table 16 it can be deduced that the proposed model was able to adapt vehicle speeds with respect to the set impact energy value under set collision severity index. The impact energy value was pre-set for each vehicle as given in Equation . The actual gain in kinetic energy against maximum engine speeds illustrates the possible scenario when the full speed profile is utilised. Whereas, adapted speed limit illustrates a scenario when an impact energy value is included to limit speeds depending on real-time vehicle loads and monitored speeds.

The oscilloscope graphs in Figure 18 and Figure 19 shows the speed profiles exhibited before and after application of the proposed system model.

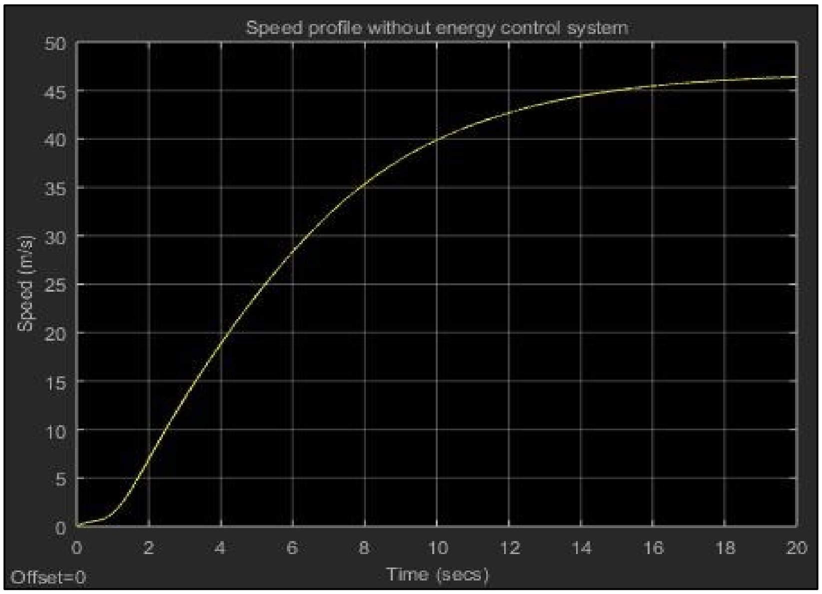

Figure 18: Speed profile without inclusion of impact energy value

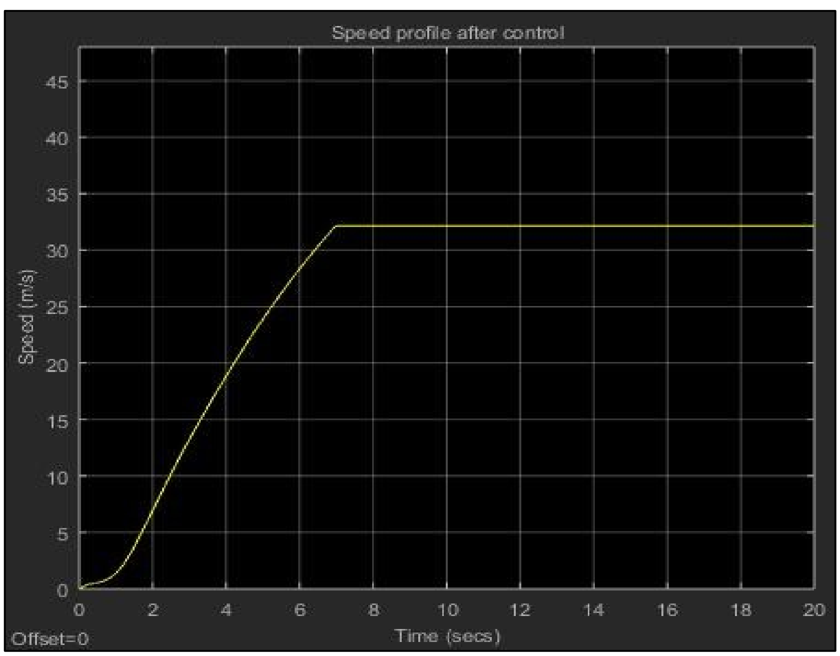

Figure 19: Speed profile with inclusion of impact energy value

Form this graphs the expected scenarios when the suggested algorithm is utilised is as shown in Figure 19 for adapted speed limit. This translates to a new KE profile given in Figure 21. For example, the Figure 19 shows a new speed limit adapted by the system for Chevrolet Crew cab Silverado 2003-7 under curb weight specifications with negligible weight of the driver. The limit however changes with respect to the vehicle net weight as depicted Table 14. Similar speed adaptation characteristics were recorded for both Corvette C6 Z06 and Blazer LS 2000 shown in Table 15 and Table 16 respectively for comparative analysis.

Likewise the control system depicts a scenario when energy is not subjected to a reference factor as shown in Figure 18 where the vehicle attained maximum engine speed and hence maximum KE which translates to impact energy at crash as given in Figure 20.

