The Behaviour of Rollover Protective Structures subjected to Static and Dynamic Loading Conditions

By

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Rollover Protective Structures, ROPS, Safety, Occupant protection, Impact, Energy absorption, Destructive testing, Dynamic loading, Impulse loads, Finite Element Analysis (FEA), Energy Absorption Enhancement
ABSTRACT

The Rollover of heavy vehicles operating in the construction, mining and agricultural sectors is a common occurrence that may result in death or severe injury for the vehicle occupants. Safety frames called ROPS (Rollover Protective Structures) that enclose the vehicle cabin, have been used by heavy vehicle manufacturers to provide protection to vehicle occupants during rollover accidents. The design of a ROPS requires that a dual criteria be fulfilled that ensures that the ROPS has sufficient stiffness to offer protection, whilst possessing an appropriate level of flexibility to absorb some or most of the impact energy during a roll.

Over the last four decades significant research has been performed on these types of safety devices which has resulted in the generation of performance standards that may be used to assess the adequacy of a ROPS design for a particular vehicle type. At present these performance standards require that destructive full scale testing methods be used to assess the adequacy of a ROPS. This method of ROPS certification can be extremely expensive given the size and weight of many vehicles that operate in these sectors. The use of analytical methods to assess the performance of a ROPS is currently prohibited by these standards. Reasons for this are attributed to a lack of available fundamental research information on the nonlinear inelastic response of safety frame structures such as this.

The main aim of this project was to therefore generate fundamental research information on the nonlinear response behaviour of ROPS subjected to both static and dynamic loading conditions that could be used to contribute towards the development of an efficient analytical design procedure that may lessen the need for destructive full scale testing. In addition to this, the project also aspired to develop methods for promoting increased levels of operator safety during vehicle rollover through enhancing the level of energy absorbed by the ROPS. The methods used to fulfil these aims involved the implementation of an extensive analytical modelling program using Finite Element Analysis (FEA) in association with a detailed experimental testing program. From these studies comprehensive research information was developed on both the dynamic impact response and energy absorption capabilities of these types of structures. The established finite element
models were then used to extend the investigation further and to carry out parametric studies. Important parameters such as ROPS post stiffness, rollslope inclination and impact duration were identified and their effects quantified.

The final stage of the project examined the enhancement of the energy absorption capabilities of a ROPS through the incorporation of a supplementary energy absorbing device within the framework of the ROPS. The device that was chosen for numerical evaluation was a thin walled tapered tube known as frusta that was designed to crush under a sideways rollover and hence lessen the energy absorption demand placed upon the ROPS. The inclusion of this device was found to be beneficial in absorbing energy and enhancing the level of safety afforded to the vehicle occupants.
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The work contained in this thesis has not been previously submitted for a degree or diploma at any other higher education institution. To the best of my knowledge and belief, the thesis contains no material previously published or written by another person except where due reference is made.

Signed:  __________________________________________

Date:  __________________________________________
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Chapter 1 Introduction

Heavy vehicles which are commonly used in the rural, mining and construction industries are particularly susceptible to rollovers as they possess a high centre of gravity and commonly operate on sloping and uneven terrain. In recent times fatal vehicle upsets associated with off-road machinery have been recognised as a key problem needing attention and many methods to prevent vehicle overturns have been investigated and trialled. A safety frame known as a rollover protective structure (ROPS) has emerged as the most suitable method for affording protection to the operators of such vehicles during rollovers. The role of this safety device is to absorb some of the kinetic energy of the rollover, whilst maintaining a survival zone for the operator. Three common forms of ROPS are commonly used to meet this purpose for most types of heavy machinery, namely: a two post cantilever ROPS, a two post rollbar ROPS and a four post ROPS. Figure 1-1 shows an illustration of a typical four post ROPS that is fitted to an earthmoving machine. The design and analysis of these types of structures is complex and requires that a dual criterion be established that ensures that they are flexible enough to absorb energy, whilst being rigid enough to maintain a survival zone around the operator.

Figure 1-1 Typical Four Post ROPS
1.1 Implications for Australia

Galloway (1998) targeted accidents associated with rollovers of tractors on farms as the largest killer of people working with heavy machinery. She suggested that “nearly nine out of every ten people killed on farms died as a result of accidents involving tractors. Of these, rollovers, where drivers or passengers are crushed beneath the tractor, is the most common type of accident.” (Galloway, 1998) In addition to this Day (1999) indicated that in the state of Victoria between the years of 1992 and 1996, tractors had accounted for 61% of farm work related fatalities of which 53% were attributed to accidents involving rollovers. Based on these statistics, it is evident that rollover accidents occur frequently and have resulted in significant loss of human lives as well as considerable economic loss to both the government and private sectors. In most cases injury may well have been avoided if the vehicles had been fitted with an adequately designed ROPS. Figure 1-2 and Figure 1-3 present graphic illustrations of heavy vehicles rollovers that have occurred on mining sites within Australia in recent years. It is evident from these figures that significant damage is sustained by these types of vehicles during a rollover and that the implementation of a protective structure in the form of a ROPS is vital to the operator’s chances of survival.

This figure is not available online. Please consult the hardcopy thesis available from the QUT Library

Figure 1-2 Dump Truck Rollover
1.2 Current ROPS Evaluation Techniques

Evaluation techniques used in the current Australian standard for earth moving machinery protective structures AS2294–1997 are simplified and involve the implementation of full scale destructive testing procedures. These procedures require that a full scale prototype ROPS be subjected to static loads in the direction of their lateral, vertical and longitudinal axes. The standard is performance based, with certain force and energy absorption criteria which are derived from empirical formulae related to the type of machine and operating mass. Deflection restrictions are also employed to enable a survival space known as the dynamic limiting volume (DLV) to be maintained for the vehicle operator.

For particular machine types, this form of certification can be extremely expensive as establishing the force and energy criteria can involve large loads which may therefore require the use of a specialized testing facility. In addition to this, the nature of the testing procedure is destructive which means that the ROPS will incur irrecoverable permanent deformation. Consequently, failure of a ROPS to meet the requirements of the standard will mean that another improved prototype will have to be fabricated and re-tested. This process can be both time consuming and extremely expensive and will usually be avoided by most fabricators through providing
additional strength and stiffness to the ROPS. The addition of increased stiffness to the ROPS to avoid premature failure may not be the most desirable solution for the operator’s chances of survival, as a ROPS is an energy absorption device that requires a balance between strength and stiffness to be maintained.

1.3 Analytical ROPS Certification

Heavy vehicle rollover is a complex problem that cannot be solved directly with a simple mathematical model. Procedures currently used in performance standards throughout the world are simplified and have been developed from previous dynamic rollover testing of a variety of machine types. The established procedures in the current standards do not replicate exactly, the complex series of impact forces that may occur during a rollover. The provisions outlined in these standards, do, however provide design guidelines that will substantially improve the operator’s chances of survival during the occurrence of such an event.

Certification of ROPS by a more economical means through the use of analytical modelling techniques is currently not permitted by ROPS standards for earthmoving machinery both in Australia and internationally. Reasons for the exclusion of this provision are attributed to a lack of knowledge and research information available on the behaviour of structures in the post yield region and energy absorption techniques. Early research has shown promise for the use of analytical modelling techniques to model the nonlinear response behaviour of ROPS. These analytical methods have been very simplified and have primarily involved the use of elasto-plastic beam elements to simulate the behaviour of a ROPS subjected to a static lateral load. In recent years, substantial advances have been made in both computational power and the implementation of advanced element types in FE techniques that can accurately predict the nonlinear response of structures, particularly in the post yield region. Based on these advances, the major aim of this project has been to develop an analytical modelling technique that may lessen the need for destructive full scale testing. This aim coupled with the desire to improve operator safety through enhancing the level of energy absorbed by the ROPS during a rollover has formed the basis of this thesis.
1.4 Energy Absorption Techniques

Energy absorption techniques used in the design of ROPS have predominantly relied on the formation of localized yield zones called plastic hinges to absorb some of the kinetic energy of the rollover. This form of energy absorption is characterized by extensive plastic deformation, involving buckling, tearing and possibly fracture of the section. The term plastic hinge is used as a result of the hinge like rotation of the undeformed portion of the structural member about the local deformation zone. Collapse of a ROPS is deemed to take place once a sufficient number of plastic hinges have formed. This type of energy absorption can be extremely effective and could be vital to the operator’s chances of survival in the event of a rollover. Whilst the failure modes of a structure that forms plastic hinges is well understood, little guidance has been given to ROPS designers on how to adequately size a ROPS to meet the requirements of the current performance standards. Through developing a comprehensive analytical modelling technique for ROPS certification, it has been possible to examine the key parameters that control the energy absorbing capability of a ROPS. Based on these findings it has been possible to present design guidelines that will assist ROPS designers to develop adequately proportioned ROPS that meet the requirements of the regulatory standards.

1.5 Aims and Objectives

The main aim of this project has been to develop and demonstrate an analytical modelling technique that will lessen the need for destructive full scale testing. In addition to this the following sub-aims and objectives have been established within the scope of the project and have been outlined as below:

- The examination of ROPS behaviour under static loads in accordance with current ROPS standards using FEA and comparison with experimental testing
- Development of suitable dynamic loads for sideward impact and their application to FE models
- Examination of response behaviour of FE models to dynamic loads and their resulting comparison with behaviour under static loads
- Examination of critical regions, failure modes and energy absorption capabilities of ROPS subjected to both static and dynamic loads
Development of simplified design guidelines that will assist ROPS designers in proportioning ROPS designs that adequately meet the requirements of the ROPS performance standards and provide an enhanced level of safety to heavy vehicle operators.

- To enhance the performance of ROPS by implementing appropriate energy absorption devices, resulting in a higher level of operator safety
- To quantify the loading conditions that ROPS are subjected to during a rollover by studying the parameters that control their response
- To gain an in-depth understanding of the structural behaviour of ROPS under all possible operating conditions, by using analytical and experimental techniques

1.6 Innovation

This thesis has developed comprehensive research information on the nonlinear response of rollover protective structures (ROPS) subjected to both static and dynamic loading conditions. Significant advances have been made towards the development of an analytical modelling procedure that may lessen the need for expensive destructive full scale testing of ROPS. In addition to this, detailed research information has been presented that examines the energy absorption enhancement of ROPS through the inclusion of supplementary energy absorbers. Inclusion of such devices within the framework of a ROPS has led to an increase in the level of safety afforded to the occupants of heavy vehicles during the occurrence of a rollover. These findings will lead to significant advances in the design and fabrication of safety frame structures and will have beneficial outcomes both socially and economically.

1.7 Scope

The scope definition for this project includes the static and dynamic response evaluation of both two post rollbar or cantilever type ROPS with fixed base connections that are suitable for attachment to construction and mining vehicles. Determination of static loading criteria has been in accordance with the recommendations presented in the current Australian standard for earthmoving protective structures AS2294.2-1997 and involved load application in the direction of the vehicle’s lateral, vertical and longitudinal axes only. The dynamic response of a
ROPS during a sideways vehicle rollover has also been evaluated, however, this evaluation has been limited to the first lateral impact between the ROPS and the ground surface. Enhancement of the energy absorption capabilities of a ROPS under dynamic loads through the inclusion of a supplementary energy absorbing device has also been studied, however, this study has been limited to the use of thin walled tapered tube energy absorbers only. It is envisaged that the findings presented within this thesis may be extended and applied to other ROPS types with a variety of geometries and boundary conditions.

1.8 Research and Methodology

The research program that has been employed to study the response behaviour of rollover protective structures, has involved the use of an extensive testing facility with accurate data acquisition capabilities that have been complemented and supplemented by comprehensive computer simulations using innovative FE software. The research components and methodologies that have been used to complete this project successfully are summarised as follows:

1.8.1 Testing of ROPS Frames for a Variety of Machine Types

A number of reduced scale and full scale ROPS frames were fabricated and tested in accordance with the static testing procedure outlined in the Australian Standard for Earthmoving Protective structures AS2294.2-1997. The load deflection characteristics for each ROPS were recorded to enable their energy absorbing capabilities to be determined. Failure modes exhibited by each ROPS during testing were examined and verified using strain gauges to develop an enhanced understanding of the response behaviour of each ROPS. The results obtained from the testing phase were used to assist with calibration of finite element model of each ROPS.

1.8.2 Finite Element Analysis of ROPS Frames under Static Loads

A finite element modelling procedure that replicated the loading stages of the Australian Standard was developed and calibrated against the results obtained experimentally. After verification of this procedure, a numerical study was performed that enabled the important parameters that were responsible for maximising the energy absorbing capabilities of each ROPS to be determined.
Guidelines were developed to assist ROPS designers in proportioning ROPS to adequately meet the requirements of the standards.

1.8.3 Finite Element Analysis of ROPS Frames under Dynamic Loads
The calibrated FE models were subjected to a further analytical study using FEA that involved the application of dynamic impact loads to each ROPS that were characteristic of those that would be experienced during the initial sideways impact during a rollover. This analytical study enabled possible dynamic amplifications to be studied that may be used to assess the adequacy of current performance standard requirements. In addition to this each ROPS model was also subjected to a transient dynamic pulse analysis. This type of analysis enabled the influence of the impact surface properties on the energy absorbing capability of a ROPS to be determined.

1.8.4 Energy Absorption Enhancement
The energy absorbing capabilities of ROPS were enhanced through the inclusion of thin walled tapered tube energy absorbers. Calibrated finite element models of the combined ROPS and energy absorber assembly were subjected to dynamic impact loads that were characteristic of those experienced during a sideways vehicle rollover. The implementation of this study enabled the feasibility of such a device inclusion to be studied in addition to the assessment of the possible improvement of operator safety.

1.9 Thesis Layout
Chapter 2 provides a detailed literature review of the work that was performed both experimentally and analytically on rollover protective structures. Initially, the chapter provides an historical perspective of the development of ROPS performance standards and the methods used by early researchers to develop performance criteria necessary for ensuring adequate operator safety. After completion of this section, the provisions included in the current performance standards that are used both in Australia and worldwide are comprehensively reviewed. Following this review, a detailed investigation into the use of analytical modelling techniques for simulating the structural behaviour of ROPS under static loads in accordance with regulatory provisions is also performed. Finally, the use of supplementary energy absorbing
devices that may improve the energy absorbing capabilities of safety frames such as a ROPS is addressed.

Chapter 3 details the experimental testing that was undertaken during the project on a number of different ROPS frames. Within the scope of this chapter a detailed explanation is provided that outlines the procedure that are used to perform each loading stage and the instrumentation and data acquisition systems that were used to record the results for each ROPS test.

Chapter 4 is concerned with the development of an analytical modelling technique that simulates the loading procedure that is used in the current ROPS standard for earthmoving machinery AS2294.2-1997. Within this chapter a detailed explanation is also provided on the nonlinear modelling technique that was employed to model the response of a number of the ROPS that were tested in Chapter 3. In addition to this, further modelling is performed which examines the importance of key parameters on the response of each ROPS to the loading and energy requirements of the standard. The energy absorption capability of each ROPS is also verified by adjusting key parameters and examining the resulting influence on how energy absorption behaviour is affected under applied static loading.

Chapter 5 involves the use of FEA to study the behaviour of two of the ROPS models that were studied earlier in chapter 4 to simulated dynamic impacts. A simplified procedure based on a conservation of angular momentum approach is employed to determine the necessary dynamic loading parameters. Within the scope of the chapter, the response behaviour of each ROPS is studied for firm surface impacts for varying roll slope inclinations. In addition to this a parametric study is performed which examines the influence of ROPS stiffness on its energy absorption capability under the applied impact loads. A comparison between the static and dynamic response is also studied which enables possible dynamic amplifications to be determined and the adequacy of the code provisions verified.

Chapter 6 involves further dynamic analysis using transient pulse loads applied to one of the ROPS that was examined in chapter 5. The implementation of this loading
procedure was designed to study the response of the ROPS under impacts on varying surface conditions.

Chapter 7 discusses the use of a supplementary energy absorbing device that was fitted to the framework of the ROPS. The inclusion of this device was evaluated under dynamic impact loads. The aim of this study was to promote an enhanced level of operator safety by using a secondary energy absorbing device that was capable of absorbing some of the kinetic energy of the impact, thus lessening the energy demand placed upon the ROPS and increasing the level of operator safety.

Chapter 8 presents the implications of this project and a summary of the research findings and suggestions for further work.
Chapter 2 Literature Review

This chapter provides an in depth review of all literature pertinent to the analysis, design, testing and energy absorption capabilities of rollover protective structures for construction, mining and agricultural machinery. The review is detailed and provides a critical assessment of the literature associated with the following important topic areas:

- Historical development of ROPS Standards
- The performance standards used to evaluate ROPS through full scale testing
- Adequacy of ROPS Standards
- Fundamental principles of ROPS design
- The use of analytical modelling in ROPS design
- Static and dynamic ROPS testing
- Plastic design methods for ROPS
- Energy absorbing devices
- Alternative safety devices for operator protection

The chapter concludes with a summary of the literature review findings and how this thesis will address the shortcomings of the research to date and assist with the development of a more comprehensive understanding of the response behaviour of ROPS.

2.1 Historical Evolution of ROPS Standard Development

The development of performance standards for assessing the capabilities of rollover protective structures, initially began in Sweden in the early 1950’s and was primarily devoted to providing anti-crush protection for the operators of agricultural wheeled tractors. The Swedish developed test methods involving the use of a swinging pendulum and furthermore by application of a static load. Sample tractors with ROPS frames installed, were tested under these conditions and the results compared with those obtained from complete dynamic rollovers. Based on these results, test criteria were established using vehicle weight-kinetic energy relationships Staab (1971). In 1964 the French began a review of all the work undertaken by the Swedish and drafted their own testing procedure which included an additional limitation on deflection that was included to prevent operator injury. In 1966 the British also
developed a draft code of their own that was based on the previous findings of the Swedish and French.

The initial work performed in Sweden using basic energy absorption concepts was considered to be the best approach for developing ROPS performance criteria. In the USA in the mid 1960’s, a need for the development of ROPS performance criteria for construction machinery was sought, however, it was soon acknowledged that a dynamic test using a swinging pendulum was not appropriate for the larger heavier construction machines. Staab (1971) highlighted that a subcommittee formed from members of the society of automotive engineers (SAE) began a review of the previous work conducted on agricultural machinery by the Swedish, French and English. This subcommittee identified a number of critical parameters that were essential for the development of a standard that was applicable to the construction industry, details of which are outlined as follows:

- Test procedures must be established to provide consistent reproducible results
- Tests must be on experience proven structures
- Safety structures must absorb energy
- Safety structures must be matched to tractor frames
- Extrapolation of Swedish data to construction vehicles would have given a clumsy test setup

Through adopting a simplified energy absorption approach, a static force laboratory test was devised by the SAE. The first code, SAE J320, required the ROPS to absorb all of the kinetic energy of the rolling vehicle. Staab (1971) noted that this requirement was not realistic and could be disproved by future roll testing of vehicles equipped with a ROPS. “We found that most parts of a machine and the ground absorbed energy during an actual roll” (Staab, 1971). Aside from these findings, SAE J320 was developed with the objective of testing for structural adequacy of a ROPS under the following conditions:

- Vehicle operating 0-10 mph on a hard clay surface
- Maximum roll angle of 360 degrees
- Down slope of 45 degrees

The testing procedure involved the application of a side load to the ROPS and machine frame assembly until the required level of energy absorption was achieved. Once the structure had absorbed this level of energy a vertical load equal to the
maximum weight of the vehicle was applied to the ROPS. Deflection limitations were also included in the standard to provide protection to the operator during a rollover. A minimum force requirement was also added to the standard to prevent premature failure under multiple rolls by moving the ROPS into a more desirable force deflection range.

The initial ROPS performance standard for construction machinery SAE J320 was the pioneer standard for the development of ROPS performance criteria for construction and industrial machines. The provisions incorporated into this standard were revised on a regular basis and resulted in the formation of the current U.S surface vehicle standard SAE J1040. This standard in association with the current International Standard ISO 3471, Australian Standard AS2294 and Canadian Standard B352.2 form the basis of ROPS performance criteria for construction and earthmoving machinery on an International scale. The following section addresses the unified requirements presented by each of these standards for ROPS evaluation for earthmoving machinery and agricultural tractors.

2.2 Current ROPS Standards for Earthmoving Machinery

Current performance standards, both in Australia and worldwide, employ destructive full scale testing methods to determine the adequacy of a ROPS to be fitted to an earthmoving machine. The common aim adopted by these standards is to “establish a consistent, repeatable means of evaluating the load carrying characteristics of ROPS under static loading and to prescribe performance requirements of a specimen under such loading conditions” (J1040, 1994). It must be noted that the provisions outlined in these standards do not guarantee operator safety under all possible rollover conditions. The intention is rather to prevent or minimize the effects of typical rollover accidents and increase the operator’s chances of survival during such an event. The U.S, Australian, Canadian and International standards indicate that they “will not necessarily duplicate structural deformations due to a given actual roll and will attempt to ensure the safety of a seat-belted operator under the conditions of:

- Forward velocity of 0 to 16 km/h
- Hard clay surface of 30 degrees maximum slope
- 360 degrees of roll about the machine longitudinal axis without losing contact with the ground
Application of each standard is limited to the following earth-moving vehicle types:

- Crawler tractors and loaders
- Graders
- Wheeled loaders and wheeled tractors
- Wheeled industrial tractors
- The tractor portion of prime movers, tractor scrapers, water wagons, articulated steer dumpers, bottom-dump wagons, side-dump wagons, rear-dump wagons and towel fifth wheel attachments
- Rollers and compactors
- Rigid frame dumpers

Each standard is similar in its layout and presents force and energy absorption requirements as well as deflection limiting criteria that a ROPS must fulfil in order to obtain certification. The loading procedure presented by each standard involves the application of static loads in the lateral, vertical and longitudinal directions of the ROPS. The magnitude of each load and the energy absorption criteria are based on empirical equations that are directly related to the type of machine and operating mass. These equations were derived from previous roll testing conducted by the SAE and vehicle weight kinetic energy relationships. The minimum lateral load provision $F_s$ and energy absorption requirement $U_s$ adopted within each standard take the form of equations 2.1 and 2.2 respectively.

$$F_s = K_2 \frac{G}{10000} \gamma^2$$  \hspace{1cm} \text{Equation 2.1}

$$U_s = K_1 \frac{G}{10000} \gamma^4$$  \hspace{1cm} \text{Equation 2.2}

The coefficients $K_1$, $K_2$, $c_1$ and $c_2$ are constants depending on the type of vehicle, whereas $G$ represents the vehicle mass. The vertical loading requirement was initially equal to the weight of the vehicle and was designed to ensure that a deformed ROPS would be able to support the weight of the machine in an upside down position. This requirement was later increased to twice the weight of the vehicle and was thought to have been influenced by the requirements imposed on agricultural tractor ROPS Moberg (1964). It is believed that the longitudinal loading provision arose from recorded measurements taken during dynamic roll tests and was included to ensure
that a ROPS was capable of resisting longitudinal components of forces due to rollover.

2.2.1 Loading Sequence

The loading sequence that is used to test ROPS is comprised of three consecutive loading stages which are outlined as follows:

- Lateral loading and unloading
- Vertical loading and unloading
- Longitudinal loading

2.2.2 Lateral Loading Phase

The lateral load is applied to the upper extremity of the ROPS in accordance with Figure 2-1. The position of the load is such that it encompasses a vertical projection of the dynamic limiting volume (DLV). The load is applied gradually and is continued until both force and energy requirements are fulfilled.

![Figure 2-1 Lateral Load Application](image)

2.2.3 Vertical Loading Phase

The vertical loading phase involves the application of the vertical load to the upper members of the ROPS. The load is positioned within the zone of the DLV and is applied gradually to the structure in order to simulate static loading conditions. Figure 2-2 displays the vertical loading sequence adopted by these standards.
2.2.4 Longitudinal Loading Phase

The final loading sequence which is shown in Figure 2-3 involves application of the longitudinal load to the upper horizontal cross member of the ROPS. The load is applied gradually in the direction that will impose the most severe effects on the ROPS and the machine frame assembly.