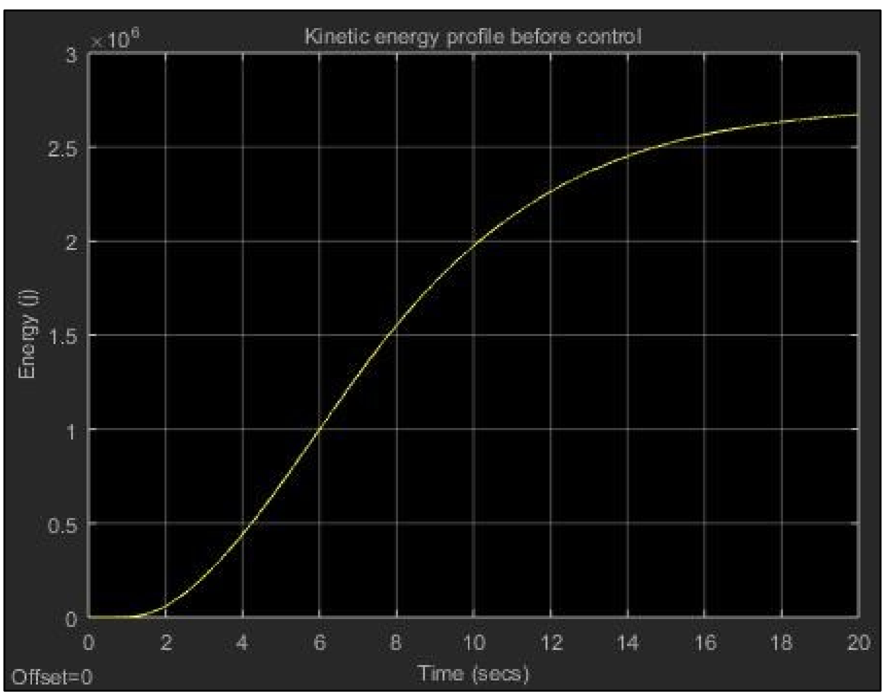

Figure 20: Gain in kinetic energy without the proposed speed control algorithm

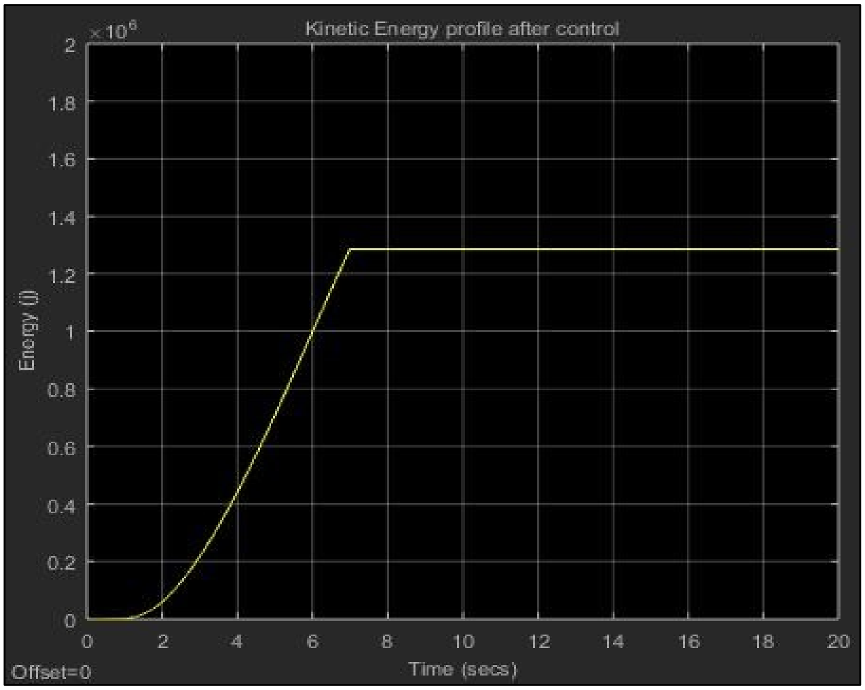

Figure 21: Gain in kinetic energy based on the proposed speed control algorithm

Figure 21 was obtained for gain in kinetic energy after application of designed system model which clearly justifies the need to adapt the engine speeds with respect to impact energy magnitudes for there to be admissible collision severity during vehicle collisions accidents.

The values for adapted speed limits varied according to the total vehicle weights at any instant with a constant reference impact energy value given by Equation . These findings concluded our final objective on investigating how intelligent speed adaptation can be achieved through analysis of full frontal impact energy as basis of collision severity.

CHAPTER FIVE

CONCLUSION AND RECOMMENDATION

5.1 Introduction

The study aimed at analysing the influence of impact energy as a basis of collision severity in vehicle accidents relating to collision and its influence on intelligent speed adaptation.

5.2 Conclusion

From first principles energy is a function of both vehicle weight and speeds. Likewise, energy can be related to force-deflection properties based on work energy principle as found in this study. Using the adopted methodology, we prepared crash test data, analysed and extracted relevant relation models needed during the study. Further experiments to study the influence of various vehicle dynamic parameters on crush energy and collision severity during accidents were conducted. From this experiments, an inequality was derived to ascertain the need for using estimated impact energy value in speed adaptation as shown in Equation 4.7. A speed adaptation algorithm was developed using this inequality for our final objective of the study which was executed using the designed Simulink™ control system in Figure 8. The system was used to analyse the influence of full frontal impact energy on speed monitoring and adaptation mechanism since speed influences collision magnitude. An investigation was done on possible effects of adapting vehicle speed with respect to real time vehicle weights and speeds as depicted in from Figure 18 - Figure 21.

It was also observed that impact speed has a direct influence on the collision severity inflicted during full frontal impacts given as $v = b_0 + b_1C$. These coefficients are seen to vary for the vehicle tested and this is attributed to vehicle body structural designs, country of origin and year manufacture. The b_0 and b_1 coefficients are later used to characterise force-deflection properties based on work energy theorem as high lightened in this study. Furthermore, impact energy taken as a function of both vehicle speeds and vehicle weights monitored in real-time was observed to have a direct influence on collision severity inflicted in full frontal impacts expressed as root factor by $E^{*} = d_0 + d_1 C$. This energy factor root relates to conserved KE defined in Equation 4.5. The coefficients d_0 and d_1 were found to vary for each vehicle sample tested as given in the data discussion section. Lastly, it was shown that impact energy can be estimated using force-deflection properties and crush damage (collision severity) as stated in Equation for frontal impacts.

It was concluded that using an average impact energy value, vehicle speeds can be monitored in real time and adapted to limits estimated to result in admissible collision severity in full frontal impacts. This is illustrated by our findings in Figure 18, Figure 19, Figure 20 and Figure 21 for the sample vehicles models. The overall effect of impact energy magnitudes on intelligent speed adaptation was summarised in Table 14, Table 15 and Table 16 for the vehicle samples used. It was observed that the system was able to limit speed profiles using the analysed KE with respect to suggested collision severity. By so doing, the vehicle will be limited within a momentum threshold which the body structure can withstand in case of vehicle collision accidents. This will result in tolerable collision severity in vehicle collision accidents.