2.2.5 Acceptance Criteria

A ROPS has been deemed to satisfy the requirements of each regulatory standard, provided it has met both the force and energy absorption requirements without intrusion of the DLV or premature failure of the structural members or connections.
2.2.6 Testing Facility Requirements

Each standard requires that the facility conducting the ROPS testing be capable of adequately securing the ROPS and machine frame assembly rigidly to a bedplate which is able to sustain the applied lateral, vertical and longitudinal loads. The machine frame, which is a representative portion of the vehicle’s chassis, is also required to possess the following attributes:

- Rigid connections to the bedplate
- Machine/ground suspension elements externally blocked so that they do not contribute to the load deflection behaviour
- Locked articulations

2.2.7 Material Requirements

Material requirements are specified by each standard in order to ensure that good ductility will be achieved and brittle fracture avoided. These requirements are satisfied by either conducting ROPS testing at a reduced temperature of -18°Celsius or through ensuring that the material that the ROPS is manufactured from has a Charpy V-notch impact strength in accordance with Table 2 of AS2294.2-1997 or equivalent standard recommendation.

2.3 Current ROPS Standards for Agricultural Tractors

The Australian standard for Agricultural machinery protective structures AS1636 in association with existing Canadian Standards B352.2 and U.S. Surface Vehicle standards J1194 and J2194, provide both static and dynamic testing methods which may be used to achieve certification for ROPS that are attached to agricultural tractors. Static testing is limited to tractors that fall within the range of 560kg – 15000kg, whereas the dynamic testing provisions are only suitable for tractors with a mass in the range of 800kg to 6000kg. Each standard is similar in its layout and presents force, energy absorption and deflection limiting criteria that a ROPS must obtain in order to gain appropriate certification. Provisions for the use of analytical modelling techniques are discussed in AS1636 and permitted for use as an alternative to dynamic or static testing, however, each standard requires that the analysis must be proven by “physical testing of materially and dimensionally similar frames”. (AS1636.1,1996)
2.3.1 Dynamic Testing Procedure

The dynamic testing procedure outlined in each standard is designed to simulate the dynamic loads experienced by the ROPS during an overturn and consists of a number of loading sequences which are outlined as follows:

- An impact blow from the rear
- A crushing test at the rear
- An impact blow from the front
- An impact blow at the side
- A crushing test at the front

2.3.2 Rear Impact Test

The rear impact test which is shown in Figure 2-4 involves subjecting the ROPS to an impact load created by dropping a swinging mass from a predefined height. The point of impact is located on the portion of the ROPS that will first come into contact with the ground during a rollover. The height that the weight shall be positioned at is given by the following equation:

\[ H = 2.165 \times 10^{-8} \times m_t L^2 \]

where:

- \( H \) = the vertical pendulum drop height in mm
- \( m_t \) = the reference tractor mass in kg
- \( L \) = the maximum tractor wheelbase, in mm

Figure 2-4 Rear Impact Test
2.3.3 Front Impact Test

The frontal impact test is performed in a similar manner, however, the expression relating the height that the weight should be released from is given by the following relationships:

\[ H = 25 + 0.070m_t \] for \( m_t \) between 800 kg and 2000 kg
\[ H = 125 + 0.020m_t \] for \( m_t \) between 2000 kg and 6000 kg

2.3.4 Side Impact Test

The side impact test which is shown in Figure 2-5 also uses a swinging pendulum to apply the necessary impact load to the ROPS. The height that the weight should be released from is given by the following equations:

\[ H = 25 + 0.20m_t \] for \( m_t \) between 800 kg and 2000 kg
\[ H = 125 + 0.15m_t \] for \( m_t \) between 2000 kg and 6000 kg

![Figure 2-5 Side Impact Test](image)

2.3.5 Crushing Tests

The crushing test which is shown in Figure 2-6 involves the exertion of a downward force onto the cab or frame through the use of a stiff beam which is connected to the load application device. The magnitudes of the crushing test loads both frontward and rearward are given by the following equation:

\[ F = 20m_t \text{ N} \]
2.3.6 Recorded Measurements

The measurements made during each test involve:
- The inspection of all members for fractures or cracks
- The inspection of the clearance zone after each test to check whether it has been impeded
- Examination of the maximum momentary deflection during the side impact test which involves the difference between the maximum momentary deflection and the residual deflection at a height of 840mm above and 65mm behind the seat index point
- Measurement of the permanent deflection of the ROPS after the final crushing test from the rear and the front

2.3.7 Acceptance Criteria

Certification of a ROPS is deemed to comply with each standard provided that the ROPS has passed the following criteria:
- Free from fractures and cracks
- Has not been deformed in a manner that has caused the zone of clearance around the operator to be intruded
- During the impact test, the difference between the momentary deflection and the residual deflection at a height of 840mm above and 65mm behind the seat index point shall not exceed the 250mm
2.3.8 Static Testing Procedure

The static testing procedure presented in each standard provides ROPS manufacturers with an alternative method to assess the capability of ROPS fitted to tractors which have a mass in the range of 560 kg to 15000 kg. The loading sequences adopted by this testing method are outlined as follows:

- Longitudinal loading
- First crushing test
- Side loading
- Second crushing test
- Second longitudinal loading

2.3.9 Longitudinal Loading Phase

The longitudinal loading test involves the application of a load in the horizontal plane, which is applied to the uppermost transverse structural members of the ROPS. The direction of the load is dependant on how the mass of the tractor is supported. For tractors where greater than 50% of the mass is supported by the rear wheels, the load is applied from the rear, whereas for those tractors which do not fall into this category, the load will be applied from the front. Load will continue to be applied until either the strain energy absorbed by the ROPS is equal to or greater than the required input energy $E_{il1}$ given by the equation:

$$E_{il1} = 1.4mt \text{ Joules}$$

or the deflection of the ROPS is such that it infringes on the clearance zone.

Each standard also requires that an overload test be conducted if the force drops by more than 3% over the last 5% of the deflection attained while absorbing the required input energy.

The second longitudinal loading test involves the application of a load in the horizontal plane in the opposite direction to that which was applied to the structure in the first test. The test is continued until either the strain energy absorbed by the ROPS is greater than or equal to the required input energy $E_{il2}$ given by the equation:
E_{d2} = 0.35 \text{mt Joules} \text{ or the deflection of the ROPS is such that it infringes on the clearance zone}

### 2.3.10 Side Loading Phase

The side loading sequence involves application of a load in the horizontal plane to the upper extremity of the ROPS at a point 160mm forward of the seat index point. Load will continue to be applied until either the strain energy absorbed by the ROPS is equal to or greater than the required input energy $E_{is}$ given by the equation:

\[
E_{is} = 1.75 \text{mt Joules} \text{ or the deflection of the ROPS is such that it infringes on the clearance zone}
\]

Similar to the longitudinal test, the standard requires that an overload test be performed if the force drops by more than 3% over the last 5% of the deflection attained while absorbing the required input energy.

### 2.3.11 Crushing Tests

The crushing test in both the frontward and rearward directions, are identical to those used in the dynamic testing procedure and involve the exertion of a downward force on the cab or frame through the use of a stiff beam which is connected to the load application device.

### 2.3.12 Acceptance Criteria

Certification of the ROPS is deemed to comply with each standard provided that the following criteria are met:

- Zone of clearance has not been impeded during the test
- The force used to develop the required energy during the specified horizontal loading tests does not exceed $0.8F_{\text{max}}$ where $F_{\text{max}}$ is the maximum static load force occurring during loading
- There are no protruding members or components that are hazardous to the operator
2.4 Adequacy of ROPS Standards

In recent years, the accuracy of current methods used to evaluate the performance of ROPS has been questioned. In a report titled “Methods for optimizing the effectiveness of rollover protective systems”, Stockton (2002) provided a detailed review of current ROPS certification practices used throughout the world. Within the scope of this report the authors highlighted that “the evolution of machinery was outpacing the development of ROPS standards” and that there were numerous machine types that lacked defined approaches to the fitment of ROPS. The authors acknowledged the similarities in standards used worldwide for ROPS certification indicating that a simplified static loading approach based on machine type and weight was currently used to determine the magnitude of forces and energy absorption requirements for ROPS. They suggested that the small number of parameters considered for ROPS evaluation was questionable and that in the light of current needs a review of standards procedures was required.

Many years prior to this Woodward and Swan (1980) aimed to determine whether ROPS provided adequate protection to the operator of a vehicle during a rollover. The authors studied the statistics associated with rollover accidents and conducted a review of ROPS certification test procedures and concluded that ROPS generally exceeded the structural performance requirements presented in the standard and reduced the number of injuries and deaths associated with heavy vehicle rollovers. The authors, however, acknowledged that they were unable to confirm the adequacy of the provisions presented in the SAE standards as most ROPS generally exceeded the performance requirements of the standard, “It is possible that a ROPS designed to exactly meet the SAE recommended practices would provide less satisfactory operator protection.” (Woodward and Swan, 1980) Within this paper the authors also cited that there were large variations in rollover behaviour between the different types of machines. Based on this notion the authors recommended that ROPS performance criteria should be based on protecting the machine operator in a high percentage of the possible rollover accidents for each generic class of machine.
2.5 Principles of ROPS Design

Klose (1969) provided an in depth evaluation of the fundamental principles required for successful ROPS design and suggested that it was comprised of the following engineering disciplines:

- Statics
- Dynamics
- Strength of materials
- Mechanical behaviour of engineering materials
- Structural engineering
- Metallurgy

Within the scope of this paper, the author highlighted that rollover of heavy vehicles was a difficult area that could not be formulated exactly by mathematical expression. He suggested that a successful ROPS design must be able to absorb the energy of the rollover and that energy absorption was characterized by plastic deformation of the ROPS which was an area in design that must be treated with caution. Klose developed a simplified mathematical model that was based on a conservation of energy approach that could identify the pertinent relationships between a rolling vehicle and ground situation. His simplified model considered only a linear force-deflection relationship between the ROPS and ground and was therefore unrealistic as it is well understood that both the ROPS and the ground can undergo significant permanent deformation during the occurrence of such an event. McHenry (1976) suggested that rollover accident analysis using a conservation of energy approach was not always accurate and that it was more desirable to employ the conservation of angular momentum when dealing with such situations. This method too, was adopted by Watson (1967) to determine the likely proportion of energy that would be dissipated by a ROPS in the case of a simple sideways overturn of a tractor on a uniform slope.

Further to these mathematical expressions, Klose suggested that when a rollover occurred on a material that was soft, a stiff ROPS acting as a force transmitter would be more desirable as it would penetrate the ground while deflecting very little itself. In direct contrast to this, for rollovers that occurred on a material that was hard, a soft ROPS that would deflect substantially and behave as a good energy absorber would
be most desirable. Klose indicated that a successful ROPS design must find a balance between these two ideals and must therefore be a good energy absorber and energy imparter. Figure 2-7 illustrates schematically such an ideal relationship whereby the ROPS can be expected to perform satisfactorily under a variety of surface conditions.

For the following figure the Term P represents the lateral force resistance of the structure and ? the corresponding lateral deflection.

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2.6 Design and Testing of ROPS

The design and testing of ROPS to meet the requirements of current regulatory standards has received much attention over the last four decades. Macarus (1971) proposed a design method based on the simple plastic analysis of a portal frame, in order to guide designers in sizing a ROPS to meet the requirements of the SAE J395 standard. He verified his design procedure through conducting static testing on a ROPS designed for an international Harvester E-200 Scraper and concluded that his plastic design approach was accurate. Watson (1967) too employed similar principles for determining the capacity of a ROPS under both static and dynamic loads. Watson suggested that strain rate effects incurred in mild steel during dynamic loading could
result in amplification of member yield stress by as much as 50%. Ho (1994) developed a simplified procedure based on similar plastic analysis theory for determining the energy absorption capability of a two post ROPS prior to DLV violation. He examined frames with collapse loads equal to 100%, 124%, 150% and 200 % of the minimum lateral load provision present within SAE J1040 standard. Ho suggested that only ROPS frames that exceeded the minimum lateral load requirement of the standard by 50 % would absorb the necessary energy prior to DLV violation.

2.6.1 ROPS Testing Procedures

Testing techniques used for the evaluation of ROPS have in most cases employed static loading procedures to simulate the effect of impact loads created during a vehicle rollover. Dynamic testing methods are rarely used, however, current Australian and International Standards associated with ROPS testing for tractors do permit dynamic testing methods as an alternative means for evaluating the performance of a ROPS. The loads that these structures are required to withstand during a rollover are in essence of a dynamic nature and involve the application of a complex series of forces about a multitude of directions. Parameters that typically control the magnitude of these loads include:

- The vehicle’s mass
- The velocity of the vehicle
- The ground conditions
- Inclination of slope
- Machine mass
- Radius of gyration
- Location of the centre of gravity
- Distance of the ROPS from the centre of gravity
- Change in roll rates during impacts
- Translational velocity of the machine
- Penetration resistance of the roll hill

Determination of the true response behaviour of a ROPS under this type of loading is uncertain and will vary significantly depending on these controlling parameters. The adequacy of the application of simplified static forces to simulate the effects of vehicle rollover remains somewhat questionable. With these views in mind Swan
(1988) conducted both static and dynamic testing on a number of front-end wheeled loaders with operating masses in the range of 25-180t. The basis of Swan’s research was derived primarily from a lack of knowledge on the rollover performance of machinery exceeding 60t in weight. This lack of valuable research information, led Swan to establish ROPS performance criteria for large mobile mining equipment by conducting full scale testing on three front end loaders, each weighing 24t, 130t and 176t respectively.

The first stage of Swan’s research involved subjecting a series of machines, fitted with commercially designed ROPS to complete dynamic roll tests on firm slopes with inclinations of approximately 35 degrees. Each ROPS was fully instrumented to enable strain, deflection, acceleration, roll rate and time measurements to be made. Figure 2-8 shows the roll slope setup and the idealised roll motions that would be experienced by each vehicle during testing.

Two dynamic roll tests were conducted on a 25t front-end loader fitted with a four post ROPS. During the first test, the ROPS was subjected to three impacts and the recorded data indicated that the ROPS had experienced loads that exceeded the capacity predicted from previous static testing. The damage sustained by the ROPS during this test was substantial. The second test was conducted on a much firmer roll slope of the same inclination. Swan believed that an increase in ground stiffness
would lead to an increase in the vertical forces that the ROPS would be subjected to during the test. This assumption was found to be incorrect as the recorded loads were actually smaller than those found previously. Swan acknowledged that the ROPS had successfully passed the test requirements, however, he did not provide any information on the extent of the deformation experienced by the ROPS.

Two further dynamic roll tests were performed by the Swan on a 180t front end loader fitted with a four post ROPS. During the first test, the machine was subjected to two complete revolutions before it came to rest in an upright position at the base of the slope. Examination of the strain gauge recordings indicated that the ROPS was subjected to vertical loads that were significantly lower than those predicted from static testing. Swan believed that this had occurred as a result of the machine frame being required to absorb a significant portion of the impact energy in the vertical direction during the rollover. Swan highlighted that during this test, the ROPS had performed adequately under the first 360 degrees of roll, however, the second 360 degrees of roll had caused significant plastic deformation in the ROPS, with complete failure of the base mount for the rear left post taking place. The damage sustained by the mount, was attributed to the uneven distribution of load in the longitudinal direction of the machine during the second revolution of roll. Swan believed that the longitudinal loading experienced by the ROPS during the test, had been responsible for this damage, however, he concluded that the measured load was well within the predicted capacity. The second test was conducted under similar conditions and was reported by Swan to have successfully met the objectives of the test without any significant damage to the ROPS.

Static testing of a four post ROPS mounted to a 180t front end loader was performed in accordance with the provisions outlined in J1040 and involved the application of static loads in the lateral, vertical and longitudinal directions. Figure 2-9, Figure 2-10 and Figure 2-11 show the resulting load deflection response profiles obtained for each loading stage.
Figure 2-9 Lateral Load Deflection Response of 180t Front end Loader (Swan), 1988

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Figure 2-10 Vertical Load Deflection Response of 180t Front end Loader (Swan), 1988

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Swan concluded from his testing that it was not possible to guarantee operator safety under all loading conditions experienced during a rollover. He highlighted that when a vehicle became airborne during a roll, it was possible for large force and energy demands to be placed on the ROPS that could exceed those predicted from static testing. Swan also emphasised that although most rollovers commonly occurred about the vehicle’s longitudinal axis and was therefore characterised by predominantly lateral and vertical loads, it was also possible to generate large longitudinal loads during certain rollover situations. Based on these findings Swan recommended that the longitudinal loads applied to ROPS, be approximately 80% of the lateral load.

2.6.2 Reduced Scale Model Testing

Full scale testing methods currently used for the evaluation of ROPS can be extremely expensive, time consuming and in the absence of computer modelling
assistance, lead to hours of prototype testing in order to achieve a satisfactory product. Early research conducted on ROPS testing during the 1970’s aimed to lessen the economic burden of such tests through the implementation of reduced scale model testing. The first of such tests was performed by Brown et. al. (1974) who employed dimensional analysis theory to conduct reduced scale model testing of ROPS attached to tractors. The testing procedures used for the evaluation of these ROPS were in accordance with the dynamic loading sequence established in SAE J334a which has now been superseded by J2194. They conducted two analyses and derived two sets of design conditions. In the first analysis the strain rate was found to be higher in the model than for the prototype by a factor of the square root of the length scale factor for the same material. In the second analysis, the strain rate and the mass scale factor between the model and prototype were kept equal. The second analysis resulted in better predictions from the model results and indicated that reduced scale model testing could accurately reflect the response behaviour of full scale prototype ROPS which therefore promoted a more efficient and economical means for ROPS evaluation.

Further research conducted by Srivastava et. al. (1978) on the use of reduced scale models for ROPS testing aimed to develop design conditions and prediction equations for the evaluation of standard ROPS tests. Similar to Brown et. al. (1974) the authors employed dimensional analysis theory using Buckingham’s Pi terms to determine the appropriate parameters for the scale model analysis. One quarter and one eighth scale models were developed for testing in accordance with the static and dynamic testing procedures outlined in the ASAE S306.3. The following objectives were established by the authors within the scope of their analysis:

- Identification of the variables pertinent to the problem
- Development of tractor models to experimentally verify modelling of tractor motion during overturn
- Experimental modelling verification of ROPS behaviour under quasi-static loading
- Experimental verification modelling of the dynamic deformation of ROPS during overturn

Dynamic tests conducted by the authors differed from those performed by Brown et. al. (1974), and involved subjecting a reduced scale model tractor fitted with a two
post ROPS to a standard overturn test in accordance ASAE S306.3. The implementation of this test involved releasing of a model tractor from a predetermined height and allowing it to travel freely until it encountered an overturning ramp which subsequently caused it to rollover. Displacements were recorded about each axis at the left hand upper ROPS corner for both of the scale models. The authors highlighted from their analysis that similitude modelling techniques could be used to accurately predict the motion of larger scale test models.

The second phase of the testing program conducted by the authors involved subjecting similar reduced scale ROPS models to quasi static loads. The models used for these tests were fabricated using cold rolled steel with bolted moment resisting connections. The required static transverse loads were applied to the upper corner of each ROPS and the corresponding displacements were recorded. The results obtained from the smaller of the two models, were then used to predict the response behaviour of the larger model by applying the appropriate similitude prediction equations. The authors concluded from their analysis that the deflection predicted from dimensional theory was about 12% less than that recorded for the ¼ scale model.

Concern was expressed by the authors over the influence that the connection type would have upon the response behaviour of reduced scale model ROPS. With these views in mind, the authors performed further study on the behaviour of ROPS models that contained fully welded moment resisting connections. From their analyses the authors discovered that the measured deflections exceeded those predicted from dimensional theory by approximately 35%. Evidently, the inherent residual stresses developed in the members during the welding process had significantly influenced the results of their analysis.

In order to verify their results with full scale tests, the authors obtained the load deflection response for a commercially tested two post ROPS that was geometrically similar to the reduced scale models used in the previous tests. Figure 2-12 presents a comparison between the load deflection behaviour of this ROPS and that obtained for the ¼ scale model with the appropriate similitude principles applied.
The authors highlighted that good correlation between the results had taken place and that the use of reduced scale models was an appropriate and economical means for ROPS evaluation. Further conclusions presented by the authors in their paper are outlined as follows:

- Static test results supported the hypothesis that the equation developed to predict deflection at the yield point could be used to predict complete elastic-plastic deformation of ROPS
- Distortion in the time scale due to strain rate dependence of ROPS material could be accounted for by developing a prediction factor.
- Tractor overturning motion could be accurately predicted using reduced scale model

### 2.7 Analytical Modelling of ROPS

Over the last three decades, analytical modelling techniques employing non-linear finite element analysis have been used by designers to predict the response of rollover protective structures to impact loads created during vehicle overturns. The validity and accuracy of these analyses remains questionable, with some very simplified models being used and little verification taking place between experimental and analytical results. Recent advances in the development of nonlinear
finite element analysis techniques has, however, led to solution procedures that can simulate real life situations much more accurately than ever before. The following section critically reviews much of the previous research that has been conducted by researchers in this area.

2.7.1 Simplified Modelling

Hopler and Stewart (1973) performed nonlinear finite element analysis on a number of ROPS fitted to military vehicles as part of a U.S. army Corps ROPS retrofit program. The authors selected performance criteria in accordance with the SAE standards and used the FEA code solid SAP to conduct their analyses. Two ROPS frames were selected for analysis that were suitable for attachment to a 24t wheeled tractor and a 2.7t rough terrain forklift. The ROPS for the wheeled tractor was studied analytically under the SAE side loading requirement using non linear beam elements. The ROPS failed the test requirements and the FEA correlation was found to be poor. The inaccuracy between the two results was attributed to an error in modelling the boundary conditions of the tractor frame. For this particular test, all energy appeared to be absorbed by the tractor frame with no permanent deformation present in the ROPS. To verify the capabilities of the analytical modelling technique, the authors removed the ROPS from the tractor and tested it statically without the tractor frame in place. They found that the results of the test correlated very well with the FE model.

The ROPS for the rough terrain forklift was also studied analytically and compared with the results obtained from experiment. The model was analysed three times with different boundary conditions to correlate more accurately with the experimental results. Figure 2-13 shows the load deflection response for each simulation and its comparison with experimental results. The authors concluded that the correlation between analytical and experimental results was adequate and that the best correlation took place when the model had been adjusted to account for the machine frame and mounting brackets.
Brown and McNabb (1974) employed computer modelling techniques to simulate the response of ROPS subjected to loading conditions and energy criteria presented in accordance with the requirements of SAE J320a. The authors used the FEA package STRESS and employed nonlinear beam elements to model six different ROPS frames and then compared their results with those obtained from full scale experimental testing. STRESS was used to estimate upper and lower limits on the lateral load deflection response of each ROPS. Figure 2-14 shows a typical ROPS computer model and the application point for the lateral load. Each model was loaded until the force and energy requirements presented in the standard were achieved. Figure 2-15 presents the comparison of the experimental response behaviour to the upper and lower bound numerical solutions for the first ROPS model. From their analyses, the authors discovered that plastic hinges formed initially at locations 3,4,11,12 and then subsequently at locations 5,6,9 and 10 as the load was increased.
Little information was provided by the authors indicating whether the experimental testing had resulted in the formation of plastic hinges at similar locations. This
indicates that strain data was not recorded during the test and therefore correlation could only take place based on the load deflection response. The analyses that were performed appear to have been limited to the lateral loading phase only and therefore did not account for the vertical and longitudinal loading sequences specified in the standard. Within the scope of this paper, the authors made no mention of the type of boundary conditions that were employed during the analysis. Comparison with experimental results is therefore difficult as the type of boundary conditions used in the analysis has a significant influence on the response behaviour of the structure. Based on the information obtained from the analytical modelling and experimental testing, the authors concluded that:

- Simple plastic theory was satisfactory for analysis of a ROPS to meet the requirements of SAE ROPS specifications provided that the material from which the structure was fabricated was sufficiently ductile to meet the energy absorption requirements.
- The program STRESS provided a means for carrying out an approximate three dimensional analysis of a ROPS. Estimated upper and lower limit curves were readily obtained for load-deflection response of a given ROPS subjected to lateral loading.
- Experimental results confirmed the analyses such that one could predict the performance of ROPS accurately enough by analytical means.

Pistler (1975) adopted a similar approach to the previous authors by employing nonlinear beam elements to model the response of a ROPS subjected to a static sideward load. He suggested that there was a need for a practical computerised analysis technique that could predict the maximum force level and energy absorption capabilities of a ROPS. A simplified design criteria was proposed that predicted the formation of a plastic hinge in a three dimensional frame. The FEA package LAGS, which incorporated the simplified criteria was used to conduct nonlinear analysis of a series of ROPS frames. Pistler found that good correlation between analytical and experimental results was obtained and that analytical modelling could be used successfully to design ROPS. Pistler highlighted that for accurate results to be obtained, it was essential for the machine frame and the support stiffness of the ROPS to be modelled and for correct material properties to be used.
Further support for the analytical modelling of ROPS was proposed by Yeh et. al. (1976) who developed a simplified analytical procedure involving elasto-plastic beam elements. The authors suggested that “it became obvious that a structural analysis of ROPS design was essential to effectively produce an adequate structure compatible with its mounting vehicle chassis.” The authors summarized the essential elements required for adequate design of a ROPS which are outlined as follows:

- The structure and its mounting adaptors must be strong enough to absorb the required energy without intruding into the zone of protection
- Sufficient flexibility of the ROPS was necessary to prevent intolerable shocks to the operator and to prevent significant damage to the supporting vehicle chassis

The authors suggested that successful ROPS design required controlled yielding in localized zones in the structure in order for energy to be absorbed. They proposed a simplified Von Mises yield criteria that ignored the influence of shear and axial forces and incorporated this criteria into the FEA package SAPROPS. SAPROPS was used to conduct analyses on a number of different ROPS frames. The example provided in the paper was a ROPS for an AE 656 agricultural tractor. The authors stated that the agreement between analytical and experimental results was very good with a load deflection response under an applied sideward load shown in Figure 2-16.

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Figure 2-16 Experimental and Computer Model Load Deflection Comparison (Yeh et. al.), 1976
It was concluded that the SAPROPS package was a cost effective design tool for the analysis of ROPS and that the simplified yield criteria based on biaxial bending and torsion was valid as these design actions dictated the performance of most ROPS. Limitations in the program were acknowledged with its inability to model the vertical crush loading sequence adequately. The reason for this was attributed to the need for the axial force component to be included in the yield criteria and also the program’s inability to consider buckling of the ROPS members.

There have been a large number of deaths associated with the rollover of heavy vehicles predominantly by those used in the rural and mining industries. Rollover protective structures find alternative uses for other types of vehicles as well, and are commonly fitted to military trucks in order to protect the occupants. Published research in this area is rarely available, however, information has recently been presented on a ROPS retrofitment program that was performed on an existing U.S. military fleet of trucks. The following section presents a detailed discussion on the analysis technique used to validate the performance of the ROPS used in the retrofitment program.

The large numbers of accidents involving vehicle rollovers and the inherent concern for driver safety, prompted the U.S. Army’s Tank and Automotive Command unit TACOM to seek specialist advice for the retrofitment of their military trucks with a ROPS. A prototype of the ROPS was developed by AM General and subjected to static loading conditions that were in accordance with current regulatory standards. The manufacturer believed that “the frame design would be a success if it absorbed the energy of the impact and did not deform into the driver and passenger area”(AM General, 1998). Initial prototype testing resulted in premature failure of the ROPS. It was evident from the preliminary tests conducted by TACOM that the structure was too flexible. Based on this initial failure, the manufacturer employed virtual prototype testing using the FEA package Algor. The failed prototype test model was initially analysed to study the regions of the structure that would experience the largest deflections and highest stresses during loading. Elasto-plastic beam elements were used to model the ROPS with static loading sequences applied in the lateral, vertical and longitudinal directions. After completion of the analysis, AM General studied the resulting deflections and maximum principal stress contours in order to
determine the regions of the structure that needed to be strengthened in order to prevent deformation into the driver and passenger zones. Based on the results of the analysis, the manufacturer concluded that the use of FEA to simulate the behaviour of rollover protective structures was extremely beneficial and equated to large economic savings.

Recent research performed by Fabbri and Ward (2001) developed a simplified finite element analysis program with the aim of simplifying the design and analysis of ROPS for agricultural tractors. The developed program conducted static analysis in accordance with the provisions outlined in the OECD/EEC codes and was validated against test results on a series of ROPS fitted to tractors. The author’s program involved the use of beam elements that were able to account for both nonlinear material and geometric effects. The authors concluded that the program was able to predict the force displacement characteristics of ROPS fitted to tractors to within 20% of those obtained from experimental testing. Both authors emphasized that the developed program was a cost effective design tool, however, the reliability was purely dependent on the accuracy of the geometry, material properties and boundary conditions.

2.7.2 Beam Element Evaluation

The early research performed on ROPS using beam elements involved application of the sideward force only and was not characteristic of the loading sequence adopted in the present ROPS standards. The yield functions used to describe the formation of plastic hinges were simplified and did not account for design action effects that contributed to plastic hinge formation. Localised geometric deformation that takes place during plastic hinge formation can not be accounted for accurately when using beam elements. In addition to this, nonlinear beam elements consider the hinge location to be isolated over a defined zone and do not permit spreading of the yield zone as the hinge fully develops.

2.7.3 Advanced FEA Modelling of ROPS

Further advances in FEA led to the modelling of a Terex scraper ROPS by Hunckler et al. (1985). In this paper the authors emphasised the need for a reliable analytical design method for ROPS evaluation to be developed that would lessen the need for
expensive full scale testing. MSC/NASTRAN was employed to model the non-linear response of the Terex scraper ROPS under static loads determined in accordance with SAE J1040. The non-linear modelling capabilities of MSC/NASTRAN enabled effects such as geometric stiffening and nonlinear material behaviour to be studied.

The authors of this paper were the first researchers to conduct advanced analysis of ROPS using advanced non-linear shell elements. Simplifying assumptions were made in the model development which involved the omission of elastic buckling and the limitation of large rotations by the structure. These simplifying assumptions were considered valid as the analysis was concerned primarily with the prediction of material yielding. The non-linear analysis of the ROPS involved inclusion of a stress strain curve for the ROPS material that reflected perfectly plastic conditions in the post yield region. Analysis problems were experienced with the selection of such idealised material properties. Based on these difficulties an arbitrary curve with a very small strain hardening gradient was then selected which gave rise to more realistic results. Based on the defined material and geometric properties, static analysis was conducted on the rollover cab for both the lateral and vertical loading sequences established in the standard. Figure 2-17 and Figure 2-18 present the load deflection response that was obtained from the analysis for each loading condition.

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Figure 2-17 Horizontal Load Deflection Response (Hunckler et al.), 1985
The authors concluded from their analyses that

- Yielding took place at a horizontal force of 7000 psi for the model with perfectly plastic properties
- The analysis incorporating a minor strain hardening gradient resulted in a much stiffer response with only minor yielding
- The vertical loading phase resulted in a perfectly linear response

Within the scope of this paper it is evident that the authors were unsure as to how well the finite element analysis could accurately predict the true response of the ROPS. The authors indicated that their model only incorporated the cabin section of the vehicle and therefore the results obtained could not accurately be compared with full scale tests that incorporated both the cabin and vehicle frame. Boundary conditions used in the analyses were not discussed and it is evident that model calibration did not take place as no experimental tests were conducted. The importance of these parameters in association with the incorporation of precise material properties, particularly in the post yield region is vital to the accuracy of this type of non-linear analysis. It is evident therefore, from examination of this paper that further research into non-linear analysis of ROPS employing shell elements is required.
Tomas et al. (1997) developed a computer based modelling technique for the certification of ROPS to (AS2294 -1994). This technique was developed primarily as a result of the large number of deaths that occurred as a result of heavy vehicle rollovers in the state of Victoria between 1985 and 1993. The authors believed that it was necessary for an economical ROPS evaluation technique to be developed that would allow owners of existing heavy vehicles to have ROPS retrofitted to their machines without the need for expensive full scale destructive tests.

The authors developed their computer based technique in accordance with the analytical modelling guidelines presented in a report published by SAE J1040. Based on these guidelines, it was established that an analytical modelling procedure involving finite element analysis that could accurately predict local and column buckling of beams and extensive plasticity was essential to predict the response behaviour of a ROPS accurately. The authors also emphasized that the use of FEA provided a number of added benefits that were not available through physical testing, details of which are outlined as follows:

- Efficient evaluation of alternative designs
- Identification of stress concentrations in critical regions
- Elimination of chassis damage caused during testing
- Evaluation of existing uncertified ROPS

The authors conducted FEA of a ROPS retrofitted to a CATERPILLAR 12G grader using the static loading provisions of AS2294. The FEA package ABAQUS was used in association with the preprocessor HYPER-MESH to conduct the analysis. The model was subjected to the static loads in the lateral, vertical and longitudinal directions and was found to pass the requirements of each of these tests adequately without any intrusion into the DLV. Figure 2-19 shows the deformed shape of the structure after the application of these loads. The authors noted that during the test, the lateral load had to be increased to approximately three times the minimum load specified in the standard in order to achieve the required level of energy absorption. This led the authors to conclude that the ROPS was too stiff and that the member sizes could have been reduced in order to meet the loading requirements.
A three dimensional dynamic rollover simulation was conducted to study the adequacy of the standard and the importance of the operator restraint system. A bulldozer fitted with a ROPS, was chosen as the test case for this analysis and the computer program MADYMO was employed to perform the necessary calculations. Tomas et al. employed rigid bodies to model the dozer and ground slope and finite elements to model the ROPS. To study the effect of the restraint system, the model also incorporated a 50% hybrid III dummy. From their analysis the authors concluded that:

- ROPS provide minimal protection for an unrestrained operator
- Operators restrained by a lap belt can be in danger of violating the DLV zone during a rollover, therefore indicating that the DLV space is not a true indication of the required survival space.
- Operators restrained by a harness for an adequately designed ROPS will remain within the zone of the DLV.

The importance of ROPS stiffness was also thoroughly investigated by the authors during their analyses. They suggested that the design of high stiffness ROPS should be avoided as it resulted in very little of the kinetic energy of the rollover being absorbed by the ROPS and allowed significant rebounding of the vehicle which imposed adverse effects on the driver. The implementation of a low stiffness ROPS was suggested by the authors to be inappropriate also, as it allowed for possible intrusion of the DLV during rollover and reduced the ability of the ROPS to sustain the impact from multiple roll revolutions.

Based on these analyses, the authors concluded that medium stiffness ROPS provided the best overall solution for occupant protection, as they resulted in
moderate loads being transmitted to both the ROPS and the occupant, whilst maintaining a survival space around the operator. The authors proposed that the current standard include the introduction of a stiffness ratio for ROPS which would be defined as the ratio of the force required to meet the energy requirements divided by the standard minimum lateral load requirement. It is believed that the introduction of such a requirement whereby the ratio was limited to a maximum value between 2 and 3 would reduce the possibility of overly stiff ROPS designs.

2.7.4 Development of a Hinge Super Element

Recent research performed by Kim and Reid (2001) attempted to provide a simplified computational tool for the design of ROPS to the regulatory quasi-static loading tests present in SAE J1040. The authors studied the biaxial bending collapse of thin walled rectangular tubes using finite element analysis and developed constitutive relationships that enabled a hinge super-element to be formulated. The hinge super-element was used in conjunction with beams elements in a simplified model of a four post ROPS and was subjected to static loads in accordance with the SAE J1040 procedures. Results were compared with a full shell element model of the same ROPS with good correlation taking place. The authors concluded that the new approach provided useful information about the sequence of member collapse which would assist with making modifications at the early design stage.

2.7.5 Analytical Modelling of ROPS under Dynamic Loads

Most of the research that has been conducted on ROPS has been concerned with static force application to simulate rollover impact loads. This method of analysis has been considered a simplified, yet accurate approach for assessing the performance of these types of structures when used on earthmoving machinery. The analysis of ROPS for use on tractors, however, presents designers with the option of conducting a series of dynamic tests to assess the performance of these types of structures as an alternative to typical static testing methods. With these views in mind, Harris et al. (2000) used non-linear finite element modelling techniques to compare the stresses in ROPS subjected to simulated (SAE J2194) static testing methods, with stresses induced by simulated dynamic rearward rollovers. In this paper the authors acknowledged that static testing alone was sufficient for certification of ROPS, however, they highlighted that the actual forces applied to the structure during a
rollover were of a dynamic nature and could be applied from various orientations. Based on this, the authors aimed to “gain insight into how the structural response of ROPS to dynamic overturns compared with the response to the static test sequence of (SAE J2194).” (Harris et al., 2000)

The finite element model developed for both the static and dynamic analyses was based on a 50 HP tractor fitted with a manufacturer supplied ROPS. Nonlinear beam element were used to model the ROPS/Tractor assembly details of which are shown in Figure 2-20.

Analysis of the ROPS under the static loading conditions established in SAE J2194, involved the application of loads in the following directions:
- Rear longitudinal
- Vertical
- Transverse
- Front longitudinal

Further requirements presented by the standard, were energy absorption in the rear longitudinal, transverse and front longitudinal directions. The second stage of the analysis involved the simulation of a rear overturning event. The existing finite element model used in the static modelling phase was adjusted slightly to incorporate...
the tractor frame, so that inertial effects could be approximated. The mass 
distribution required to simulate the rollover was approximated by placing lumped 
masses at the position of the tyres and the calculated centre of gravity for the tractor. 
Rigid beam elements were used to connect the lumped masses to the remainder of 
the model in order to contain the majority of the rollover deformation energy to the 
ROPS. Solid elements were used to model the response behaviour of the ground 
plane and contact elements were defined for the ROPS-ground interface. The tyres of 
the tractor were modelled using compression only members so that they could 
support the weight of the tractor in the at rest position as well as leave the ground 
during the rollover.

The simulation of the rear overturning event was conducted for an assumed 60 
degrees slope and for rotational velocities that varied between 1 and 3 radian/s. 
These parameters were chosen in order to simulate the field upset test requirements 
outlined in SAE J2194. Figure 2-21 shows the FE model used in the analysis and the 
positioning of the lumped masses within the frame.

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Figure 2-21 FE ROPS Model (Harris et al.), 2000

The authors recorded the maximum axial stresses at the nodes for each beam element 
under both the static and dynamic loading conditions. The authors found that the
static prediction underestimated the stresses experienced by the dynamic simulation by a figure of approximately 7% in four of the eight beam elements. The stress levels in the remaining elements, however, tended to be overpredicted by as much as 32%. Details of the comparison between these stress levels are shown in Figure 2-22.

Figure 2-22 Element Stress Level Comparison (Harris et al.), 2000

From their analyses the authors concluded that static testing methods presented in SAE J2194 were not always conservative in predicting the response behaviour of ROPS in comparison to simulated dynamic behaviour. The authors highlighted that their analysis was very simplified and that many assumptions had to be made which are summarised as follows:

- Material properties for the ROPS were approximated for mild steel
- The ROPS uprights were given uniform cross section where in fact these members are commonly tapered
- Mass distribution for the tractor was approximated by lumped masses at the wheels and centre of gravity
- The amount of energy absorbed in the tractor frame was negligible compared to the amount absorbed by the ROPS
- Fastener details were not modelled

This paper outlined an in depth analysis of the behaviour of ROPS frames fitted to tractors under both static and dynamic loading conditions. Similar to previous analyses performed by other authors, no experimental testing program was conducted to enable validation of the analytical results to take place. Little information has been provided on the comparison of the energy absorption attained by the structure under each loading condition and no reference to the bending stresses induced in the
members has been provided. The authors highlighted that the true dynamic behaviour of a ROPS could not be accurately predicted from static testing methods presented in current design standards and that further research in this area was required. Critical assessment of the author’s findings with relation to the simulation of dynamic loads cannot be made as little information has been provided by the author on this topic area within the paper.

2.7.6 Analytical Modelling Provisions Present within Current Design Standards

Whilst certification of ROPS through analytical means has not been permitted in many of the current design standards, the SAE published a report in 1989 which made reference to the use of specialised analytical procedures that could be used to predict the structural behaviour of ROPS and machine frame systems that were subjected to testing in accordance with the provisions outlined in SAE J1040.

The report acknowledged that the use of analytical methods for modelling ROPS was an economical and efficient method for accessing their adequacy prior to testing and also for predicting the effects of changes to proven ROPS designs before retesting. The report divides the analysis of ROPS into two distinct areas which are outlined as follows:

- Structural analysis where the members of the ROPS are subjected to loading conditions that do not cause them to exceed their elastic limit and the analysis therefore remains linear
- Structural analysis whereby the members receive loads that induce stresses that exceed the elastic limit of the material and results in permanent deformation of the structure. The analysis therefore becomes non-linear and must take into account the plastic properties of the material.

The analytical approach is discussed in detail within the scope of the report and is summarised as follows:

- The modelling of the ROPS and machine frame may involve the use of beam and shell elements to develop an accurate model.
- Force versus deflection response must be determined under each loading condition for both the elastic and plastic ranges of the material. Local and column buckling must be taken into account when employing beam elements
in the analysis and the element types must be capable of retaining residual stresses under multiple and sequential loading conditions.

- The analysis must be capable of updating the structure geometry during deflection and the necessary changes in the boundary conditions must be considered.
- The joints are important to the response behaviour of the structure and must be modelled accurately.

The report focuses heavily on the implementation of the correct elastic and plastic material properties to be used in the analysis and emphasises that the correct use of these properties will increase the accuracy of the force-deflection curve prediction. The report acknowledges that analytical modelling techniques can result in the optimisation of the ROPS by studying changes in material thicknesses, section geometries and material types.

### 2.7.7 Development of a ROPS Standard Based on Analytical Methods

In mid 1999, the SAE initiated a research study to determine the feasibility of developing an interactive J1040 ROPS smart standard that would employ computer simulation techniques to verify the adequacy of ROPS. Comments made by Blanchard (2003) suggest that this particular project has come to an end. He cites the difficulties in regulating the use of FEA to conduct such procedures and the difficulties associated with nonlinear analysis techniques as the main reason for the inability to develop such a standard “you would need to certify the knowledge and experience of the people doing the work. You would end up creating a new organization of people who certify the people doing the certification analysis.” (Blanchard, 2003)

### 2.8 Alternative Safety Equipment

The concept of a rollover protective structure originated through the desire to prevent drivers being crushed when their vehicles overturned. When this type of structure was originally conceived, it was the most logical option for providing protection to the operator. In modern day society, however, vehicles have changed dramatically and it is possible that other forms of safety equipment may be more suitable for providing rollover protection to vehicle operators than the common place ROPS. Table 2-1 outlines a detailed list of some of the options that have been developed as
alternatives to a ROPS as well as providing additional information on their feasibility and current applications.

Table 2-1 Methods for Preventing or Protecting against Rollover

<table>
<thead>
<tr>
<th>Concept</th>
<th>Existing Application</th>
<th>Suitability</th>
<th>Commercial Status</th>
<th>Comments</th>
</tr>
</thead>
<tbody>
<tr>
<td>Structural protective devices incorporated into operator’s seat</td>
<td>Motor sports</td>
<td>Light machines</td>
<td>Needs further development</td>
<td>Research required to determine level of protection provided</td>
</tr>
<tr>
<td>Seat airbag</td>
<td>Automobiles</td>
<td>Small ride on machines</td>
<td>Widely adopted but for protection in one direction only</td>
<td>Beneficial in automobile sector more research needed</td>
</tr>
<tr>
<td>Break away cab</td>
<td>Unknown</td>
<td>Medium to large size machines</td>
<td>Unknown</td>
<td>Force required to effect breakaway could be as harmful as rollover impact</td>
</tr>
<tr>
<td>Energy absorbers</td>
<td>No specific devices but integrated into machine</td>
<td>All machines</td>
<td>Accepted concept</td>
<td>Account for energy absorption in standards</td>
</tr>
<tr>
<td>Pop-up protective guard</td>
<td>High performance cars</td>
<td>Small to medium size machines</td>
<td>Adopted by Volvo and Mercedes Benz</td>
<td>Similar function to ROPS but not permanently in place</td>
</tr>
<tr>
<td>Remote control</td>
<td>Many types of vehicle</td>
<td>Most machines</td>
<td>Widely adopted in hazardous environments</td>
<td>Introduction of additional risks</td>
</tr>
<tr>
<td>Helmets</td>
<td>Many occupations</td>
<td>ATV’s</td>
<td>Widely adopted where head injury is an accepted risk</td>
<td>Marginally reduces risk of severe head injuries</td>
</tr>
<tr>
<td>Explosive devices</td>
<td>Unknown</td>
<td>Unknown</td>
<td>Trialled but not used</td>
<td>Possibility of other safety problems</td>
</tr>
<tr>
<td>Chassis suspension</td>
<td>All road and some</td>
<td>All machines</td>
<td>Widely adopted</td>
<td>Marginal</td>
</tr>
</tbody>
</table>

51
<table>
<thead>
<tr>
<th>System</th>
<th>Off Road Vehicles</th>
<th>Improvement</th>
</tr>
</thead>
<tbody>
<tr>
<td>Active suspension system</td>
<td>Some road and off road vehicles</td>
<td>Commercially proven Suspension could be tuned to extend a safe working range</td>
</tr>
<tr>
<td>Stability Aids</td>
<td>Usually for Stationary machines</td>
<td>Could be acted by operator or automatically</td>
</tr>
<tr>
<td>Machine with low centre of gravity</td>
<td>Location of batteries in forklift trucks</td>
<td>Lowering of c.g by adding mass may make it unnecessarily heavy</td>
</tr>
<tr>
<td>Traction control</td>
<td>Road and off road vehicles</td>
<td>Marginal improvement</td>
</tr>
<tr>
<td>Operator training</td>
<td>Widely but inconsistently used</td>
<td>Difficult to evaluate</td>
</tr>
<tr>
<td>Mapping risks to a positioning system</td>
<td>Patch spraying</td>
<td>Use of GPS and further research required</td>
</tr>
</tbody>
</table>

2.9 Energy Absorption Devices

Energy absorption devices are commonly incorporated into structures to reduce peak loads transferred to critical structural elements. The applications for such devices are numerous and include fields such as structural crashworthiness of automobiles, ship fenders and the seismic mitigation or isolation of building structures. The principles behind such devices are simple and involve the conversion of total or partial kinetic energy into another energy form. Little information has been published on the use of supplementary energy absorbing devices for improving the performance of ROPS. Castejon et al. (2001), however, did propose the use of a grill-like energy absorber comprised of fibreglass and polyester 3D fabric that was incorporated into a bus rollover structure and used to improve its energy absorption characteristics. He suggested that this device could reduce the impact energy transferred to the ROPS by as much as 30%. The following section provides a review of the different types of energy absorbing devices that are commonly available and may be feasible for enhancing the performance of a ROPS.
2.9.1 Thin Walled Tubing

Thin-walled tubing is commonly used to absorb energy during impacts. These types of energy absorbers are fabricated from either steel, aluminium alloy or fibre reinforced composites. Metallic tubing dissipates energy through gross plastic deformation, whereas composite tubing involves a complex combination of fracture mechanisms including matrix cracking, delamination and fibre breakage. Common applications for such devices include: Car bumpers, train buffers and at the base of lift shafts.

2.9.2 Thin Walled Frusta

Frusta consist of tapered thin walled tubes that absorb energy through axial crushing of the side walls. The tapered geometry of frusta gives them a most stable configuration and makes them particularly good energy absorbers for both axial and off axis oblique loads. Figure 2-23 shows some of the commonly available frusta geometries that are used to absorb energy. Nagel (2004) conducted numerical modelling of thin walled tapered frusta and found them to be an efficient energy absorber under dynamic impact loads due to their relatively stable mean load deflection response.

Figure 2-23 Typical Frusta Geometries (Nagel), 2004

2.9.3 Honeycombs

Honeycombs are one of the most widely used energy absorbers and are commonly used for applications involving the absorption of impact energy such as in the case of automobile bumper bars. These types of energy absorption devices are fabricated from a variety of materials including: steel, aluminium alloy, thermoplastics, injected moulded polyolefin and elastomeric material. These types of energy absorbers rely
on crushing and collapse of the cell walls to dissipate energy. Figure 2-24 presents an illustration of a typical honeycomb cell.