5.3 Recommendation

For further research in this area, more concern should be on establishing a concrete approach of estimating impact energy using crush coefficients k_0 and k_1. This is because the data collected was from virtual simulation that need further validation against other test vehicle models like Toyota, Nissan, Volkswagen etc. The approach is also considerate of specific vehicles and hence more data for coefficients b_0 and b_1 is needed to effectively implement the suggested method for all vehicle categories i.e. compact, subcompact, intermediate and full size across all vehicle brands. Crush damage was chosen as a measure of collision severity but possess limitations if modern vehicle occupant safety systems is mentioned e.g. crash zones for impact energy absorption. Hence more work is needed to justify the use of crush damage as an estimate in this study.

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APPENDICES

APPENDIX I: Full Frontal Crash Test Data for Sampled Test Vehicles

Table A$_1$: Crush damage versus Impact Speeds-Chevrolet Blazer LS 2000

Crash Test	Crush damage m	Acceleration. m/s^2	EES km/h	Test speed km/h	Pre Impact speed km/h	Post Impact speed km/h	Impulse N.s	Kinetic Energy J
1	0.170	1.500	19.982	20.000	43.789	23.677	7513.700	40782.000
2	0.175	1.500	24.466	30.000	52.670	27.553	9338.900	61134.660
3	0.236	1.500	29.028	45.000	62.480	32.689	11082.260	86038.140
4	0.278	1.500	34.407	60.000	74.045	38.741	13139.400	120863.068
5	0.307	1.500	38.277	70.000	82.366	43.093	14618.980	149571.652
6	0.344	1.500	44.368	85.000	95.458	49.940	16950.300	200949.915
7	0.431	1.500	48.588	95.000	104.491	54.658	18573.000	240917.540
8	0.426	1.500	50.714	100.000	109.491	57.047	19385.000	263447.010

Table A_2: **Crush damage versus Impact Speeds (Chevrolet Corvette C6 Z06)**

Crash Test	Crush damage m	Acceleration. m/s^2	EES km/h	Test speed km/h	Pre Impact speed km/h	Post Impact speed km/h	Impulse N.s	Kinetic Energy J
1	0.174	1.500	22.149	20.000	47.682	24.944	8454.200	50102.730
2	0.179	1.500	24.466	30.000	52.670	27.554	9338.800	61133.900
3	0.240	1.500	29.028	45.000	62.480	32.689	11082.200	86037.107
4	0.292	1.500	34.407	60.000	74.045	38.741	13139.300	120861.641
5	0.311	1.500	38.276	70.000	82.366	43.094	14618.800	149569.896
6	0.352	1.500	44.368	85.000	95.458	46.090	16950.180	200947.577
7	0.435	1.500	48.583	95.000	104.491	54.659	18572.830	240914.770
8	0.440	1.500	50.707	100.000	109.059	57.047	19385.900	262447.000

Table A3: Crush damage versus Impact Speeds (Chevrolet Crew cab Silverado 2003-7)

Crash Test	Crush damage m	Acceleration. m/s^2	EES km/h	Test speed km/h	Pre Impact speed km/h	Post Impact speed km/h	Impulse N.s	Kinetic Energy J
1	0.21	1.500	22.104	20.000	47.656	24.964	8434.400	49967.814
2	0.250	1.500	24.436	30.000	52.670	27.586	9326.400	61054.554
3	0.310	1.500	28.997	45.000	62.480	32.716	11072.700	85962.534
4	0.311	1.500	34.358	60.000	74.020	38.760	13117.400	120644.614
5	0.381	1.500	38.247	70.000	82.366	43.120	14610.800	149478.875
6	0.422	1.500	44.337	85.000	95.458	49.967	16942.900	200846.615
7	0.505	1.500	48.553	95.000	104.491	54.678	18569.500	240846.245
8	0.510	1.500	50.677	100.000	109.059	57.068	19382.600	262373.816