![Figure 2-24 Typical Honeycomb Cell](image)

2.9.4 Foams
Foams are made from either a polymer or metallic substance and rely on the crushing of the cellular structure in order to dissipate energy. Aluminium foam has been identified as one of the highest efficiency energy absorbers since its load deflection profile is almost rectangular in shape which represents maximum energy absorption. Research performed by Santosa et al. (2001) highlighted that aluminium foam could be used to increase the specific energy absorption and bending resistance of thin walled hollow sections. Chen (2001) conducted similar experimental and numerical evaluation of aluminium foam filled hat profiles and also concluded that both the bending resistance and specific energy absorption of thin walled sections could be substantially increased through the inclusion of foam filling.

2.9.5 Passive Dampers
Passive dampers are another form of energy absorbing device that are commonly used to reduce energy absorption demand on primary structural elements. These types of devices have been used successfully in building structures to reduce the
adverse effects created from large scale loadings events such as earthquakes. The most commonly used dampers include:

- Metallic dampers
- Friction dampers
- Viscoelastic dampers

Metallic dampers are one of the most effective mechanisms available for energy dissipation and rely on the inelastic deformation of metallic substances. Figure 2-25 shows a common type of metallic damper that is commercially available.

Friction dampers generally have moving parts which slide over each other and dissipate energy through friction between the opposing components. Viscoelastic dampers may consist of either solid properties such as copolymers that dissipate energy through shear deformation or closed cylinders such as in the case of an isolators that force fluid or gel through an orifice located inside a piston. Hydraulic isolators are relatively efficient, however, their major disadvantage is that they are impact velocity sensitive and therefore under high speed impacts result in the transfer of large forces to the critical structural elements. Figure 2-26 shows a viscoelastic damper suitable for installation in a typical building frame system.

Figure 2-25 Metallic Damper

Figure 2-26 Viscoelastic Damper
2.10 Literature Review Findings

This chapter has presented a detailed literature review on all relevant topics associated with the design, testing and energy absorption enhancement of rollover protective structures for construction and earthmoving machinery. The following key critical findings have been outlined in order to assist with the fulfilment of the objectives of this project

- There is a certain degree of uncertainty over the adequacy of current ROPS standards and their applicability to current machinery
- The ROPS testing standards do not give designers enough guidance on how to adequately design ROPS to provide an optimum level of protection to the operator
- Current ROPS certification methods are expensive and may involve a trial and error process to develop a ROPS that adequately meets the specified requirements
- Currently the use of analytical modelling techniques is prohibited for certifying ROPS for earthmoving machinery
- Most of the research conducted on the analytical modelling of ROPS has been simplified and inconclusive
- A knowledge gap exists on the nonlinear response of safety frame structures such as ROPS and there is a need for further research on advanced numerical modelling techniques using finite element analysis
- The use of energy absorbers to enhance the performance of ROPS has received little attention and this is an area that needs further research as the level of safety provided to vehicle occupants could be improved.
- The response behaviour of a ROPS subjected to dynamic impact loads is an area that analytically has received little attention.
- Simple plastic theory can be applied to the design of ROPS, however, no guidelines are given on the maximum acceptable force levels that the ROPS should withstand and the possible adverse effects that it may impose on the operator and vehicle
- The use of reduced scale models has been shown to reduce the costs of testing
Alternative measures to ROPS may be used to provide protection to occupants during vehicle rollovers

Elimination of hazards causing rollover accidents, through the implementation of appropriate training and site safety measures may be used to lessen the frequency of heavy vehicle rollovers on construction and mining sites

Overall the review reveals the need for further research intended to enhance the understanding of ROPS behaviour involving dynamic analysis techniques and the need to improve ROPS performance through increased energy absorption levels.
Chapter 3 Experimental Testing

The testing of a number of different ROPS configurations has formed the basis for developing a comprehensive understanding of the response behaviour exhibited by these structures when subjected to loading requirements adopted in both current Australian and International performance standards. Within this chapter, a detailed description is provided on the testing procedure that was used to test three different ROPS configurations that were suitable for attachment to the following vehicles:

- A 100 tonne rigid frame dump truck
- A 50 tonne bulldozer
- A 13.5 tonne Powertrans dump truck

Selection of these particular ROPS types was based on the availability of working drawings and vehicle details from the ROPS manufacturer. In addition to this the configurations tested were also characteristic of common ROPS geometries that are used to provide operator protection for many earthmoving vehicles types. The results obtained from the testing phase enabled possible failure modes and energy absorption characteristics of ROPS to be monitored and assessed. In addition to this it was also found that large stiffness variations may be used to adequately fulfil the requirements of the standard.

3.1 Experimental Test Setup

The experimental test setup involved the testing of three independent ROPS frames both at the Queensland University of Technology and the testing premises of the PTE group, who are a large manufacturer of ROPS. Testing of the ROPS for the 630E dump truck and the K275 bulldozer was performed at ½ scale at the Queensland University of Technology’s testing laboratory, whereas the testing of the full scale ROPS for the Powertrans dump truck was performed at the PTE group’s facility located at Wacol, Brisbane.

In order to meet the performance criteria of the Australian Standard AS2294-1997, a specialised testing frame which could deliver loads in the lateral, vertical and longitudinal directions of each ROPS was designed, fabricated and assembled at the QUT structures laboratory. The testing frame was fabricated primarily from hot
rolled structural steel sections that were securely mounted to the strong testing floor of the laboratory. The general arrangement of the testing frame setup, showing the position of the major structural members is shown in Figure 3-1.

The testing frame was arranged in a manner that enabled static loads to be applied to the ROPS in the direction of its lateral, vertical and longitudinal axes. On the strong floor four sets of horizontal gridlines in the plan horizontal and vertical directions were assigned to assist with the erection of the testing frame. For the defined grid positions which are shown in Figure 3-1 only grids 1, 3 and 4 were used in the horizontal direction and Grids B and D in the vertical direction. To achieve this, a series of two-dimensional frames consisting of two columns and two tying beams were arranged along three independent grid lines. Each frame was then linked together by means of a longitudinal beam that was supported by the uppermost tying beam of each frame. Two additional horizontal members linking grids 1 and 3 were
also provided in the frame assemblage. The first beam which was positioned along grid B acted as a jacking beam to enable delivery of the lateral load to the ROPS. The second beam which was positioned along grid D was designed to restrain the column supporting the ROPS from twisting during the lateral loading sequence. Diagonal bracing members were also positioned predominantly at loading or reaction zones in the frame in order to minimize the displacement of the loading frame during jacking and to assist with transfer of reaction forces into the strong floor.

3.2 Testing of ½ Scale ROPS for 630E Dump Truck

The 630E dump truck is commonly used in the mining industry for the haulage of bulk material on site. This type of truck when fully loaded may carry up to 170 tonnes of material, which gives rise to a total gross vehicle weight of approximately 295 tonnes. In the event of a rollover, protection to the operator is provided through a combination of the ROPS positioned around the vehicle’s cabin and the dump body. Figure 3-2 shows a typical 630E with the ROPS and dump body in place. Positioning of the ROPS is such that it encloses the cabin of the vehicle and permits a survival space (DLV) for the operator in the event of a rollover. For the ROPS design to be adequate, it must be capable of withstanding the load and energy requirements adopted by the relevant regulatory standard without intruding into the zone of the DLV.

![Figure 3-2 630E Dump Truck including ROPS and Dump Body](image)
3.2.1 ROPS Description

A half scale model ROPS for the 630E dump truck was fabricated by the PTE group and tested at the QUT structures laboratory. The dimensions and member sizes adopted for this ROPS were based on an existing design developed by the PTE group that was used for the retrofitment of a fleet of 630E dump trucks. The member sizes and overall dimensions of this ROPS were reduced using scaling laws developed from a similitude study that was performed on this ROPS. The structural configuration of the half scale 630E ROPS consisted primarily of a two post three dimensional moment resisting frame that cantilevered directly from a tapering box section beam that was rigidly connected to the testing frame. Member types employed for this structure consisted of 350 grade steel rectangular/square (RHS/SHS) hollow sections with full penetration butt welded moment resisting connections. The box beam section that the ROPS was connected to was fabricated from 16mm 350 grade steel plate. Non destructive magnetic particle testing was undertaken by the fabricator of the ROPS prior to testing to ensure that no defects were present in the welds and that premature failure would not take place in these zones during testing. Haunches fabricated from the same RHS/SHS material were also included in the ROPS design to strengthen zones that would be highly stressed during loading. Figure 3-3 shows an illustration of the half scale ROPS specimen prior to testing.

![Figure 3-3 Half Scale 630E ROPS](image.jpg)
3.2.2 Similitude Modelling

Previous research performed by Srivastava et. al. (1978) had shown that the principles of similitude modelling could be successfully applied to ROPS testing techniques. The authors had suggested that the use of such principles could lead to large scale economic savings when performing tests on ROPS. Based on the research findings of these authors, similitude was used for the 630E ROPS in order to lessen fabrication costs and reduce the magnitude of the loads that were required to be applied to the ROPS during testing. Reduction of the load intensity was essential, as the full scale testing of a ROPS for a vehicle such as this requires the application of extremely large forces which would exceed the capacity of the testing facility.

Buckingham’s pi theorem was employed to determine the number of independent dimensionless parameters that would influence the behaviour of the system. Once the independent pi terms were determined, they were equated between the model and prototype ROPS to establish the model design conditions. A scaling factor of $\frac{1}{2}$ between the model and prototype was then chosen, which gave rise to the following relationships which are outlined in Table 3-1.

Table 3-1 Relationship between half scale model and prototype parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Model and Prototype Relationship</th>
</tr>
</thead>
<tbody>
<tr>
<td>$I_{XX}^m$</td>
<td>$I_{XX}^m = \frac{I_{XX}^p}{16}$</td>
</tr>
<tr>
<td>$I_{YY}^m$</td>
<td>$I_{YY}^m = \frac{I_{YY}^p}{16}$</td>
</tr>
<tr>
<td>$F_{\text{Lateral}}^m$</td>
<td>$F_{\text{Lateral}}^m = \frac{F_{\text{Lateral}}^p}{4}$</td>
</tr>
<tr>
<td>$F_{\text{Vertical}}^m$</td>
<td>$F_{\text{Vertical}}^m = \frac{F_{\text{Vertical}}^p}{4}$</td>
</tr>
<tr>
<td>$F_{\text{Longitudinal}}^m$</td>
<td>$F_{\text{Longitudinal}}^m = \frac{F_{\text{Longitudinal}}^p}{4}$</td>
</tr>
<tr>
<td>$U_{\text{Lateral}}^m$</td>
<td>$U_{\text{Lateral}}^m = \frac{U_{\text{Lateral}}^p}{8}$</td>
</tr>
</tbody>
</table>
Where:

$I_{xx}$: Second moment of area of the ROPS structural members about their $x$ axis

$I_{yy}$: Second moment of area of the ROPS structural members about their $y$ axis

$F_{\text{Lateral}}$: Lateral load requirement for the ROPS

$F_{\text{Vertical}}$: Vertical load requirement for the ROPS

$F_{\text{Longitudinal}}$: Longitudinal load requirement for the ROPS

$U_{\text{Lateral}}$: Lateral energy absorption requirement for the ROPS

$\delta_{\text{Lateral}}$: Deflection of the ROPS in the lateral direction

$\delta_{\text{Vertical}}$: Deflection of the ROPS in the vertical direction

$\delta_{\text{Longitudinal}}$: Deflection of the ROPS in the longitudinal direction

For the above mentioned formulae, the subscripts $M$ and $P$ represent the reduced scale model and prototype respectively.

### 3.2.3 Experimental Investigation

The half scale 630E ROPS model was subjected to the loading and energy requirements of AS2294.2-1997 which have been addressed in detail in chapter 2. Briefly, this standard requires a ROPS to withstand consecutive loading and unloading sequences that are applied in the directions of the structure’s lateral, vertical and longitudinal axes, whilst maintaining a protective zone around the operator. Further to these requirements, the ROPS must also absorb a predefined amount of energy which is related to the vehicle type and its overall mass. Load magnitudes and energy absorption levels are based on empirical equations relating to the type of vehicle to be tested and its overall mass. Figure 3-4 shows an illustration of the ROPS showing how the static loads are applied to it in accordance with the Australian Standard provisions.
In addition to these requirements, for rigid frame dumpers such as the 630E, the standard allows the peak lateral and longitudinal loads only, as well as the energy absorption in the lateral direction to be reduced by 40% to allow for sharing action between the ROPS and the vehicle’s dump body to take place. A summary of the loads that were applied to the half scale ROPS taking this 40% reduction into account as well as the scaling laws established from the similitude study, are presented in Table 3-2.

Table 3-2 AS2294.2-1997 Load and Energy Requirements for Half Scale 630E ROPS

<table>
<thead>
<tr>
<th>ROPS Performance Criteria</th>
<th>AS2294.2-1997 Formula with Scaling Provision</th>
<th>Result for Half Scale Model</th>
</tr>
</thead>
<tbody>
<tr>
<td>$F_{\text{Lateral-Min}}$</td>
<td>$\frac{1}{4} ?? 0.6 ?? 413500 \frac{M}{?10000} \frac{M}{?}^{0.2}$</td>
<td>98 kN</td>
</tr>
</tbody>
</table>

Figure 3-4 Half Scale 630E ROPS Load Application
### 3.2.4 Instrumentation and Measurement Parameters

Strain and deflection measurements were taken during each loading sequence through the use of an Agilent Technologies 120 channel VXI data acquisition system. Eight Strain rosettes with 5mm gauge lengths were strategically positioned throughout the structure in regions of predicted high levels of strain and recordings were taken at consistent intervals during the lateral loading stage only. Deflection of the ROPS was measured using LVDT’s that were placed at four locations within the structure. The LVDT’s were securely mounted to an independent steel frame that enabled true readings to be taken about each axis of the ROPS. Polished stainless steel plates were attached to the surface of the ROPS at the location of each LVDT. These plates were used to allow the ROPS to slide independently of each deflection monitoring device. The use of these plates was particularly important as the ROPS underwent significant deflection about all three axes during each loading stage. The use of another deflection measuring device such as a stringplot, would have given rise to deflection measurements that were not purely confined to the axis of measurement and would have therefore been inaccurate. Figure 3-5 shows the location of the strain rosettes and the LVDT’s throughout the ROPS.

Load generation for each loading stage was performed using hydraulic rams that were powered by an electric pump. Each ram was calibrated prior to testing using the Tinius Olsen universal testing machine. From the calibration process, suitable load-voltage coefficients were obtained which were included in the data acquisition software. These coefficients were used to convert the recorded pressure reading from the hydraulic rams into suitable force measurements.
3.2.5 Lateral Loading Phase

Lateral loading of the ROPS was performed using a 50 tonne Enerpac hydraulic ram that was mounted to the horizontal loading beam of the testing frame. The load was positioned 235mm from the inside face of the left hand side ROPS post which corresponded to a vertical projection of the DLV, as required by the standard. Load spreading to prevent localized deformation of the ROPS material directly behind the load application point, was achieved through using a unique system that consisted of two 50mm thick steel reinforced elastomeric bearings that were sandwiched between two 25mm thick steel plates. Details of this system are shown clearly in Figure 3-6.

The specialised load spreading system was devised as a result of the expected multidirectional displacement that the ROPS would undergo during application of the lateral load. The nature of the ROPS and its sole point of fixity, indicated that it would move significantly forward, upward and sideward during application of the lateral load. Concern was raised over the expected eccentric loading of the hydraulic jack and its probable damage during this loading sequence. The incorporation of the bridge bearing system meant that the localized shear deformation capabilities of each
bearing could be utilised to transfer the load from the jack into the ROPS. This arrangement allowed the jack to remain stationary during application of the lateral load which therefore prevented any damage from occurring.

3.2.6 Lateral Load Deflection Response

The lateral load was applied to the side of the ROPS gradually up to the minimum requirement specified by the standard. Examination of the resulting load deflection profile at this level of loading indicated that the ROPS had not absorbed enough energy to fulfil the energy criteria of the standard. A substantial increase in both the lateral load and the lateral deflection of the structure was needed to achieve this energy requirement. Loading was subsequently continued until the area under the load deflection profile equated to approximately 9634 Joules. This requirement was achieved at a peak load of 325 kN and a corresponding lateral deflection of 53 mm. Once the energy criteria had been satisfied, the ROPS was unloaded with a resulting permanent deflection of 13 mm in the lateral direction. The load deflection response of the ROPS displayed a gradient change at a lateral load level of 200 kN. This reduction in stiffness, was characterised by the presence of yielding within the structure which has been exemplified by the strain rosette recordings which are...
shown in Figure 3-10 and Figure 3-12. The load deflection response of the ROPS at each LVDT location is also shown in Figure 3-7 and Figure 3-8 whereas Figure 3-9 highlights the deflected shape of the ROPS under full lateral load.

![Figure 3-7 Lateral Load Deflection Response (LVDT 1)](image)

![Figure 3-8 Lateral Load Deflection Response (LVDT 2, 3 and 4)](image)
3.2.7 Strain Rosette Recordings

The permanent plastic deformation sustained by the ROPS was monitored accurately with the strain rosette readings taken at consistent intervals throughout the test. Visual inspection of the ROPS indicated the presence of significant yielding predominantly at the base of the right post as viewed from the rear in accordance with Figure 3-9. The material on the outside face of the post showed significant signs of plastic hinge formation with the presence of an inward buckle shape taking place at this location. Reference to the strain gauge readings in this zone particularly at strain gauge locations D and F enforce the presence of distinct yielding within this region. The principal strains that were obtained at each of these gauge locations have also been shown in Figure 3-10 and Figure 3-12, whereas Figure 3-11 and Figure 3-13 show the corresponding Von Mises stress distribution throughout the ROPS at these corresponding locations.
Figure 3-10 Strain Gauge D Principal Strains

Figure 3-11 Strain Gauge D Von Mises Stress
3.2.8 Vertical Loading Phase

A 150 tonne Enerpac hydraulic jack was used to deliver the required vertical load to the ROPS. The jack was mounted to a loading beam located directly above the ROPS and was positioned in line with the back face of the DLV. The load distribution device used for the vertical loading phase employed the use of three steel reinforced
bridge bearings that were sandwiched in between two 25mm thick plates. A ball and seat arrangement was similarly applied to the end of the jack and the combination of these two systems accounted for the predicted large forward and vertical deflection of the ROPS whilst minimising damage to the jack. A stiff RHS loading beam was used to distribute the load evenly to each arm of the ROPS. An illustration of this load distribution device is presented in Figure 3-14.

3.2.9 Vertical Load Deflection Response

The load was applied gradually with the vertical load deflection response recorded at regular intervals using the data acquisition system. Figure 3-15 shows the load deflection response of the ROPS measured by LVDT 3, whereas Figure 3-16 shows the vertical load deflection response about the other axes measured by LVDT’s 1, 2 and 4. The deflected shape of the structure under the full vertical load is displayed clearly in Figure 3-17 and in addition to this the shear deformation of the bridge bearing load distribution system is also highlighted in Figure 3-18. The 490 kN vertical load requirement was reached at a vertical deflection of 48mm. Once this load requirement was achieved the pressure in the jack was released which resulted in approximately 14mm of permanent vertical deflection in the structure.
Figure 3-15 Vertical Load Deflection Response (LVDT 3)

Figure 3-16 Vertical Load Deflection Response LVDT 1, 2 and 4
Figure 3-17 Deflected Shape of ROPS under Full Vertical Load

Figure 3-18 Transverse Shear Deformation of Load Distribution Device
3.2.10 Longitudinal Loading Phase

The final loading phase involved the application of the longitudinal load by means of an Enerpac 10t hydraulic jack. This jack was mounted to the loading frame and was positioned at the midpoint of the front horizontal cross member of the ROPS. The jack was accommodated with a ball and seat at each end to prevent damage during loading. Load spreading was permitted through the inclusion of a 50mm thick section of solid steel on the front face of the horizontal cross member of the ROPS. Details of the loading system for this phase have been shown in Figure 3-21.

3.2.11 Longitudinal Load Deflection Response

The load was gradually applied with the displacements and strains recorded at regular intervals until the code specified load was achieved. The load deflection response of the ROPS measured by each LVDT during this testing phase has been shown in Figure 3-19 and Figure 3-20, whereas the deflected shape of the structure has been shown in Figure 3-21. The 79 kN longitudinal load requirement was reached at a corresponding longitudinal deflection of 48mm. Unloading of the ROPS after the application of this load resulted in a permanent longitudinal deflection of 10mm.

![Figure 3-19 Longitudinal Load Deflection Response LVDT 4](image-url)
Figure 3-20 Longitudinal Load Deflection Response LVDT 1, 2 and 3

Figure 3-21 Deflected Shape of ROPS under Full Longitudinal Load
3.3 Testing of ½ Scale ROPS for K275 Bulldozer

The K275 bulldozer is a crawler type dozer with a gross vehicle weight of approximately 50 tonnes. This type of dozer is commonly used in the construction and mining industries for earthmoving purposes. Rollover protection for the occupant is afforded through a two post rollbar type ROPS, which is clearly shown in Figure 3-22.

![Figure 3-22 K275 Bulldozer with ROPS](image)

3.3.1 ROPS Description

The ROPS for the K275 bulldozer is primarily a fixed base portal frame consisting of two posts and a beam that are rigidly connected to the chassis of the vehicle. In addition to the ROPS, an additional roof canopy section which is known as the FOPS (Falling Object Protective Structure), is incorporated to provide protection to the operator under falling objects. For the purposes of this study, the FOPS, which is a separate detachable structure, was omitted from the testing program conducted on this ROPS. The overall geometry of the full scale K275 ROPS model was established from site measurements taken from a K275 dozer located at the manufacturer’s storage yard. Similar to the 630E ROPS, a half scale model representation of the K275 ROPS was fabricated by the PTE group and subjected to the loading and energy requirements of AS2294-1997. The scaling laws established from the similitude study that was conducted for the 630E ROPS were similarly applied to this
particular ROPS with the aims of reducing fabrication costs and test load intensities. The member types used for the ROPS consisted of 350 grade RHS with full penetration butt welded moment resisting connections.

3.3.2 Experimental Investigation

The half scale K275 ROPS model was rigidly mounted to the same testing frame used for the 630E ROPS. Slight modifications were made to the testing frame to accommodate the new ROPS which included the raising of the lower cross head along Grid B and mounting of the lateral loading ram to the 400WC column located at the intersection of Grids 3 and D. Attachment of the ROPS to the loading frame was performed using a full penetration butt weld between the base of the ROPS and the lower crosshead located along Grid 3. The attachment of the ROPS to this stiff cross head was designed to simulate full base fixity and to replicate the fixity condition of the in-service ROPS/chassis assembly of the K275 bulldozer. Details of the test setup outlining the location of ROPS with respect to the testing frame are displayed graphically in Figure 3-23.

![Figure 3-23 Half Scale K275 ROPS Attachment to Loading Frame](image)

The ROPS was subjected to the loading and energy requirements of AS2294-1997 which similarly involved the consecutive loading and unloading of the ROPS about
its lateral, vertical and longitudinal axes. The load and energy magnitudes established from Table 1 of AS2294.2-1997 were modified to take into account the similitude relationships established previously. Table 3-3 presents a summary of the loads and energy criteria that were determined for the half scale K275 ROPS.

Table 3-3 AS2294.2-1997 Load and Energy Requirements for Half Scale K275 ROPS

<table>
<thead>
<tr>
<th>ROPS Performance Criteria</th>
<th>AS2294.2-1997 Formula with Scaling Provision Allowed</th>
<th>Result for Half Scale Model</th>
</tr>
</thead>
<tbody>
<tr>
<td>FLateral-Min</td>
<td>[\frac{1}{4} \cdot \frac{3}{7} \cdot 70000 \cdot \frac{M}{10000}^{1.2}]</td>
<td>121 kN</td>
</tr>
<tr>
<td>ULateral</td>
<td>[\frac{1}{8} \cdot \frac{13}{3} \cdot 10000 \cdot \frac{M}{10000}^{1.25}]</td>
<td>12104 J</td>
</tr>
<tr>
<td>FVertical</td>
<td>[\frac{1}{4} \cdot 19.61M]</td>
<td>245 kN</td>
</tr>
<tr>
<td>FLongitudinal</td>
<td>[\frac{1}{4} \cdot \frac{3}{5} \cdot 56000 \cdot \frac{M}{10000}^{1.2}]</td>
<td>96 kN</td>
</tr>
</tbody>
</table>

3.3.3 Instrumentation and Measurement Parameters

The data acquisition system that was employed for the previous 630E ROPS test was similarly used for the K275 test. Five strain rosettes in total were attached to the top and base of each post of the ROPS and strain recordings were taken at consistent intervals during the lateral loading stage only. Deflection of the ROPS during each loading stage was measured using LVDT’s that were positioned in the direction of the applied load. Each LVDT was securely mounted to an independent steel frame which enabled true deflection readings to be taken that were free of any loading frame movement. Figure 3-24 shows the position of the strain rosettes and the LVDT’s throughout the ROPS. Load generation for each loading stage was similarly performed using hydraulic rams that were powered by an electric pump.
3.3.4 Lateral Loading Phase

Lateral loading of the ROPS was performed gradually using a 50 tonne hydraulic jack that was securely mounted to the loading frame. The load was applied to the upper portion of the right hand side post of the ROPS as viewed from the rear in accordance with Figure 3-25. Load spreading to alleviate localised deformation was performed using a 100x100x20 section of mild steel plate. A double ball and seat arrangement which is also shown in Figure 3-25 was incorporated into the loading system to prevent damage to the hydraulic jack.
3.3.5 Lateral Load Deflection Response

The half scale K275 ROPS was tested using the same principles as the 630E ROPS. Similar to the previous test on the 630E ROPS, the energy absorption requirement of the standard had not been reached after attainment of the minimum lateral load of 121 kN. Clearly from inspection of the resulting load deflection profile which is shown in Figure 3-26, the ROPS was still predominantly in its elastic state with very little deflection taking place and therefore little energy absorption as well. Loading was increased up to approximately the 200 kN mark, which saw the initiation of plastic hinges at the top and base of each post. The load deflection profile of the ROPS plateaued for a short period of time and then began to fall gradually until it reached a peak deflection of 70mm at approximately the 175 kN mark. At this level of deflection, the area under the load deflection profile had equated to the code requirement of approximately 12100 J. During the loading sequence the ROPS had undergone significant plastic deformation. The resulting load deflection response shown in Figure 3-26 indicated 60mm of permanent plastic deformation in the lateral direction. The most clearly defined plastic hinge in the structure was located at the base of the left post as viewed from the rear in accordance with Figure 3-27. Figure
3-28 shows the extent of the localised plastic hinge formation in this region. Strain rosette D was located within this region and clearly monitored the yielding in this region.

![Figure 3-26 Lateral Load Deflection Response (LVDT 1)](image)

Figure 3-26 Lateral Load Deflection Response (LVDT 1)

![Figure 3-27 Deflected Shape of ROPS under Full Lateral Load](image)

Figure 3-27 Deflected Shape of ROPS under Full Lateral Load
3.3.6 Strain Rosette Recordings

The permanent plastic deformation sustained by the ROPS was monitored accurately with the strain rosette readings taken at consistent intervals throughout the test. As discussed previously, visual inspection of the ROPS showed the presence of plastic hinge formation at the top and base of each post. The readings taken during the test, further emphasised the extent of yielding in these zones. Figure 3-29, Figure 3-31, Figure 3-33 and Figure 3-35 show the principal strains recorded at these locations and their relationship with increasing lateral load. In addition to this, the corresponding Von Mises stress profile and its relationship to increasing lateral load is presented at each gauge location in accordance with Figure 3-30, Figure 3-32, Figure 3-34 and Figure 3-36.
Figure 3-29 Strain Gauge A Principal Strains

Figure 3-30 Strain Gauge A Von Mises Stress
Figure 3-31 Strain Gauge B Principal Strains

Figure 3-32 Strain Gauge B Von Mises Stress
Figure 3-33 Strain Gauge C Principal Strains

Figure 3-34 Strain Gauge C Von Mises Stress
3.3.7 Vertical Loading Phase

Similar to the vertical loading phase conducted on the 630E ROPS, the same 150 t hydraulic jack was employed to enforce the required vertical load for this loading sequence. Load spreading, to alleviate localised deformation was performed using a 250x125x10 section of flat mild steel that was tack welded to the top of the ROPS.
beam. A ball and seat arrangement was again employed to prevent any eccentric loading of the jack. Details of the loading system and the load spreading device that was used for this loading phase are presented in Figure 3-38.

3.3.8 Vertical Load Deflection Response

Loading of the ROPS in its already pre-deformed position, was gradual and resulted in a load deflection response profile that is shown in accordance with Figure 3-37. It is evident from reference to this figure that the ROPS had exhibited a fairly stiff response with only 8mm of vertical deflection taking place at the point of maximum vertical load. It is also clear from reference to this figure that the residual amount of permanent vertical deflection in the ROPS was very small after release of the pressure in the jack. The deflected shape of the structure under the applied vertical load is shown in Figure 3-38. Reference to this figure shows little deformation in the vertical direction as the structure’s behaviour has been predominantly influenced by the previous lateral loading phase. It is clear from the results of this loading phase, that a structure that had undergone significant plastic deformation was still able to withstand a further loading sequence.

![Figure 3-37 Vertical Load Deflection Response (LVDT 2)](image-url)
3.3.9 Longitudinal Loading Phase

The final loading sequence involved application of a load in the direction of the longitudinal axes of the ROPS. A 50 tonne hydraulic ram was used to apply the necessary longitudinal load to the midpoint of the horizontal beam that linked each post. Load spreading was achieved through use of a 125x125x20 section of flat mild steel and a ball and seat arrangement was employed to alleviate any eccentric loading of the jack.

3.3.10 Longitudinal Load Deflection response

The load deflection response displayed by the structure is shown in Figure 3-39. This response behaviour indicated a maximum deflection of the ROPS in the lateral direction of approximately 20mm with a permanent deflection of 7 mm being sustained during unloading. The deflected shape of the ROPS under full longitudinal load is shown in Figure 3-40.
Figure 3-39 Longitudinal Load Deflection Response (LVDT 3)

Figure 3-40 Deflected Shape of ROPS under Full Longitudinal Load
3.4 Testing of ROPS for Powertrans Dump Truck

A rollover protective structure suitable for attachment to a Powertrans 13.5 tonne dump truck was tested at full scale using the static loading procedure outlined in AS2294-1997. This particular ROPS was substantially different from those tested previously as it involved the complete replacement of the vehicle’s cabin with a four post moment resisting frame that was fabricated from 350 grade SHS. Pinned connections were used to attach the ROPS to a modified version of the vehicle’s chassis which consisted of two large welded beams that were rigidly attached to the loading frame. Figure 3-41 shows an illustration of the ROPS and associated vehicle chassis.

3.4.1 Experimental Investigation

The loads and energy requirement for the ROPS were obtained by using table 1 of AS2294-1997. Table 3-4 presents a summary of the equations used and the resulting values obtained for this particular ROPS.
Table 3-4 AS2294.2-1997 Load and Energy Requirements for Half Scale K275 ROPS

<table>
<thead>
<tr>
<th>ROPS Performance Criteria</th>
<th>AS2294.2-1997 Formula</th>
<th>Result for Full Scale ROPS</th>
</tr>
</thead>
<tbody>
<tr>
<td>( F_{\text{Lateral-Min}} )</td>
<td>( \frac{85000}{10000} M )</td>
<td>122 kN</td>
</tr>
<tr>
<td>( U_{\text{Lateral}} )</td>
<td>( \frac{15000}{10000} M )</td>
<td>21828 J</td>
</tr>
<tr>
<td>( F_{\text{Vertical}} )</td>
<td>19.61 ( M )</td>
<td>265 kN</td>
</tr>
<tr>
<td>( F_{\text{Longitudinal}} )</td>
<td>( \frac{68000}{10000} M )</td>
<td>95 kN</td>
</tr>
</tbody>
</table>

3.4.2 Test Rig Setup

The ROPS and associated chassis were rigidly mounted to a specially fabricated test rig that could safely apply the required lateral, vertical and longitudinal loads. The loading frame relied on equilibrium being established between the jacking forces transmitted to the ROPS and the reaction forces taken by the frame. The implementation of such an arrangement, ensured that the only net transfer of load into the ground was due to the self-weight of the ROPS, chassis and loading frame which meant that no tie down or lateral bracing system was required.

3.4.3 Instrumentation and Measurement Parameters

The data acquisition system used for each loading sequence was identical to that used previously. For this application, 19 Strain rosettes were positioned throughout the structure in regions of predicted high levels of strain and recordings were taken at consistent intervals during the lateral loading sequence. LVDT probes were positioned at locations in the structure that experienced the highest deflections during each loading sequence and recordings were similarly taken at consistent intervals. The load application device used for each loading phase involved the use of a 50t capacity Enerpac hydraulic RAM that was powered by an electric pump.
3.4.4 Lateral Loading Phase

The lateral load was applied to the upper horizontal side member of the ROPS at a position located 700mm from the inside face of the front vertical post. The hydraulic ram used to apply the lateral load, incorporated a ball and seat that was housed in a section of CHS, that was tack welded to the face of the ROPS. Figure 3-43 shows the position of the hydraulic ram located against the side face of the ROPS under the applied lateral load.

3.4.5 Lateral Load Deflection Response

Loading of the ROPS was gradual with displacements and strains being recorded at consistent intervals until the code specified minimum lateral load of 122 kN was achieved. Examination of the resulting load deflection response profile which is shown in Figure 3-42 indicated that at this level of loading, the ROPS was still within its elastic range and had not absorbed enough energy to fulfil the requirements of the standard. Loading was continued until the energy absorption criterion was achieved. This requirement occurred at a lateral load of 280 kN and a corresponding lateral deflection of 145mm. Figure 3-43 shows the deflected shape of the ROPS under the full lateral load. Examination of this resulting load deflection profile indicated that the structure behaved elastically until it reached a load of approximately 225 kN. Upon reaching this load, yielding began to occur and the structure began to plastically deform in order to dissipate energy. The degree of plastic deformation experienced by the ROPS was characterised by a significant increase in the lateral displacement of the structure in the load range between 225 kN and 279 kN. This can clearly be seen from the load deflection response where the structure deflected approximately 70mm within this loading range. Once the ROPS had absorbed the required amount of energy, it was unloaded which resulted in 36mm of permanent lateral deflection.
3.4.6 Strain Gauge Results for Lateral Loading Sequence

The strain gauge readings taken during the lateral loading sequence indicated the presence of yielding in the structure predominantly at the top of each corner post. The principal strain and the corresponding Von Mises stress distribution recorded at
strain gauges G and Q have been presented in accordance with Figure 3-44, Figure 3-45, Figure 3-46 and Figure 3-47. Examination of each graph indicates the onset of yielding at a lateral load of approximately 225 kN which follows closely the results obtained from the lateral load deflection graph for the entire structure shown in Figure 3-42.
3.4.7 Vertical Loading

The vertical load determined in accordance with the standard was applied to each upper horizontal side member of the ROPS at a position located 700mm from the inside face of the front vertical post. A stiff spreader beam was used to transfer the
load from the hydraulic ram to each horizontal cross member of the ROPS, as shown in accordance with Figure 3-49. The load was applied gradually and displacements were recorded at the load application point in regular intervals until the code specified value of 265 kN was achieved. Figure 3-49 illustrates the corresponding load deflection response displayed by the ROPS for each level of loading.
Deflections experienced by the ROPS in the vertical direction were reasonably small compared with those obtained during the lateral loading phase. Examination of the load deflection response profile displayed by the ROPS during the test indicated a fairly linear elastic response with only minimal permanent deformation taking place.

### 3.4.8 Longitudinal Load

Application of the final longitudinal load to the ROPS was performed using a single hydraulic jack positioned at the midpoint of the front horizontal cross member of the ROPS. Application of the load was gradual and the displacements were recorded at the load application point in regular intervals until the code specified value of 95 kN was achieved. The deflected shape of the structure under the full longitudinal load can be seen in Figure 3-50, whereas Figure 3-51 displays the corresponding load deflection response exhibited by the ROPS during the test. Examination of this load deflection response indicated that the behaviour of the ROPS was predominantly linear with little permanent residual deflection taking place upon unloading.

![Deflected Shape of ROPS under Full Longitudinal Load](image-url)
Figure 3-51 Longitudinal Load Deflection Response (LVDT 1)

3.5 Chapter Summary

Three ROPS were tested in accordance with the static load provisions outlined in AS2294.2-1997. The first test which was conducted at half scale, involved testing of a two post cantilever type ROPS. This ROPS passed the requirements of the standard with reference to load, deflection and energy absorption criteria, however, was found to exhibit a fairly stiff response. The energy requirement was reached at a lateral load that was approximately three times the minimum requirement of the standard. Whilst this response behaviour was within acceptable limits, it is believed that response behaviour such as this may be undesirable for the occupant during the occurrence of a rollover as it leads to the development of large reaction forces which may carry with them large peak deceleration as the vehicle attempts to come to rest. The second test was for a two post rollbar type ROPS that was suitable for attachment to a 50 tonne K275 bulldozer. This ROPS also successfully passed the requirements of the standard, however, the stiffness distribution was low and this ROPS was more flexible than expected. The energy requirement for this ROPS was fulfilled using a low force/ high deflection response. Despite the formation of hinges at the described locations, the ROPS possessed sufficient capacity to withstand the subsequent vertical and longitudinal loading sequences without violation of the DLV. The third
test involved subjecting the four post replacement type ROPS of the Powertrans dump truck to the requirement of the standard. This ROPS also passed the requirements of the standard successfully, however, its response was characterised by a high force/low deflection demand in order to dissipate the necessary quantity of energy.

Overall, this chapter has showed that a variety of ROPS configurations may be tested using the procedures outlined in the current Australian Standard for Earthmoving machinery AS2294.2-1997. The test results have shown that ROPS designs are stiff and are commonly fabricated to withstand forces that are as much as three times the minimum specified level present within the standard. A more flexible design approach that allows the structure to adequately develop plastic hinges at specified locations during application of the lateral load, has also been tested and has been found to be successful. The use of an appropriate level of stiffness in ROPS design will be tested further using FEA in chapter 4, 5 and 6. It is hoped that this numerical modelling study will assist ROPS designers to develop appropriate safety frames that provide adequate energy dissipating capabilities that will enhance an operator’s chances of survival during the occurrence of a rollover.
Chapter 4 Finite Element Analysis of ROPS

This chapter outlines the use of finite element analysis software (FEA) to model and analyse the response behaviour of three different ROPS configurations subjected to static loads in accordance with the loading and energy requirements outlined in the Australian Standard for Earthmoving Protective Structures AS2294.2-1997. Within the scope of this chapter a detailed discussion is presented on the current numerical analysis techniques that are used by finite element solvers to obtain solutions to highly nonlinear problems such as for the case of a rollover protective structure. In addition to this discussion further comment is also made on the importance of appropriate program selection.

Detailed three dimensional finite element models that accurately reflected the correct stiffness distribution of each in service ROPS were developed for study. Results obtained from the FE analysis for two of the models were compared against their corresponding experimental values from the testing phase performed in chapter 3. This calibration process was performed in order to permit further dynamic FE modelling to take place which has been addressed in chapters 5 and 6. Comparison of the FE and experimental results has also been performed to determine whether FEA can be used as a valuable tool for certifying ROPS, thus lessening the need for destructive full scale testing. Examination of the critical parameters that control the energy absorption capabilities of a ROPS and its ability to fulfil the requirements of the standard has also been studied within the chapter and reported accordingly.

4.1 Finite Element Software

Throughout the duration of this project, two FEA programs were employed to carry out the necessary analyses, namely ABAQUS standard v6.3 and LS-DYNA v970. The capabilities of both of these programs are extremely versatile with the ability to perform highly specialised analyses for a variety of applications. ABAQUS standard is a general purpose FEA package, with the ability to model both linear and highly nonlinear structural problems. ABAQUS Standard uses the implicit solution method to solve the governing equilibrium equations and involves the division of the problem history into steps and their corresponding incremental solution using the
Newton Raphson procedure. LS-DYNA, is similarly a general purpose FEA package that is particularly effective in solving transient dynamic problems. This package, whilst possessing limited implicit capabilities, predominantly uses an explicit solution procedure to solve both linear and nonlinear problems.

The terms implicit and explicit refer to the numerical time integration scheme adopted by each package to solve for the unknown displacement solution, which is the basis for calculating resulting strains and stresses. An implicit integration scheme as employed by ABAQUS, assumes a constant average acceleration over each time step, between time \( t_n \) and \( t_{n+1} \). The term \( t_n \) is the time at the beginning of each time step whereas the term \( t_{n+1} \) represents the time at the end of each time step. The governing equations are evaluated and the resulting accelerations and velocities at time \( t_{n+1} \) are determined. Once these parameters have been evaluated, the unknown displacements at time \( t_{n+1} \) may be calculated. The explicit integration scheme employed by LS-DYNA, is a central difference method which assumes a linear change in displacement over each time step. The governing equation is evaluated and the resulting accelerations and velocities at time \( t_n \) are calculated which leads to determination of the unknown displacements at time \( t_{n+1} \).

The numerical technique that is used by each program to solve for the unknown displacements at time \( t_{n+1} \) differs vastly. An implicit solution procedure involves inversion of the structural stiffness matrix and multiplication with the nodal forces to determine the corresponding nodal displacements. Once the nodal displacements have been calculated, they are used to test and verify the governing equilibrium equations. A test for convergence between the external and internal energies must then be reached in order for the solver to incrementally increase the time step and progressively move toward obtaining a final solution to the problem. The advantages of a solution procedure such as this, is that the user has control over the time step size which makes it especially suitable for the solution of long duration quasi-static problems, which would be very computationally expensive if an explicit solver was used. The disadvantages of using the implicit solution procedure are outlined as follows:
Convergence can be a problem for highly nonlinear problems as an equilibrium condition must be obtained for the solver to incrementally increase the time step.

Difficulties can arise in achieving equilibrium states particularly for discrete bodies experiencing contact.

Memory requirements can be large.

CPU requirements can be large.

The explicit solution procedure relies on the summation of the internal and external forces at each node and their corresponding division by the nodal mass to find the resulting nodal acceleration. The solution is advanced by integrating the acceleration with respect to time, where the time step size is limited by a requirement known as the Courant condition which stipulates that the distance travelled by an infinitesimal wave in one time step must never exceed the distance between computational nodes. The restriction placed on the time step size by this condition, gives rise to a solution that involves a large number of small time steps, which makes the explicit solution technique more suitable for short duration transient problems. Whilst a large number of small time steps are required for a solution procedure such as this, the explicit solution technique can be significantly more efficient than that of an implicit solution as no matrix inversion is required. The advantages and disadvantages of this solution procedure are outlined as follows:

- The number of operations / steps are small
- Memory requirements are small
- The procedure is robust
- No tangents are required therefore promoting computational simplicity
- No checks are made for equilibrium
- The stability of the problem is checked by monitoring energy
- Approximations are made within the software

Both packages were used extensively throughout this project to model the behaviour of a series of ROPS configurations to a variety of loading conditions. ABAQUS was employed primarily to model the response of three ROPS configurations to static loads determined in accordance with the procedure used in AS2294.2-1997. The ROPS configurations that were chosen for modelling with ABAQUS included:

- A two post cantilever ROPS suitable for attachment to a 125 tonne 630E rigid frame dumper
A two post rollbar ROPS suitable for attachment to a 50 tonne K275 bulldozer.

A two post cantilever ROPS suitable for attachment to a 50 tonne Wa600 wheeled loader.

For all static analyses, the pre and post processing software package MSC Patran 2004 was used to construct each finite element model and visualise the results after each FE computer simulation. The Queensland University of Technology’s Super Computer, Sirius, which uses an IRIX operating system was employed to run all of the necessary ABAQUS models. Dynamic modelling of two of the ROPS configurations was later performed using the finite element analysis package LS-DYNA. This program was operated on a PC using the Microsoft Windows XP operating system. Details of the modelling procedure for these dynamic simulations as well as the results that were obtained are outlined in chapters 5 and 6.

4.2 FEA of ½ Scale 630E Dump Truck ROPS

A three dimensional finite element model of the ½ scale 630E ROPS that was tested in chapter 3, was developed and subjected to static loads about the model’s lateral, vertical and longitudinal axes in accordance with the loading requirements of AS2294-1997. As discussed previously, the finite element analysis code ABAQUS V6.3 was used to perform the necessary numerical modelling simulation.

4.2.1 Model Setup

The geometry of this ROPS was established from the shop drawings that were produced by the manufacturer of the corresponding ½ scale test model ROPS. Once the dimensions of the ROPS had been accurately obtained, the solid modelling software package SOLIDWORKS 2004 was used to establish the geometrical definition of the ROPS. The established geometry was based on the mid thickness surface of each section that was used to construct the structure of the ROPS. Use of this package permitted the corner radii regions of each RHS/SHS member to be included as well as providing a convenient way to construct the difficult geometry associated with this type of ROPS configuration. The model consisted predominantly of two main parts which included the ROPS and the associated tapering box beam that was rigidly connected to the base of each post. Once the geometry was completed in SOLIDWORKS, it was then imported into the pre-processor MSC.
PATRAN where the mid thickness outer surface of the model was meshed using quadrilateral shell elements. Figure 4-1 shows a graphical illustration of the geometry used for the \( \frac{1}{2} \) scale 630E ROPS model.

![Figure 4-1 – \( \frac{1}{2} \) scale 630E ROPS model](image)

4.2.2 Elements Definition

The ABAQUS shell element type S4R was selected to model the response of the ROPS to the established static loading requirements of the standard. The S4R element is a four node, quadratic, stress/displacement shell element with reduced integration. This particular element type is a general purpose element which can account for finite membrane strains and will allow for the change in thickness of the shell element when yielding of the section takes place. The enhanced capabilities of this element made it particularly suitable for modelling the behaviour of a structure such as a ROPS, as these structures undergo extensive plastic deformation during
loading. The FE model which is shown in Figure 4-2 was discretised into approximately 10mm square elements using the isomesh capability within Patran.

![Figure 4-2 - ½ scale 630E ROPS Element Definition](image)

### 4.2.3 Material Properties

The ABAQUS material model that was used to simulate the material behaviour during loading was a classical metal plasticity model that employed a Mises yield surface with an associated plastic flow and isotropic hardening condition. Use of such a material model required that the material properties beyond yield be defined in terms of true stress and plastic strain. The material properties that were selected for the FE model were based on uniaxial tensile tests conducted on specimens taken from the same RHS/SHS and mild steel plate that were used to construct the ½ scale ROPS that was tested in chapter 3. The material properties obtained from these tests, were converted into true stress and plastic strain using Equation 4-1 and Equation 4-2
suitable for input into ABAQUS. Figure 4-3 displays the engineering stress versus strain relationship that was obtained from the testing of the tensile specimens, whereas Figure 4-4 shows the corresponding true stress versus plastic strain hardening curve. In addition to these plastic material properties, the following elastic material properties were used to define the behaviour of the material in the elastic region.

- Material Density $\rho = 7850 \text{ kg/m}^3$
- Elastic Modulus $E = 200000 \text{ MPa}$
- Poisson’s ratio $\nu = 0.3$

It must be noted that the following equations are valid for incompressible materials which have a Poisson’s ratio equal to 0.5 and have therefore been used as an approximation for the tested material type that has a Poisson’s ratio equal to 0.3.

$$\sigma_{\text{true}} = \sigma_{\text{Eng}} \left( 1 + \frac{\nu}{2} \right)$$  \hspace{1cm} \text{(Equation 4.1)}

$$\sigma_{\text{plastic}} = \ln \left( 1 + \frac{\nu}{2} \right) \sigma_{\text{Eng}} + \frac{\sigma_{\text{true}}}{E}$$  \hspace{1cm} \text{(Equation 4.2)}

Figure 4-3 – 630E Dump Truck ROPS Material Properties
4.2.4 Section Properties

The section properties that were used for each member of the ROPS have been summarised in Table 4-1. The reader is referred to Figure 4-5 for an explanation of the terms that have been used.