APPENDIX II: Full Frontal Crash Tests data of Impact Force versus Collision severity

Table B_1: Crush damage versus Impact Force-Chevrolet Blazer LS 2000

Crash Test	Crush damage m	b_0 m/s	b_1 ms^{-1}/m	Width m	k_0 N/m	k_1 N/m^2	G N	Force N
1	0.17	1.05	30.48	1.91	20278.30	591466.47	347.62	120827.60
2	0.18	1.05	30.48	1.91	20278.30	591466.47	347.62	123784.93
3	0.24	1.05	30.48	1.91	20278.30	591466.47	347.62	159864.38
4	0.28	1.05	30.48	1.91	20278.30	591466.47	347.62	184705.98
5	0.31	1.05	30.48	1.91	20278.30	591466.47	347.62	201858.50
6	0.34	1.05	30.48	1.91	20278.30	591466.47	347.62	223742.76
7	0.43	1.05	30.48	1.91	20278.30	591466.47	347.62	275200.35
8	0.43	1.05	30.48	1.91	20278.30	591466.47	347.62	272243.01

Table B_2: Crush damage versus Impact force (Chevrolet Corvette C6 Z06)

Crash Test	Crush damage m	b_0 m/s	b_1 ms^{-1}/m	Width m	k_0 N/m	k_1 N/m^2	G N	Force N
1	0.174	1.36	28.90	1.84	31418.58	667575.46	739.34	147576.71
2	0.179	1.36	28.90	1.84	31418.58	667575.46	739.34	150914.59
3	0.240	1.36	28.90	1.84	31418.58	667575.46	739.34	191636.69
4	0.292	1.36	28.90	1.84	31418.58	667575.46	739.34	226350.61
5	0.311	1.36	28.90	1.84	31418.58	667575.46	739.34	239034.55
6	0.352	1.36	28.90	1.84	31418.58	667575.46	739.34	266405.14
7	0.435	1.36	28.90	1.84	31418.58	667575.46	739.34	321813.90
8	0.440	1.36	28.90	1.84	31418.58	667575.46	739.34	325151.78

Table B_3: Crush damage versus Impact force (Chevrolet Crew Cab Silverado 2003-7)

Crash Test	Crush damage m	b_0 m/s	b_1 ms^{-1}/m	Width m	k_0 N/m	k_1 N/m^2	G N	Force N
1	0.210	0.48	26.62	1.95	20313.11	1126531.10	183.14	256884.64
2	0.250	0.48	26.62	1.95	20313.11	1126531.10	183.14	301945.88
3	0.310	0.48	26.62	1.95	20313.11	1126531.10	183.14	369537.75
4	0.311	0.48	26.62	1.95	20313.11	1126531.10	183.14	370664.28
5	0.381	0.48	26.62	1.95	20313.11	1126531.10	183.14	449521.46
6	0.422	0.48	26.62	1.95	20313.11	1126531.10	183.14	495709.23
7	0.505	0.48	26.62	1.95	20313.11	1126531.10	183.14	589211.31
8	0.510	0.48	26.62	1.95	20313.11	1126531.10	183.14	594843.97

APPENDIX III: Frontal Crash Test data presented by General Motors Corporation

Test vehicle model	Standard weight (lbs)	Width w_0 (Inches)	b_0 (mph)	b_1 (mph/in)	A (Lb./Inch)	B (Lb./in^2)	G Lb.
71-72 Std. Full size	4500	79.2	6.85	0.85	274.6	35.27	1068.6
73-74 Std. Full size	4500	79.2	7.5	0.90	307.5	36.89	1281.1
73-74 Intermediate	4000	76.8	7.5	0.90	281.8	33.82	1174.3
71-74 Compact	3400	71.4	3.0	1.35	154.6	69.57	171.78
71-74 Subcompact	2500	62.2	3.0	1.35	130.5	58.72	144.94