Table 4-1 Half Scale 630E Dump Truck ROPS Section Properties

<table>
<thead>
<tr>
<th>ROPS Member</th>
<th>Member Type</th>
</tr>
</thead>
<tbody>
<tr>
<td>Posts</td>
<td>350 grade 150x150x6 RHS</td>
</tr>
<tr>
<td>Front Cross Beam</td>
<td>350 grade 125x75x5 RHS</td>
</tr>
<tr>
<td>Rear Cross Beam</td>
<td>350 grade 125x75x5 RHS</td>
</tr>
<tr>
<td>Side Beams</td>
<td>350 grade 125x75x5 RHS</td>
</tr>
<tr>
<td>Haunches</td>
<td>350 grade 125x125x5 SHS</td>
</tr>
<tr>
<td>Internal Stiffener plates</td>
<td>350 grade 6mm plate</td>
</tr>
<tr>
<td>Chassis Beam</td>
<td>350 grade 16mm plate</td>
</tr>
</tbody>
</table>
4.2.5 Boundary Conditions

In order to replicate the base fixity condition of the experimental model accurately, a stiff end plate was incorporated into the FE model and was positioned to the left end of the chassis beam. This plate, which was rigidly connected to the box beam, was restrained at twelve locations about each global translational degree of freedom. The position of each of these restraining points, corresponded to nodal positions that represented the centreline location of each bolt that was used to restrain the corresponding experimental model ROPS. Figure 4-6 shows the position of the restraining boundary conditions that were used in the FE model ROPS.

Figure 4-6 ½ Scale 630E Dump Truck ROPS Boundary Conditions
4.2.6 Loading Procedure

The loading procedure that is used in AS2294.2-1997 requires that the ROPS be subjected to three consecutive loading stages that involve loading and unloading of the ROPS in the direction of its lateral, vertical and longitudinal axes. An illustration showing the direction of each of these loading stages, is displayed graphically in Figure 4-7.

Figure 4-7 – ROPS Loading Sequences

The loading procedure for the FE model involved the application of equivalent face pressures to the surface of the ROPS at suitable locations that corresponded to those that were used in the experimental model. Excessive plastic deformation directly behind the load application point was prevented through the inclusion of an elastic strip. The use of such a strip meant that the elements within these zones could undergo elastic deformation only. Definition of the elastic strip in these zones was considered appropriate as load spreader plates had been used to prevent the occurrence of excessive plastic deformation during the testing of the experimental model.
In order to replicate the AS2294.2 loading procedure accurately, a solution procedure that had the ability to model the accumulated effects of plastic deformation that would be sustained by the ROPS during each loading sequence was required. In addition to this, it was necessary for the model to be capable of releasing the stored elastic deformation that would be experienced by the ROPS during unloading. The procedure that was used to model this behaviour was based on a load controlled technique that was guided by a unit load time history. Details of the loading histories that were used to guide the structure through the relevant loading stages of the standard are outlined in Figure 4-8.

![Figure 4-8 Unit Load Time Histories](image)

Converged solutions were obtained easily for the 630E ROPS model using this procedure, however, this solution method was later found to be inadequate when the ROPS exhibited a more flexible response that was characterised by extensive plastic deformation and large displacements. The convergence problems that were experienced during this type of FE simulation are commonly encountered when using an implicit solver to obtain a solution to a highly nonlinear problem. In order to overcome this numerical instability problem, a displacement controlled solution procedure was used for all further analyses. This procedure involved displacing the structure by a specified distance and measuring the corresponding base reactions to obtain the load deflection response profile. For a detailed discussion on the solution
algorithms that are used by modern FEA codes to obtain solutions to highly nonlinear problems, the reader is referred to research performed by Argyris (1965), Pian and Tong (1970) and Zienkiewicz (1971).

4.3 Results of the Numerical Study

The results from the FE simulation for the 630E model were compared with those from the corresponding experimental testing phase for each loading sequence. A comparison between the load deflection response of the FE model ROPS and the experimental model ROPS during the lateral loading phase has been shown in accordance with Figure 4-9. Clearly from this figure very good correlation between the two results has been achieved for this loading phase. It is evident therefore, that for the lateral loading sequence the FE model has captured the nonlinear response of the ROPS accurately.

![Figure 4-9 Comparison of FEA and Experimental Lateral Load Deflection Profiles](image)

Figure 4-9 Comparison of FEA and Experimental Lateral Load Deflection Profiles

The distribution of Von-Mises stress that occurred throughout the ROPS during application of the peak lateral load has been displayed graphically in Figure 4-10. The stress contours that are present within this figure, show that significant yielding took place in isolated zones that were located at the base of the left post and the ends of the rear cross beam that provided a link between the top ends of each post. The distinct lack of significant yielding at the base of the right post clearly exemplifies
the asymmetrical behaviour of the ROPS that has arisen due to the positioning of the structure’s supports.

![Figure 4-10 Von Mises Stress Distribution under Lateral Load](image)

After attainment of the code specified energy absorption level, the FE ROPS model was unloaded which resulted in a residual Von Mises stress distribution that has been shown in accordance with Figure 4-11. The blue and cyan stress contours which represent very low stress levels have been maintained throughout most zones of the ROPS after unloading. Some high residual stress have, however, been sustained by the model along the front face of the left post which is highlighted by the red and yellow colouring within this region. This result is to be expected as the degree of plastic deformation experienced by the model in this region was high.
4.3.1 Vertical Loading Phase

The load deflection response of the ROPS model that was obtained from the FE analysis was compared against its corresponding experimental values from chapter 3 and has been shown in Figure 4-12. It must be noted that both of these load deflection curves have been plotted with reference to the structures pre-deformed position after removal of the lateral load. Examination of this figure shows that the response behaviour is similar, however, it appears that the FE model has under-predicted the vertical deflection of the ROPS by approximately 15mm. Possible reasons for this inaccuracy may be attributed to the influence of residual stresses that exist within the RHS/SHS members that were used to construct the ROPS and have not been considered in the FE model. The presence of these residual may be due to the heavy welding and forming processes that were used during ROPS fabrication. The extent and distribution of these inherent residual stresses are difficult to quantify and model. Another possible cause for the lack of close agreement between these results may be due to the influence of the Bauschinger effect, which is a known problem that results in a reduced yield stress upon load reversal after the onset of
plastic deformation during the initial loading phase. Modelling of this effect is difficult and can not be accounted for with the traditional ABAQUS metal plasticity material model that has been used for this FE simulation.

![Graph showing vertical load vs. deflection comparison (FEA vs. Experimental)](image)

**Figure 4-12 Half Scale 630E Vertical Load Deflection Profile Comparison**

The Von Mises stress distribution throughout the ROPS that was obtained from the FE simulation for this loading phase has been shown in accordance with Figure 4-13. Within this figure, the orange and red contours highlight the yielding that was experienced by the ROPS during this loading phase. It is evident from examination of this figure, that the extent of the yielding that was experienced by the ROPS was most pronounced along the front face of the left post.
Figure 4-13 Von Mises Stress under Vertical Load

Similar to the previous loading sequence, the vertical load was released after the code specified loading level was attained. Removal of this load resulted in Von Mises stress distribution throughout the ROPS that is shown in Figure 4-14.
4.3.2 The Longitudinal Loading Phase

The load deflection response that was obtained for the FE model was not in close agreement with the corresponding experimental response. The FE model predicted a fairly linear response with only small amounts of yielding taking place in localized regions. It must be noted that it is difficult to calibrate a model over multiple loading sequences after the onset of significant plastic deformation during the first loading stage. With these views in mind, the reasons that were discussed previously for the vertical loading phase may be responsible for the lack of correlation between the results for this loading procedure. Whilst the correlation for this loading phase was poor it must be noted that this thesis is concerned predominantly with the behaviour of ROPS during the first sidewards impact of which the correlation was very good. In acknowledging this principal aim, no further attempt has been made to improve the correlation between the results for the vertical and longitudinal loading sequences.

Figure 4-14 Residual Von Mises Stress after Vertical Load Removal
4.4 FEA of a K275 Bulldozer ROPS

The second model that was studied using finite element analysis was the ½ scale K275 ROPS that was tested in chapter 3. ABAQUS was again employed to perform the necessary numerical simulation under the static force requirements of AS2294.2-1997. A similar full scale FE model of the ROPS was also developed and analysed numerically in order to verify that the established similitude relationships that were derived in chapter 3 were accurate and could be applied to the design and analysis of ROPS. A calibration process was also performed on this ROPS by comparing the load deflection response profiles of the numerical and experimental models. This calibration process was used to ascertain the accuracy of the FE modelling procedure and to permit further dynamic impact investigations to be undertaken with confidence in chapters 5 and 6.

4.4.1 Model Setup

The geometry of the full scale FE ROPS model was established from site measurements taken from a K275 dozer located at the manufacturer's storage yard. After attainment of these measurements, appropriate RHS/SHS member sizes were selected so that the ROPS would possess sufficient strength and energy absorption characteristics that would enable it to successfully pass the requirements of the Australian Standard. Scaling laws that were derived from a similitude study of the ROPS were used to develop the necessary geometry for the FE model. MSC Patran was employed to construct the geometry and the necessary parameters for development of the finite element model. For this particular model, the corner radius for each member of the ROPS was omitted. This was performed in order to simplify the modelling process of the connection region between the posts and beam of the ROPS. The influence of this omission on the global energy absorbing capabilities of the ROPS was studied within this chapter and found to be negligible. Figure 4-15 shows the geometry of the ROPS that was used for the FE model. It must be noted that the roof section which also serves as a falling object protective structure (FOPS) for the vehicle has been omitted from the analysis.
4.4.2 Elements

The meshing procedure that was used for the model was similar to the one employed previously for the 630E ROPS. A 10 mm and 5 mm mesh density were selected for the full scale prototype and half scale model ROPS respectively. The ABAQUS S4R shell element was chosen to simulate the response behaviour of each ROPS under the established loading conditions. The shell thickness that was used throughout each FE model was 10mm for the prototype and 5mm for the model ROPS. Figure 4-16 shows the mesh discretisation for the full scale prototype ROPS.
4.4.3 Material Properties

The ABAQUS classical metal plasticity material model that was used previously for the 630E ROPS was again employed for this application. The material properties for each model were obtained from tensile testing of specimens that were cut from the experimental model.

4.4.4 Section Properties

Table 4-2 outlines the section properties that were used for the model and prototype ROPS.

<table>
<thead>
<tr>
<th>Member</th>
<th>Full scale K275 prototype</th>
<th>½ scale K275 model</th>
</tr>
</thead>
<tbody>
<tr>
<td>Post</td>
<td>350 grade 250x150x10 RHS</td>
<td>350 grade 250x150x5 RHS</td>
</tr>
<tr>
<td>Beam</td>
<td>350 grade 250x250x10 SHS</td>
<td>350 grade 250x250x5 SHS</td>
</tr>
</tbody>
</table>
4.4.5 Boundary Conditions

The boundary conditions for each FE model were designed to simulate full base fixity and involved the implementation of translational and rotational restraints about the global X, Y and Z axes of the ROPS. Implementation of this type of boundary condition was performed using a fixed multipoint constraint (MPC) that tied the base perimeter dependent nodes of each post to a single independent node located at the base centroid of each post. This independent node was restrained both translationally and rotationally about all global degrees of freedom. This restraint condition was numerically identical to restraining all base perimeter nodes, however, it enabled the resultant reaction forces to be obtained at two single nodes only, which assisted greatly in developing the resulting load deflection response profiles for each loading condition. Figure 4-17 shows a graphical representation of this boundary condition at the base of the left post of the ROPS.

![Figure 4-17 Boundary Condition Applied to MPC at Base of Left Post](image)

4.4.6 Loading Procedure

The loading procedure for each model involved using a displacement controlled method for the lateral loading phase and a load controlled procedure for the vertical and longitudinal loading phases. Details of the loading procedure that simulated the requirements of the standard are outlined as follows:
Stage 1 of the analysis was comprised of the following procedures:

- Application of an arbitrary enforced displacement to the ROPS in the lateral direction and measurement of the base reactions to obtain the load deflection response.
- Determination of the energy absorbed by the ROPS for the applied deflection from the area under the load deflection curve and determination of the correct enforced displacement to apply to the ROPS to obtain the energy level requirement in accordance with AS2294.2-1997.

Stage 2 of the analysis involved subjecting each FE model to six loading stages in sequence which has been summarised below:

- Load Case 1: Application of the determined enforced displacement in the lateral direction necessary to give the required amount of energy absorption.
- Load Case 2: Unloading of the ROPS through application of the base reactions to the model.
- Load Case 3: Application of the required vertical load to the ROPS using face pressures distributed over an appropriate area.
- Load Case 4: Unloading of the ROPS through application of the base reactions to the model.
- Load Case 5: Application of the longitudinal load to the ROPS using equivalent face pressures distributed over an appropriate area.
- Load Case 6: Unloading of the ROPS through application of the base reactions to the model.

### 4.5 Results of Numerical Study

#### 4.5.1 Lateral Loading Phase

Finite element simulations were performed on the ½ scale model and full scale prototype K275 ROPS using the Finite element analysis code ABAQUS. Calibration of the numerical results was performed by comparing the experimental and numerical lateral load deflection response profiles for the ½ scale model ROPS. Figure 4-18 shows that very close agreement between the two results was obtained under the lateral loading phase. As shown by the distinctive shape of this figure, the FE model ROPS has exhibited a stable response with a constant load carrying
capacity of approximately 190 kN. The shape of this load deflection profile is characteristic of a structure that has absorbed energy in a smooth, efficient and ductile manner.

Figure 4-18 Comparison of FEA and Experimental Lateral Load Deflection Profiles

The severity of the plastic deformation experienced by the ½ scale model ROPS is further exemplified through reference to Figure 4-19. Examination of this figure which shows the Von Mises stress distribution throughout the ROPS, indicates that plastic hinges had formed at the top and base of each post of the ROPS, which was an identical replication of the findings presented in chapter 3 for the corresponding experimental model. In addition to this, the residual Von Mises stress distribution throughout the model after removal of the lateral load has also been shown in Figure 4-20.
The plastic hinge zones that were located at the top and base of each post underwent significant deformation during the lateral loading phase. This deformation involved substantial distortion of the finite element mesh and was characterised by the inward...
folding of the elements that were located on the extreme compression face of the member. The distortion of the mesh at two of these hinges locations, has been shown in Figure 4-21.

Figure 4-21 Mesh Distortion in Plastic Hinge Region

4.5.2 Vertical Loading Phase

Vertical loading of the K275 ROPS model resulted in an under-prediction of the vertical load deflection response obtained during the experimental analysis. Figure 4-22 shows the resulting comparison of these two results plotted with reference to the structure’s pre-deformed position after removal of the lateral load. The response shown by the FE model ROPS to the applied vertical load, appeared to be fairly linear with no residual vertical deflection taking place after removal of the load. This response was significantly stiffer than the one that was displayed by the corresponding experimental model during testing. The stiffness variation between the two models can easily be distinguished from this figure by examining the gradients of each curve. It is evident therefore that the material model may not be able to adequately simulate the reduced stiffness that is inherent within the ROPS as a result of the sustained plastic deformation that takes place after the initial lateral loading phase.
Figure 4-22 Comparison of FEA and Experimental Vertical Load Deflection Profiles

The Von Mises stress distribution throughout the model ROPS during the vertical loading and unloading stages is presented in Figure 4-23 and Figure 4-24 respectively. Examination of Figure 4-23, indicates that high stresses have occurred locally in the beam zone, however, these stresses are still within the elastic limitations of the material. In addition to this, some localised yielding has taken place in the corner region at the connection zone between the beam and the post. This localised yielding appears to have had only a minor influence on the deformation that has been experienced by the FE ROPS model in the vertical direction.

Figure 4-23 Von Mises Stress Distribution under Vertical Load
4.5.3 The longitudinal Loading Phase

The load deflection profile for the ROPS under the applied longitudinal load did not compare well with the results obtained from the corresponding experimental loading phase. The reasons for these differences are attributed to similar reasons that have been discussed previously for the vertical loading phase. Figure 4-25 and Figure 4-26 show the Von Mises stress distribution throughout the ROPS during the longitudinal loading and unloading stages respectively. Careful examination of Figure 4-25 shows the presence of distinct yielding at the base of the front and back faces of each post at the point of maximum longitudinal load. This result was expected, as the ROPS is behaving as a two post cantilever about this direction which would therefore give rise to high stresses at the base of each post.

Figure 4-24 Residual Von Mises Stress after Vertical Load Removal
4.5.4 Similitude Verification

The scaling laws and relationships determined between pertinent variables that were established in chapter 3, were verified using FEA for the full scale prototype ROPS and the ½ scale model ROPS for the K275 bulldozer. The response behaviour
exhibited by each model when subjected to the loading requirements of the standard, are outlined in accordance with Figure 4-27, Figure 4-28 and Figure 4-29.

Figure 4-27 Lateral Load Deflection Response Comparison between Full Scale and Half Scale K275 ROPS

Figure 4-28 Vertical Load Deflection Response Comparison between Full Scale and Half Scale K275 ROPS
Examination of these figures in association with reference to Table 4-3, indicates that the response behaviour of a full scale ROPS can be accurately predicted by using the principles of similitude modelling and implementing an appropriate scaling law.

Table 4-3 Model and Prototype Results Comparison

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Full scale K275 prototype</th>
<th>Half scale K275 model</th>
<th>Model / Prototype FEA</th>
<th>Model / Prototype Similitude</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lateral Load</td>
<td>752 kN</td>
<td>188 kN</td>
<td>1/4</td>
<td>1/4</td>
</tr>
<tr>
<td>Lateral Deflection</td>
<td>140 mm</td>
<td>70 mm</td>
<td>1/2</td>
<td>1/2</td>
</tr>
<tr>
<td>Energy Absorption</td>
<td>97188 J</td>
<td>12148 J</td>
<td>1/8</td>
<td>1/8</td>
</tr>
<tr>
<td>Vertical Load</td>
<td>978 kN</td>
<td>244 kN</td>
<td>1/4</td>
<td>1/4</td>
</tr>
<tr>
<td>Vertical Deflection</td>
<td>5.1 mm</td>
<td>2.55 mm</td>
<td>1/2</td>
<td>1/2</td>
</tr>
<tr>
<td>Longitudinal Load</td>
<td>385 kN</td>
<td>96 kN</td>
<td>1/4</td>
<td>1/4</td>
</tr>
<tr>
<td>Longitudinal Deflection</td>
<td>14.6 mm</td>
<td>7.3 mm</td>
<td>1/2</td>
<td>1/2</td>
</tr>
</tbody>
</table>

### 4.5.5 Influence Corner Radii in RHS/SHS Members

The global influence of including the corner radii of the RHS/SHS members was examined under application of the static lateral loading phase only. Reference to Figure 4-30 shows the resulting load deflection profile that was obtained for the half scale model with the inclusion and omission of the corner radii. Examination of this graph indicates minimal difference between the two results, with a closer inspection revealing that exclusion of the corner radii resulted in an overestimation in the peak load and energy absorption capacity of the ROPS of approximately 1.5% and 3% respectively. These differences were considered to be minimal and it was therefore concluded that for future modelling considerations, the corner radii could be omitted in order to reduce time in the model development phase.

![Figure 4-30 Lateral Load Deflection Response Showing Influence of Corner Radii](image)

Figure 4-30 Lateral Load Deflection Response Showing Influence of Corner Radii
4.6 FEA of a Wa600 Wheeled Loader ROPS

The final ROPS that was modelled using FEA was a full scale two post cantilever type ROPS that was suitable for attachment to a 50t wheeled loader. A similar modelling procedure was employed to study the behaviour of the ROPS under the loading conditions established in the Australian Standard. The following section provides a description of the modelling process and the results that were obtained from the analysis.

4.6.1 Model Setup

The geometry of the full scale FE ROPS was again based on site measurements taken from a Wa600 wheeled loader that was located at the manufacturer’s storage yard. Appropriate member sizes were assigned to the ROPS so that it would pass the requirements of the standard. MSC Patran was similarly employed for constructing the ROPS geometry. Figure 4-31 shows an illustration of the Wa600 wheeled loader that is fully equipped with a manufacturer supplied ROPS. Similar to the K275 ROPS, the detachable roofing FOPS section which provides protection to the operator under falling objects was omitted from the FE model. Details of the ROPS geometry that was used to construct the FE model has been presented in Figure 4-32.
4.6.2 Elements

The element definition was similar to that used previously and involved the use of the ABAQUS shell element type S4R. The shell thickness that was chosen for use throughout the model was 10mm for the posts and 12mm for the beams. Meshing of the FE model was performed using the isomesh capability within Patran with a discretisation density of 10mm selected. Figure 4-33 shows an illustration of the mesh distribution throughout the model.
4.6.3 Material Properties

The material model that was used for this ROPS was identical to the one that was chosen previously for the 630E and K275 ROPS models. Whilst no experimental testing was conducted on this particular ROPS, the section sizes used for the posts and beams of the ROPS were identical to those chosen for the K275 ROPS model.

4.6.4 Section Properties

Table 4-4 outlines the section properties that were used for the FE model and ROPS during the analysis.

<table>
<thead>
<tr>
<th>ROPS Member</th>
<th>Member Type</th>
</tr>
</thead>
<tbody>
<tr>
<td>Posts</td>
<td>350 grade 250x150x10 RHS</td>
</tr>
</tbody>
</table>
Front Cross Beam | 350 grade 250x150x10 RHS
Rear Cross Beam | 350 grade 250x250x12 SHS
Side Beam | 350 grade 250x150x10 RHS
Stiffener | 350 grade 12mm plate

4.6.5 Loading Procedure

The loading sequence used for the FE modelling was identical to the procedure that was employed for the previous K275 ROPS. Details of the load magnitudes that were applied to the FE ROPS model and the energy absorption requirement are presented in Table 4-5.

Table 4-5 AS2294.2 Load and Energy Requirements

<table>
<thead>
<tr>
<th>Requirement</th>
<th>AS2294.2 - 1997 Formula</th>
<th>Result</th>
</tr>
</thead>
<tbody>
<tr>
<td>$F_{Lateral-Min}$</td>
<td>$F_{Lati} \cdot \frac{M}{10000^{0.5}}$</td>
<td>408 kN</td>
</tr>
<tr>
<td>$U_{Lateral}$</td>
<td>$U_{Lat} = \frac{M}{10000^{0.25}}$</td>
<td>92060 J</td>
</tr>
<tr>
<td>$F_{Vertical}$</td>
<td>$F_{Vert} = 19.61M$</td>
<td>968 kN</td>
</tr>
<tr>
<td>$F_{Longitudinal}$</td>
<td>$F_{Long} = \frac{M}{10000^{0.5}}$</td>
<td>326 kN</td>
</tr>
</tbody>
</table>

4.7 Results from Finite Element Analysis

4.7.1 Lateral Loading Phase

The lateral load deflection response of the FE model ROPS which is shown in Figure 4-34 was very similar in shape to the response displayed by the previous K275 model. As shown by this graph, the model’s peak load carrying capacity was approximately 600 kN which occurred at a deflection level of 40mm. Through examining this graph, it is evident that the peak load carrying capacity of the ROPS was maintained over a considerable deflection range which indicates that the ROPS had exhibited a very ductile response to the applied loads which is a positive characteristic of an efficient energy absorbing structure. UMTRI (1998)
Figure 4-34 Lateral Load Deflection Response of Wa600 ROPS

The yielding experienced by the model during this loading phase has been shown clearly in Figure 4-35 which highlights the Von Mises stress distribution throughout the structure under full lateral load. The Orange and Red stress contours which occur at the top and base of each post, highlight the yield zones where the plastic hinges have formed in the structure. This yielding pattern is similar to the one displayed by the previous K275 ROPS model and is characteristic of the collapse mode exhibited by a fixed base framed structure under an applied sideways load. In addition to these yield patterns, the Von Mises stress distribution throughout the model after removal of the lateral load has been shown in Figure 4-36. As found from previous analyses, high residual stresses remain in the hinge locations where significant plastic deformation has taken place, whereas the remainder of the model shows the presence of only very minor stress residuals.
Figure 4-35 Von Mises Stress under Lateral Load

Figure 4-36 Residual Von Mises Stress after Lateral Load Removal
4.7.2 Vertical Loading Phase

The vertical loading phase resulted in the model displaying a very stiff, almost linear load deflection response. Figure 4-37 shows the shape of the load deflection response profile for the model under the applied vertical load.

![Figure 4-37 Vertical Load Deflection Response of Wa600 ROPS](image)

The Von Mises stress distribution throughout the model under the applied maximum vertical load has been plotted in Figure 4-38. The maximum stresses in this figure occur predominantly at the connection zone between the beams and posts of the ROPS. After removal of the vertical load, the form of the Von Mises stress residuals throughout the model, was found to be very similar to those that occurred for the previous unloading stage in the lateral direction. This finding further emphasises the elastic behaviour that was shown by the ROPS during this loading phase. Details of this residual stress distribution have been shown in Figure 4-39.
Figure 4-38 Von Mises Stress under Vertical Load

Figure 4-39 Residual Von Mises Stress after Vertical Load Removal
4.7.3 The Longitudinal Loading Phase

The longitudinal load deflection profile which is shown in Figure 4-40 was plotted with reference to the structure’s predeformed position. The shape of this graph shows that the model exhibited a stiff response to the applied loads with only a small amount of residual deflection taking place after removal of the load.

![Figure 4-40 Longitudinal Load Deflection Response of Wa600 ROPS](image)

Figure 4-41 and Figure 4-42 show the Von Mises stress distribution throughout the ROPS during the longitudinal loading and unloading stages respectively. It is evident from examination of Figure 4-41 that yielding has taken place at the base of each post of the ROPS. The extent of this yielding, however, has had a very minor influence on the residual stress distribution throughout the ROPS after removal of the load which can be seen through reference to Figure 4-42 which shows that the stress contours are very similar to those shown in Figure 4-36 and Figure 4-39.
Figure 4-41 Von Mises Stress under Longitudinal Load

Figure 4-42 Residual Von Mises Stress after Longitudinal Load Removal
4.8 Numerical Investigation of Post Yield Behaviour of ROPS

At present current ROPS performance standards such as AS2294-1997 give little assistance to designers on how to adequately size a ROPS to meet their loading and energy requirements. Commonly, designers will specify overly stiff member sizes to ensure that the prototype ROPS will pass the requirements without premature failure. Once a successful design has been developed, it may be reproduced without further test and put into production. Based on this common day approach, it is clear that there is a lack of knowledge that exists on the fundamental principles required for successful ROPS design and it is therefore essential that design guidelines be established to ensure that an optimum level of safety is provided to heavy vehicle occupants during rollover accidents.

The adequacy of the requirements presented in current ROPS regulatory standards has been questioned by authors such as Steinbruegge (1975), Woodward and Swan (1980) and more recently Ho (1994). Ho used a simplified collapse load approach to study the energy absorption capability of a fixed base two post ROPS prior to encroachment of the DLV. Ho examined a number of two post ROPS configurations with collapse loads equal to 100%, 124%, 150% and 200% of the minimum lateral load provision of SAE J1040. Ho concluded that a ROPS which had a collapse load greater than 150% of the minimum lateral load requirement would be able to absorb the necessary amount of energy prior to encroachment of the DLV. He suggested that structures that were proportioned with collapse loads below this value would prematurely fail the performance criteria of SAE J1040. The direction in which Ho concentrated his research has been extended further by using finite element analysis to verify the adequacy of two independent ROPS frames. The scope of the numerical investigation has involved varying the sectional geometry of each frame to achieve ROPS configurations that possess a wide range of collapse loads. The investigation has been comprised of two stages which has involved assessing the capacity of each ROPS to adequately fulfil the requirements of the standard and to secondly assess the energy absorption capacity of each ROPS prior to DLV infringement. It has been envisaged that the implementation of such a study will provide meaningful results that may assist with the development of ROPS design guidelines. The ROPS frames that were chosen for the study were those for the K275 bulldozer and Wa600
wheeled loader. Variation in the stiffness of each ROPS frame was achieved by adjusting the sectional geometry of the posts. For all analyses the geometry of each ROPS model was proportioned carefully to ensure that the formation of plastic hinges required for energy dissipation took place at the top and base of each post during application of the lateral load. Control of this deformation process was achieved by maintaining a constant beam section size that possessed a much higher plastic moment capacity than the posts of the ROPS. As mentioned previously, the sectional geometry of the post was varied to develop a number of different ROPS configurations that possessed a wide range of collapse loads. To achieve the variation in post stiffness, the width and thickness of the post was kept constant, while the depth of the section was varied. This process was performed to limit the number of variable and minimise the number of numerical simulations.

In the following sections of this chapter, the reader is presented with a brief introduction into plastic bending theory and how this may be applied to the proportioning of ROPS frames to adequately fulfil the requirements of current regulatory standards. In addition to this, the results that were obtained from the numerical investigation are presented.

4.8.1 Plastic Analysis of ROPS

Conventional engineering design is based on an elastic approach whereby the maximum stress occurring in a structure is limited to the allowable working stress of the material being used. Implementation of an approach such as this, results in a structure that remains un-deformed after removal of the applied load. The principles involved in the design of safety frames such as a ROPS differ substantially from those that are employed during conventional engineering design. Rollover Protective Structures rely on the absorption of rollover impact energy through the permanent plastic deformation of their structural members. The following section provides a brief overview of the plastic design principles that may be used to develop appropriate member sizes for ROPS.

4.8.2 Plastic Bending Theory

An arbitrary beam section subjected to an applied bending moment will yield at the section’s extreme fibres as the bending moment is gradually increased. Once this has
taken place, the strain at the outermost fibres will progressively increase without any further increase in stress. At this stage of loading, yielding will begin to spread towards the axis of zero strain until the entire cross-section has become fully plastic and a plastic hinge has deemed to have formed. A structure will remain stable until a sufficient number of plastic hinges have formed and it can no longer carry any further load. At this stage of loading a collapse mechanism is deemed to have occurred and the structure no longer remains stable. Figure 4-43 displays graphically the stress distribution that takes place within the cross section of a member during progressive bending.

Figure 4-43 Plastic Bending of a Beam

This theory may be easily applied to a framed structure such as a ROPS in order to determine the maximum lateral load (collapse load) that the structure may withstand. A simple example is provided for the case of a fixed base two post ROPS as shown
in accordance with Figure 4-44. For this example the plastic moment capacity of each post of the ROPS is represented by the term \( MP_1 \), whereas \( MP_2 \) represents the plastic moment capacity of the beam linking the two posts. It is well understood from conventional plastic theory that a simple ROPS frame will develop plastic hinges at locations A, B, C and D when subjected to an increasing horizontal force. Provided that the plastic moment capacity \( MP_2 \) of the beam is larger than that of the posts, the plastic hinges at locations B and C will form in the posts rather than the beam. By using the principle of virtual work and equating the work done by the load and by the plastic hinges after a rotation \( \theta \), the collapse load for the ROPS \( (F_c) \) may be determined by the following equation:

\[
F_c = \frac{4MP_1}{L_e} \text{ where } MP = Z_p \gamma \quad \text{(Equation 4.3)}
\]

In the above equations \( L_e \) represents the clear height from the base of the post to the underside of the beam, \( MP \) represents the plastic moment capacity of the post about its local Y-Y axis, \( Z_p \) is the plastic section modulus of the ROPS post and \( \gamma \) is the yield stress of the material. Figure 4-45 shows the section orientation of the post that needs to be used for the calculation of \( MP \) in equation 4.3.
4.8.3 Energy Absorption Enhancement

The energy absorbed by a ROPS is characterised by the area under the load deflection response profile. The typical response behaviour exhibited by a structure such as a ROPS during loading, involves a sharp rise in the load deflection response as the structure maintains its initial shape and the material responds in the elastic region. During this mode of behaviour, the structure reaches its maximum peak load with very little deflection. With continued load application, the structure will begin to plastically deform, which is characterised by a significant increase in deflection. As the structure continues to deflect, its load carrying capacity will begin to decline appreciably until complete collapse has taken place. The characteristic that results in the achievement of an optimum level of energy absorption is the preservation of a constant load response. If a constant force can be maintained, the area beneath the load deflection profile will approximate a rectangle which is the optimum shape for maximum area and hence energy absorption. If a constant force cannot be achieved, the additional energy must be provided by means of a larger initial peak force or a larger deflection.

4.8.4 Numerical Investigation for K275 ROPS

The full scale FE model of the K275 ROPS that was studied earlier in this chapter was subjected to a multi-staged numerical investigation. As discussed previously, stage one of the analysis involved adjusting the section geometry of the posts of the ROPS and studying the influence that this had on the structure’s ability to absorb
FE models of the K275 ROPS were developed using the same procedures that were employed previously. In addition to varying the post section size of the ROPS, the wall thickness of the beam linking each post was also increased to 12mm for all models to ensure that the plastic hinges developed within the posts of the ROPS. Each analysis was run using post section sizes of 120x250x10, 150x250x10, 200x250x10 and 240x250x10. These section geometries corresponded to ROPS collapse loads of 100%, 140%, 200% and 260% of the minimum lateral loading provision of the standard respectively. Table 4-6 provides a summary of the post section sizes that were used for each analysis. In addition to this, the plastic moment capacity MP and lateral collapse load $F_c$ which were determined from Equation 4.3 have also been shown in this figure. It must be noted that the parameters MP and $F_c$ were determined using simplified expressions that do not account for the influence of any secondary effects or material strain hardening. The FE analyses in general predicted slightly higher collapse loads and corresponding plastic moment capacities than those that were determined using equation 4.3 which refers to a simple analysis method. A comparison between the two results has also been shown in Table 4-6.

At present ROPS designers are given guidance on how to adequately proportion a ROPS that will adequately fulfil the requirements of the standard. The aim of this study is to rectify this situation and provide design guidelines that will assist in the adequate design of ROPS. As mentioned above, a number of ROPS post configurations have been studied to evaluate the lateral loading and energy absorption provisions of the standard. The main focus of ROPS standards such as AS2294.2-1997 is on the lateral load and energy absorption provisions. The other two loading sequences are strength cases to ensure that a ROPS has adequate strength impacts that are not directed about the lateral axis of the vehicle. It is well understood that rollovers generally occur about the sideward direction of the vehicle and that through absorbing some of the kinetic energy of the roll, the vehicle may be brought to rest. For this study, ROPS posts were proportioned to develop plastic hinges about their weak axis normal to the applied lateral load, however, in addition
to this they were also proportioned to have sufficient capacity about the other axes to
withstand the subsequent vertical and longitudinal loading stages.

Table 4-6 Lateral Collapse load and Plastic Moment Capacity of FE Models

<table>
<thead>
<tr>
<th>ROPS Post Size (mm)</th>
<th>MP_{(Empirical)} (kNm)</th>
<th>F_{C(Empirical)} (kN)</th>
<th>MP_{(FEA)} (kNm)</th>
<th>F_{C(FEA)} (kN)</th>
<th>MP_{(FEA)} / MP_{(Empirical)}</th>
</tr>
</thead>
<tbody>
<tr>
<td>120x250x10</td>
<td>138</td>
<td>498</td>
<td>161</td>
<td>582</td>
<td>1.17</td>
</tr>
<tr>
<td>150x250x10</td>
<td>192</td>
<td>689</td>
<td>218</td>
<td>785</td>
<td>1.14</td>
</tr>
<tr>
<td>200x250x10</td>
<td>280</td>
<td>1001</td>
<td>312</td>
<td>1126</td>
<td>1.12</td>
</tr>
<tr>
<td>240x250x10</td>
<td>360</td>
<td>1295</td>
<td>392</td>
<td>1412</td>
<td>1.09</td>
</tr>
</tbody>
</table>

In developing the range of post section sizes for the numerical investigation, it was
necessary to carefully proportion the members so that they would exhibit ideal
structural performance under each of the applied loading stages from the ROPS
performance standard AS2294.2-1997. The first loading stage, which is directed
about the lateral direction of the ROPS, is an energy absorbing load case that is used
to bring the vehicle to rest during the occurrence of an overturn. Due to the nature of
the lateral loading sequence, the post section geometry was therefore orientated with
its weak axis positioned normal to the direction of the structure’s lateral axis.
Implementation of this design condition therefore forced the structure to develop
plastic hinges in the posts during loading. The vertical loading sequence which
occurs directly after removal of the lateral load, is used to ensure that the ROPS can
support the mass of the vehicle if it comes to rest in an upturned positioned. This is a
strength load case which means that the structural members that resist this load must
possess adequate capacity to avoid premature failure. For the K275 ROPS, the
vertical loading is applied to the beam linking the two posts of the ROPS. For a
ROPS that has already developed plastic hinges at the post beam junction during the
lateral loading phase, it is necessary for only one further hinge to form at the
midspan of the beam before a beam collapse mechanism will be formed. Using this
approximation, the vertical collapse load for the ROPS (F_{VC}) may be estimated by
using Equation (4.4).

\[ F_{VC} = \frac{4MP}{Le} \]  

(Equation 4.4)
where $L_e$ represents the clear width of the beam between supporting posts and $MP$ the plastic moment capacity of the beam about its local X-X axis.

Similar to the vertical loading stage, the longitudinal loading sequence is also a strength load case that ensures that the ROPS has sufficient strength in the direction of its longitudinal axis to withstand non symmetrical rollovers. The collapse load capacity of a ROPS in this direction is difficult to estimate at this stage, if plastic hinges have already developed in the posts of the ROPS. From experimental testing it has been found that there is still sufficient residual capacity left in the longitudinal direction of the members to withstand the application of the longitudinal load. With these views in mind, an estimate of the maximum collapse load ($F_{LC}$) of the structure may be established using Equation (4.5). This equation assumes that the section can develop the full plastic moment capacity of each post about an axis that is normal to the longitudinal direction of the ROPS posts. For this reason it is necessary that the post members be orientated so that they possess sufficient depth parallel to this direction so that they will be able to withstand the applied longitudinal load.

$$F_{LC} \leq \frac{2MP}{L_e}$$

(Equation 4.5)

Where $L_e$ represents the clear width of the beam between supporting posts and $MP$ represents the plastic moment capacity of the ROPS posts about their local X-X axis. Details of the plastic moment capacities and collapse loads that were calculated using Equation (4.4) and Equation (4.5) have been summarised in Table 4-7 and

Table 4-8.

<table>
<thead>
<tr>
<th>ROPS Post Size (mm)</th>
<th>MP Beam (kNm)</th>
<th>$L_e$ Beam</th>
<th>$F_{CV}$</th>
<th>$F_{Vert}$ (kN)AS2294</th>
<th>$F_{Vert}/F_{CV}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>120x250x10</td>
<td>449</td>
<td>1.57</td>
<td>1144</td>
<td>908</td>
<td>1.26</td>
</tr>
<tr>
<td>150x250x10</td>
<td>449</td>
<td>1.54</td>
<td>1166</td>
<td>908</td>
<td>1.28</td>
</tr>
<tr>
<td>200x250x10</td>
<td>449</td>
<td>1.49</td>
<td>1205</td>
<td>908</td>
<td>1.33</td>
</tr>
<tr>
<td>240x250x10</td>
<td>449</td>
<td>1.45</td>
<td>1238</td>
<td>908</td>
<td>1.36</td>
</tr>
</tbody>
</table>
Through studying the response behaviour of each ROPS configuration with the stiffness distributions as shown, it was envisaged that the adequacy of the code provisions particularly with reference to the minimum lateral load and energy absorption requirement could be established. Staab (1971) suggested that the minimum lateral loading criteria had been introduced into the standards to ensure that the ROPS/Machine assembly would be capable of imparting energy into the ground. He stated that the force requirement was an exponential function of machine weight and was directly related to the energy absorbed by the ROPS. Staab also suggested that the inclusion of the minimum lateral loading provision in the standard was designed to move a ROPS into a more desirable force deflection range. He suggested that a balance is achieved when the ROPS has acceptable imparting (force) and energy absorbing (deflection) characteristics.

### 4.9 Stage 1 of Numerical Study

Each model with the post section geometries outlined in Table 4-6 was subjected to a finite element analysis using the FE code ABAQUS. The analysis procedure for each ROPS utilised the displacement controlled solution method and involved loading of the ROPS up to the point of DLV encroachment. The lateral load deflection response up to this deflection limitation was recorded for each analysis and has been presented in Figure 4-46. To assist the reader in understanding this figure, the plastic moment capacity (MP) of the ROPS posts has been introduced to categorise the stiffness of each model. This term has been discussed in detail in the preceding sections and
refers to the plastic moment capacity of each post about its local Y-Y axis in accordance with Figure 4-45.

The load deflection response of each model shown in Figure 4-46 indicates an initial stiff elastic response. After the peak load was reached, the stiffness of each model was reduced substantially and the load carrying capacity began to fall. This behaviour was characterised by significant yielding and the formation of plastic hinges at the ends of each post which gave rise to a significant increase in the lateral deflection experienced by each model. Each analysis was continued until the zone of the DLV was impeded. As expected from the relationships derived in Equation 4.3, the load deflection response for each model showed that the load carrying capacity of a ROPS was directly proportional to the plastic moment capacity of its posts.

Figure 4-46 Lateral Load Deflection Response

Figure 4-47 shows the variation in energy absorbed by each model with increasing lateral deflection. The results shown by this graph were as expected and clearly indicate that the energy absorbed by each model increases with increasing lateral deflection and ROPS post stiffness. Further to this, the direct relationship between plastic moment capacity of the ROPS posts and the amount of energy absorbed, has been further clarified in Figure 4-48, by plotting the relationship between each of these variables. Clearly from this figure, the energy absorbed by each model is an
increasing function that is directly proportional to the plastic moment capacity of the ROPS posts.

![Figure 4-47 Lateral Energy Absorption versus Lateral Deflection](image1)

![Figure 4-48 Maximum Lateral Energy Absorption versus ROPS Post Plastic Moment Capacity](image2)

4.10 Stage 2 of Numerical Study

The lateral load deflection response which satisfied the requirements of AS2294.2-1997, was plotted in accordance with Figure 4-49 for each ROPS model.
Examination of this graph, showed that each FE model satisfied the minimum loading and energy absorption requirements of the standard without violation of the 500mm deflection limitation of the DLV. The graphs indicate that the criteria established in the standard may be satisfied by placing either a high force/low deflection or a low force/high deflection demand upon the ROPS.

![Lateral Load Deflection Response for Varying MP](image)

**Figure 4-49 Lateral Load Deflection Response for Varying MP**

### 4.10.1 Vertical load response

Each model was also subjected to the vertical loading requirement of the standard. The stiffness of each model about this direction was high which resulted in a stiff elastic response which has been summarised in Figure 4-50. As expected, the FE models that were more flexible underwent further deflection during loading.
4.10.2 Longitudinal loading

The longitudinal loading phase gave rise to response profiles that have been displayed in Figure 4-51. Each model showed some nonlinear behaviour during this loading sequence which has been exemplified by the residual deflection present after removal of the load. The trends displayed by this graph were expected and indicate that the lower stiffness ROPS displayed more deflection during loading.
4.11 Numerical Study for Wa600 ROPS

A similar numerical study was also performed on the Wa600 ROPS. The primary variable that was closely examined during this study was the stiffness influence of the ROPS posts. This variable, which was quantified using the plastic moment section capacity of the posts, was adjusted to give rise to three ROPS configurations that possessed collapse loads that were equivalent to 100%, 130% and 190% of the minimum lateral load requirement of the standard. A two stage process was employed whereby the response of each configuration to the load and energy requirements of the standard was studied in addition to the ability of each ROPS to absorb maximum energy prior to infringement of the DLV. Table 4-9 provides a summary of the post section sizes that were used during this study and their corresponding plastic moment capacity $MP$ and lateral collapse load $F_c$ that were determined using Equation 4.3 and from finite element analysis. Similar to the previous study for the K275 ROPS, member sizes were arranged so that they possessed sufficient capacity to be able to withstand the applied vertical and longitudinal loads after incurring plastic deformation from the lateral loading sequence.

<table>
<thead>
<tr>
<th>ROPS Post Size (mm)</th>
<th>$MP_{(Empirical)}$ (kNm)</th>
<th>$F_c_{(Empirical)}$ (kN)</th>
<th>$MP_{(FEA)}$ (kNm)</th>
<th>$F_c_{(FEA)}$ (kN)</th>
<th>$MP_{(FEA)} / MP_{(Empirical)}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>120x250x10</td>
<td>151</td>
<td>415</td>
<td>171</td>
<td>470</td>
<td>1.13</td>
</tr>
<tr>
<td>150x250x10</td>
<td>192</td>
<td>528</td>
<td>214</td>
<td>588</td>
<td>1.11</td>
</tr>
<tr>
<td>200x250x10</td>
<td>280</td>
<td>770</td>
<td>305</td>
<td>838</td>
<td>1.09</td>
</tr>
</tbody>
</table>

4.12 Stage 1 of Numerical Study

The established FE models were subjected to an enforced displacement of 500mm that was applied laterally to the side face of each model. This displacement corresponded to the maximum deflection that the ROPS could be subjected to prior to violation of the DLV zone. The resulting reaction forces for each model were measured at the base of each post and were used to assemble the load deflection response profiles that have been shown in Figure 4-52. Examination of this figure indicates a sharp rise in the load deflection response behaviour of each model up to...
the point of maximum load. This region is characterised by minimal deflection as each model behaves elastically. After this point has been reached, the load deflection response is characterised by a progressive decline until the maximum deflection limit of 500mm was reached.

Reference to Figure 4-53 shows the energy absorbed by each ROPS up to this deflection limiting control. As envisaged, each response profile shows a smooth increase in the level of energy absorbed by each model as the lateral load is increased. It is evident from examination of this figure that for a given deflection, the ROPS that possesses the largest plastic moment capacity will also be able to absorb the greatest amount of energy. This behaviour was expected as the collapse load of a ROPS is directly proportional to the plastic moment capacity of its posts which therefore results in a larger area under the load deflection curve and hence a greater amount of energy absorption. This relationship is more clearly defined through reference to Figure 4-54. This particular figure shows the relationship between maximum lateral energy absorption and ROPS post plastic moment capacity MP, where MP has been determined using both equation 4.3 and from finite element analysis. As found from the previous study, it is evident from this figure that for a carefully proportioned ROPS, the amount of energy absorbed is directly proportional to the plastic moment capacity of its posts.

Figure 4-52 Lateral Load Deflection Response of Wa600 FE Models
4.13 Stage 2 of Numerical Study based on ROPS Post Stiffness

Stage two of the numerical study involved verifying the adequacy of the AS2294.2 provisions. The procedure that was employed for this study involved subjecting each ROPS assembly to the loading and energy requirements of the standard and examining the corresponding load deflection response. Through verifying the
standard provisions it was envisaged that further guidance could be provided to assist designers and manufacturers of rollover protective devices. Figure 4-55 shows the lateral load deflection behaviour that has been displayed by each model to satisfy the requirements of the standard. As mentioned previously for the K275 study, these results indicate that varying force deflections demands may be used to adequately meet the performance criteria of the standard. Clearly from this figure a stiffer ROPS will satisfy the requirements of the standard by displaying a high force/low deflection demand as compared to a more flexible ROPS that will display a low force/ high deflection demand.

Figure 4-55 Lateral Load Deflection Response for varying MP

The vertical loading sequence resulted in each model undergoing minimal deformation. The load deflection response profile for this loading sequence has been plotted for each model in Figure 4-56. As shown by this figure, a stiff elastic response has been used to resist the applied vertical load.
The longitudinal loading sequence showed some nonlinear behaviour with some residual deflection occurring after removal of the load. Figure 4-57 shows the resulting load deflection behaviour for each model during this loading phase.

Figure 4-57 Longitudinal Load Deflection Response for varying MP

4.14 Chapter Summary

Within the scope of this chapter nonlinear finite element analysis techniques have been developed and used to assess the performance of rollover protective structures
to the loading and energy criteria established in AS2294.2-1997. The developed modelling techniques have been calibrated against experimental testing results and have been found to be particularly accurate for the lateral loading stage of the code. Whilst the developed procedure can simulate the vertical and longitudinal loading phases, the correlation with experimental results was found to be unsatisfactory. Reasons for this have been attributed to the inability of current FE material models to handle the influence of load reversal effects such as the Baushinger effect as well as the possible inherent residual stresses from the heavy welding process during fabrication. Despite these problems it must be noted that in general the lateral loading sequence is considered to be the most important loading event of which the correlation between FEA and experimental testing was excellent. The calibrated FE models have been used to perform a series of dynamic impact studies that have been described in detail in chapter 5.

During this chapter, a numerical study was also performed which examined the influence of ROPS post stiffness on the response behaviour of two post rollover protective structures under static loading. For this study, each ROPS was carefully proportioned to force the plastic hinges to develop in the posts of the structure during application of the lateral load. Post section geometries were carefully arranged to ensure that each ROPS configuration possessed sufficient capacity to be able to withstand the further vertical and longitudinal loading phases of the standard after removal of the lateral load. The plastic moment capacity of the ROPS posts MP was introduced to assess the influence of ROPS post stiffness on the response behaviour of the structure.

A series of post section sizes were chosen for analysis with the aim of assessing the adequacy of the provisions of the ROPS performance standards. A simple collapse load approached was used to estimate the capacity of different ROPS frames and develop a series of ROPS that possessed collapse loads that varied in the range of 100 to 240% of the minimum lateral loading provision of the standard. For the two different ROPS configuration that were addressed namely the K275 bulldozer ROPS and the Wa600 wheeled loader ROPS, it was discovered that a carefully proportioned two post ROPS could adequately meet the requirements of the standard provided that it possessed a collapse load that was equivalent to that of the minimum lateral
loading provision of the standard. It was also found that the energy absorption
capability of a ROPS was an increasing function of the plastic moment capacity of its
posts. In general for the stiffness configurations that were addressed, it was found
that the response of a ROPS to the applied vertical and longitudinal loading was
approximately linear. Based on this study, it is recommended that two post ROPS be
carefully proportioned to form plastic hinges at the top and base of each post for the
lateral loading sequence as well as possessing adequate capacity to withstand the
vertical and longitudinal loading phases. In general for two post ROPS, the
simplified collapse load procedure that has been presented in Equation 4.3 may be
used to guide the designers to develop ROPS configuration that possess lateral
collapse loads that are at least equivalent to the minimum provision of the standard.
Chapter 5 Dynamic Impact Analysis

This chapter is concerned with the dynamic response of rollover protective structures subjected to impact loads that are characteristic of those that are experienced during the sideward rollover of a vehicle on a firm slope. The FE ROPS models for the K275 Bulldozer and Wa600 wheeled loader that were studied analytically in chapter 4 have been selected to carry out the necessary analyses. A simplified method based on a conservation of angular momentum approach reported by (Watson.1967) has been used to estimate the likely proportion of energy that would be absorbed by the ROPS of each vehicle during a sideward overturn. The explicit FE code LS-Dyna v970 has been used to conduct the necessary dynamic impact modelling for rollover impacts on firm slopes with inclinations of 15°, 30° and 45°. The influence that controlling variables such as ROPS stiffness, vehicle mass and impact velocity have on the dynamic response of each ROPS has been carefully studied and reported accordingly. The results from this study have been compared with those obtained from the static analysis performed in chapter 4, to establish the effect of possible dynamic amplifications and the adequacy of current standard provisions.

5.1 Rollover Simulation using FEA

Rollover simulation using finite element analysis has received little attention from researchers who have been concerned with vehicle rollover and the enhancement of occupant protection during such events. Chou et al. (1998), highlighted that the major difficulty associated with using FE for rollover analysis was the large simulation time required to capture the event accurately. In direct parallel to this, Klose (1971) also emphasised that the rollover process was extremely difficult to model as it involved the complex interaction of numerous parameters that influenced the behaviour of the rolling vehicle. In the open literature, the FE modelling of rollover protective structures under dynamic loading, has been limited to research performed by Tomas et al. (1997) and Harris et al. (2000). The work performed by Harris (2000) examined the rearward rollover of a tractor whereas Tomas’s research used the program MADYMO to study the effect of ROPS stiffness and occupant restraint during sideward rollover of an earthmoving machine equipped with a
ROPS. Whilst the modelling techniques employed by each of these authors has assisted with assessing the performance of a ROPS under simulated dynamic impact loads, little comparison has been made with reference to the adequacy of the static loading procedures adopted in current ROPS standards and the possible dynamic amplifications that may take place during such loading conditions. With these views in mind the simplified procedure proposed by Watson (1967) has been used as a basis for a dynamic impact study that has investigated the influence of critical parameters that control the response behaviour of a ROPS subjected to such loading conditions.

5.2 Development of Dynamic Impact Loads for K275 Bulldozer

The procedure employed by Watson (1967) which uses a conservation of angular momentum approach, has been used to determine the energy absorption requirements for both the K275 bulldozer and the Wa600 wheeled loader for roll slope inclinations of 15°, 30° and 45°. The simplifying assumptions of this procedure are outlined as follows:

- The effect of forward velocity is neglected and the calculations are therefore based on a two dimensional vehicle model only
- The initial conditions of the rollover assume that the vehicle’s centre of gravity lies directly above the point of rotation between the outside face of the wheel and the ground and at this point the angular velocity of the vehicle is considered to be zero.
- The kinetic energy of the vehicle at this point is deemed to be zero and therefore the effect of a tripping mechanism required to raise the centre of gravity to this level is disregarded
- The vehicle is treated as a rigid body with movement of the front axle and tyre deflection being disregarded
- The front and rear wheel tracks are equal
- The vehicle falls sideways falling freely under gravity
- The impact between the ground and the vehicle is assumed to be the ideal case of that between rigid bodies with the exception of the energy absorbing ROPS. During rollover there is considered to be no change in the angular momentum about the point of impact.
5.2.1 K275 Rollover Parameters and their Derivation

The following section provides a mathematical derivation of the estimated proportion of energy that will be dissipated under the first impact by the ROPS and ground interface during rollover of the K275 bulldozer on a constant rollslope of inclination \( \theta \). To assist the reader in interpreting the following equations, a list of symbols and their explanation has been provided in Table 5-1.

Table 5-1 Dynamic Rollover Parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
</tr>
</thead>
<tbody>
<tr>
<td>Height of centre of gravity</td>
<td>( X )</td>
</tr>
<tr>
<td>Wheel track width</td>
<td>( 2y )</td>
</tr>
<tr>
<td>Height of wheel</td>
<td>( h )</td>
</tr>
<tr>
<td>Height of safety frame</td>
<td>( H )</td>
</tr>
<tr>
<td>Width of safety frame</td>
<td>( 2B )</td>
</tr>
<tr>
<td>Moment of inertia about centre of gravity</td>
<td>( I_G )</td>
</tr>
<tr>
<td>Radius of gyration of vehicle about centre of gravity</td>
<td>( K )</td>
</tr>
<tr>
<td>Moment of inertia of vehicle about point A</td>
<td>( I_A )</td>
</tr>
<tr>
<td>Moment of inertia of vehicle about point B</td>
<td>( I_B )</td>
</tr>
<tr>
<td>Moment of inertia of vehicle about point C</td>
<td>( I_C )</td>
</tr>
<tr>
<td>Moment of inertia of vehicle about point D</td>
<td>( I_D )</td>
</tr>
<tr>
<td>Angular velocity of vehicle about point A</td>
<td>( \dot{\theta}_A )</td>
</tr>
<tr>
<td>Angular velocity of vehicle about point B</td>
<td>( \dot{\theta}_B )</td>
</tr>
<tr>
<td>Angular velocity of vehicle about</td>
<td>( \dot{\theta}_C )</td>
</tr>
</tbody>
</table>
The following equations have been developed with reference to Figure 5-1, Figure 5-2 and Figure 5-3 as the vehicle rotates and the ROPS comes into contact with the slope.
Figure 5-2 Impact on Wheel at Point B

Figure 5-3 Impact on ROPS at Point D

Gain in kinetic energy = Loss in potential energy

\[ M g \sqrt{x^2 + y^2} \sin \alpha \]

But K.E. \[ \frac{1}{2} M k^2 \left( x^2 + y^2 \right) A^2 \]
Angular momentum of bulldozer about B after impact = \( I_B \)

Gain in KE between B and D

Then total KE as the ROPS reaches the ground at D

where \( I_C \) is the angular velocity of the vehicle just before impact at D.

By equating the angular momentum before and after the impact at point D it is possible to derive the angular velocity \( I_D \) of the vehicle at D after impact and hence
the kinetic energy of the system after impact. The amount of energy dissipated between the ROPS and the ground may then be evaluated using the following expression:

\[
\text{Energy absorbed by the ROPS/Ground Surface} = KE_D(\text{Before Impact}) - KE_D(\text{After Impact})
\]

### 5.2.2 Determination of Moment of Inertia of Vehicle

The moment of inertia \( (I_G) \) about the vehicle’s centre of gravity, is a parameter that is not readily available from the vehicle manufacturer. In order to overcome this problem a two dimensional rectangular approximation of the vehicle was made, dimensions of which are outlined in Figure 5-4.

![Figure 5-4 Rectangular Approximation of K275 Bulldozer for Moment of Inertia Calculation](image_url)

Using this rectangular approximation for the vehicle and assuming that there is an even mass distribution and that the centroid of the vehicle is located 1.45m from the ground, the moment of inertia about the vehicle’s centroid may be estimated using the following equation:
\[ I_G = \frac{1}{12} M(a^2 + b^2) + Mc^2 \]  

Equation (5.1)

The terms a and b represent the length and width of the rectangle whereas M and r represent the mass of the vehicle and its distance from the centroid respectively. Applying this formula using values for a = 2.60 m, b = 2.56 m, c = 0.35 m and M = 49850 kg, the moment of inertia of the vehicle about the centroid was estimated to be approximately 62000 kgm². To determine whether this value was of the correct magnitude, it was compared with values that were used by Cobb (1976) during his numerical rollover study on tractors. Within this paper, Cobb chose a moment of inertia about the vehicle’s centroid of 60000 kgm² for a 50 tonne tractor which compared well with the value calculated from the above approximation.

5.2.3 Summary of Energy Absorption Levels of ROPS for Different Roll Slope Angles

The equations derived in the preceding sections have been applied to the K275 ROPS for roll slope angles of 15°, 30° and 45°. Table 5-2 shows a summary of the results obtained for the energy absorption requirements of the ROPS for each of these established roll slope inclinations. The calculated energy absorption levels for each rollslope angle were idealised as an equivalent amount of translational kinetic energy that would be transferred into the ROPS during the first impact of a rollover. Based on this idealisation, the classical kinetic energy equation was employed to determine the corresponding translational velocity that the ROPS would impact the ground surface with during a sideways rollover.

Table 5-2 K275 Dynamic Rollover Parameters

<table>
<thead>
<tr>
<th>Property</th>
<th>Units</th>
<th>Ground Slope (?)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>15°</td>
</tr>
<tr>
<td>(1) KE_A before impact</td>
<td>J</td>
<td>524748</td>
</tr>
<tr>
<td>(2) ?_A</td>
<td>Rad/s</td>
<td>2.05</td>
</tr>
<tr>
<td>(3) ?_B</td>
<td>Rad/s</td>
<td>1.94</td>
</tr>
<tr>
<td>(4) KE_B after impact</td>
<td>J</td>
<td>272814</td>
</tr>
<tr>
<td>(5) KE_{BD} gain</td>
<td>J</td>
<td>23015</td>
</tr>
<tr>
<td>(6) KE_D total before impact</td>
<td>J</td>
<td>295830</td>
</tr>
</tbody>
</table>
5.2.4 Energy Absorption by Soil

The amount of energy absorbed by soil during a rollover of a heavy vehicle can vary widely and is dependant on many factors. For the purposes of this study an approximation of the amount of energy absorbed by the soil was estimated using a procedure outlined by (Kacigin and Guskov, 1968). These authors suggested that the force developed in the soil normal to the ground slope could be approximated by the following equation:

\[ F_g = A p \tanh \frac{k \phi^2}{\frac{\phi}{2} \frac{\phi}{p}} \]  

Equation (5.2)

where \( A \) is the area of contact, \( k \) the coefficient of volumetric compression, \( p \) the bearing capacity of the soil and \( \phi \) the maximum soil deflection.

For this study, the maximum soil deflection \( \phi \) was set to 100mm whilst values for \( k \) and \( p \) were set to 20.7 kg/cm\(^3\) and 46.2 kg/cm\(^3\) respectively. Selection of each of these values was based on information provided by Cobb (1976) and was representative of average values that are appropriate for firm clay soils. These parameters in conjunction with the above mentioned equation, was used to construct the load deflection response profile for the soil which has been plotted in Figure 5-5. The energy absorbed by the soil was then determined by calculating the area under this curve. Once determined, this energy absorption quantity was then subtracted from the estimated amount of energy that the ROPS must absorb during the impact.
This small reduction in the energy absorption level, resulted in a slightly reduced translational velocity at impact.

![Graph showing force-deflection behaviour for hard clay soil](image)

**Figure 5-5 - Force-Deflection Behaviour for Hard Clay Soil**

### 5.3 Development of a Dynamic FE Model for K275 Bulldozer ROPS

An FE model of the K275 ROPS that was studied in chapter 4 was developed and subjected to dynamic impact forces that were characteristic of those sustained by the ROPS during the first impact of a sideways rollover. During a rollover, the impact between the ground surface and the ROPS results in energy absorption by both media and may therefore be classified as a deformable to deformable body impact. To simplify the modelling procedure, the ground surface was idealised as a rigid body that was able to transfer the estimated rollover kinetic energy into the ROPS. This kinetic energy transfer, was performed by assigning the vehicle’s mass to a rigid body and moving it laterally into the ROPS with a prescribed translational velocity. The velocity of the rigid body was adjusted to account for the estimated energy that would be absorbed by the ground surface during the impact in accordance with the recommendations provided by Kacigin and Gusakov (1968) for a hard clay surface. The kinetic energy imparted into the ROPS was derived from the results obtained using Watson’s procedure for each roll slope angle which has been summarised in
Table 5-2. A simplified illustration of the modelling procedure that was used for this dynamic impact study is shown graphically in Figure 5-6.

![Figure 5-6 Simplified Impact Modelling Procedure](image)

5.3.1 Model Setup

The geometry and mesh definition necessary to accurately model the ROPS, was developed using the pre-processor MSC Patran, in conjunction with the LS-DYNA pre-processor Femt V28.0. The surface geometry of the ROPS was defined with reference to the mid thickness section of each member and was meshed using quadrilateral shell elements.

The surface definition of the model which is shown in Figure 5-7, was composed predominantly of two major parts which included the ROPS which is shown in red and the rigid body which is shown in green. The inclusion of the corner radii for each section of the ROPS, was omitted from the analysis in order to simplify the connection region between the post and the beam. The global influence of omitting the corner radii from the model was studied in chapter 4 and found to have minimal effect on the lateral displacement and energy absorbing capability of a ROPS.
5.3.2 Elements

The Hughes-Liu shell element was chosen to model the response behaviour of the K275 ROPS under the established loading conditions. This particular element type is a reduced integration, large strain, shell element that consists of four nodes with six degrees of freedom per node. Selection of this element was based on its simplistic formulation and overall computational efficiency. The mesh density chosen for the ROPS was 20mm and the shell thicknesses that were implemented throughout the model were 10mm for the posts and 12mm for the beam. All nodes were equivalenced particularly at the connection region between the posts and the beam in order to permit uniform stress transfer throughout these regions. Figure 5-8 shows an illustration of the mesh distribution throughout the K275 ROPS model.
5.3.3 Material Properties

The performance of a ROPS, is based primarily on its ability to absorb energy, which is most commonly implemented through the formation of plastic hinges at specified locations within the structure. The selection of an adequate material model is vital to the performance of the ROPS and such a model must be capable of accounting for the nonlinear stress/strain behaviour of the chosen material. To accurately model this behaviour, the LS-DYNA nonlinear material model MAT_PIECEWISE_LINEAR_PLASTICITY was selected. This constitutive relation requires that the stress/strain behaviour of the steel be included in the form of a true stress versus plastic strain curve. The required material properties were calculated using Equation 4.1 and Equation 4.2 and were based on uni-axial tensile testing of coupons cut from the 350 grade RHS/SHS used in the experimental testing phase performed in chapter 3. Figure 5-9 shows the true stress versus plastic strain relationship that was
incorporated into LS-DYNA for all analyses. In addition to this, the following elastic properties were defined to account for the elastic component of the response.

- Material density $\rho = 7850 \text{ kg/m}^3$
- Material elastic modulus $E = 200000 \text{ MPa}$
- Poisson's Ratio $\nu = 0.3$

![Figure 5-9 – True stress versus Plastic Strain Distribution for ROPS Material](image)

### 5.3.4 Strain Rate Effects

The influence of strain rate effects on the dynamic response behaviour of the steel RHS/SHS that is used in the fabrication of ROPS, was incorporated into the LS_DYNA material model by adopting the Cowper Symonds constitutive relation. Cowper Symonds coefficients of $D = 950 \text{ s}^{-1}$ and $q = 4$ were chosen for the model based on research conducted by Johnson (2001) who used these parameters for a similar impact study involving 350 grade RHS.

### 5.3.5 Boundary Conditions

The boundary conditions applied to the model were designed to simulate full base fixity and involved restraining the translation and rotation of the base perimeter nodes of each post about the global X, Y and Z axes. An acceleration field was also applied to the model in the negative global Y direction to simulate the effects of
gravity on the model. This acceleration was applied using the Body load command within FEMB and was guided by a unit time loading curve.

5.3.6 Loading Procedure

The loading procedure for the impact study involved the use of a rigid impacting surface that was modelled as a rectangular plane of dimensions 240mm wide by 280mm high and 10mm thick. These dimensions were chosen based on a width equivalent to that of the ROPS post and a height based on the assumption that approximately 20% of the height of the ROPS post would come into contact with the ground during a rollover. The impacting body was meshed with a 40mm density using the Hughes-Liu shell element and assigned the LS-DYNA material type 20 MAT_RIGID. A rigid material property was assigned to the impacting body primarily to reduce the computational time required to perform the necessary analyses. The impacting body was constrained globally about all translational and rotational degrees of freedom with the exception of the global X axis translation. This degree of freedom was left unrestrained to enable the impacting surface to translate in the direction of the applied lateral velocity.

To enable the impacting body to transfer the correct amount of kinetic energy to the ROPS, it was assigned a mass equal to that of the K275 bulldozer. This mass was distributed evenly throughout the body through the assignment of an appropriate mass density. The rigid impacting body was given initial translational velocities of 2.71, 3.37 and 3.94 m/s in order to represent the impact velocities that would occur for rollovers on slopes with inclinations of 15°, 30° and 45° respectively.

5.3.7 Contact Definition

The contact definition between the impacting surface and the ROPS was modelled using the LS-DYNA contact type AUTOMATIC_NODES_TO_SURFACE. For each model, the impacting body was selected as the master surface, whilst the ROPS was selected as the slave surface. The static and dynamic coefficient of friction between the two surfaces was set to 0.6, which was in accordance with the value chosen by Cobb (1976) for a similar numerical rollover study. Other variables required by LS-DYNA for contact definition were set to their default values. No self contact was defined for any part of the ROPS, as is was found from visualisation of
the deformed ROPS structure that no elements came into contact with each other during the analysis.

5.3.8 Output Requests

Four different forms of output were requested from each finite element model. All results were graphed and visualised using the programs ETA PostGL and ETAGraph respectively.

DATABASE_BINARY_D3PLOT was requested so that the results of the model could be viewed using the post-processor eta/PostGL.

DATABASE_GLSTAT was requested to obtain the global energy data during the analysis which included the kinetic, internal, sliding and total energy of the system.

DATABASE_NODOUT was requested to track the displacement, velocity and acceleration of a node located at the centroid of the rigid body.

DATABASE_SPCFORC was requested to record the reaction forces at the supports of the ROPS during the analysis and assisted in development of the load deflection profile for the ROPS.

5.4 Results of Numerical Study

The following section presents the results of a numerical impact study that was performed for a single load case, involving impact of the K275 ROPS on a firm slope with an inclination of 30°. The member sizes chosen for the ROPS during this study were 150x250x10mm and 250x250x12mm for the posts and beam respectively. Following this discussion, the results of a detailed parametric analysis will be presented that has involved examination of the influence that ROPS post stiffness and rollslope inclination have on the dynamic response behaviour of ROPS.

5.4.1 Impact of ROPS with Rigid Surface

The impact simulation involved movement of the rigid surface into the stationery ROPS with a translational velocity of 3.37 m/s. During the initial contact phase between the two bodies, the rigid surface began to impart the stored kinetic energy
into the ROPS. This transfer of energy resulted in the ROPS deforming appreciably and was characterised by the formation of plastic hinges at the top and base of each post of the ROPS. This energy dissipation mechanism displayed by the ROPS during this event is captured vividly in Figure 5-10. Careful examination of this figure, which shows the Von Mises Stress distribution throughout the ROPS, confirms the presence of yielding at the hinge locations at the top and base of each post. The response behaviour of the ROPS during this phase, was characteristic of the collapse mode displayed by a typical fixed base frame subjected to a static sideward load and was very similar to the behaviour displayed by the same FE ROPS model that was studied earlier in chapter 4 under static loads.

The contact time between the ROPS and the impacting surface required to dissipate the kinetic energy of the impact was approximately 190ms. This contact time is purely dependant on the stiffness of the ROPS and the velocity of the impacting surface. For a stiffer ROPS impacting a firm surface, the contact time will be small, which will result in the transfer of large forces and peak decelerations into the ROPS. When the stiffness of the ROPS is reduced, the contact time will increase and will result in the transfer of much smaller forces and peak decelerations into the ROPS.
This response behaviour is more desirable for the occupant, however, will be characterised by larger deformations. The above mentioned predicted dynamic response characteristics of the ROPS will be further clarified in the parametric analysis section of this chapter.

5.4.2 Velocity and Peak Deceleration Response

The velocity versus time response of the rigid surface which is shown in Figure 5-11 was also measured during the impact at a node located at the rigid surface’s centroid. Examination of this figure, shows a linear reduction in the velocity of the rigid surface during the impact as the rigid surface was brought to rest from an initial velocity of 3.37 m/s over a contact duration of approximately 190 ms.

The variation in the peak deceleration of the rigid surface with time was also monitored at the centroid of the rigid surface during this event and is shown graphically in Figure 5-12. It is evident from this figure, that the response was characterised by significant fluctuations during the first 15ms of the impact with a peak recording of 5g taking place at a time of approximately 5ms. These high reading were due to the initial stiff response of the ROPS in the elastic region as the rigid surface came into contact with the ROPS. As the structure started to yield and the plastic hinge formation throughout the structure became more pronounced, the peak deceleration response of the rigid surface stabilised to an approximate mean value of 2g.

![Figure 5-11 Velocity of Rigid Surface During Impact](image-url)
Figure 5-12 Peak Deceleration of the Rigid Surface During Impact

5.4.3 Load Deflection Response

Figure 5-13 shows the load deflection response of the ROPS during the impact. The load, which is plotted on the Y axis of this figure, was determined from the summation of the base reaction forces at each post, along the direction of the applied impact. The deflection of the ROPS was measured by monitoring the displacement of the rigid impacting surface as it came into contact with the ROPS. The load deflection response behaviour of the ROPS shown in this figure indicates some initial fluctuations during the first 30mm of lateral deflection with peak readings of approximately 1200 kN being recorded. These initial fluctuations were some 33% higher than those recorded from the corresponding static finite element analysis performed in chapter 4. After this initial fluctuation, the response stabilised to an average peak load recording of 1000 kN. This loading level was approximately 20% higher than the peak static load carrying capability of the ROPS. This load was maintained up to the 125mm deflection mark and then fell gradually as the extent of the zones of plastic deformation spread throughout the ROPS until it was no longer able to carry any further load. The differences experienced in the load carrying capacity of the ROPS under static and dynamic loading conditions may be attributed
to dynamic amplification resulting from the influence of both strain rate and inertia effects.

As previously outlined, the energy absorbed by the ROPS can be determined from the area beneath the load deflection response profile. This quantity of energy absorption should be approximately equivalent to the kinetic energy imparted to the ROPS by the rigid surface with some minor variation taking place due to some energy dissipation through friction taking place between the two contacting surfaces. The kinetic energy versus time and energy absorption versus time response profiles for the impact are shown clearly in Figure 5-14. As predicted, the two curves are a reverse mirror image of one another with some minor variations taking place due to a small percentage of energy being dissipated through friction. In examining this figure which indicates that the ROPS was required to absorb approximately 250000 J of energy, it is interesting to note that this quantity of energy is almost three times the amount required from AS2294.2-1997. The difference between these two energy absorption levels suggests that there are some distinct discrepancies between the two approaches. As mentioned previously the methodology that was used to develop the code required energy absorption levels for varying forms of machinery is difficult to ascertain and quantify. In addition to this, the reader must also be reminded that the dynamic loads that have been used in this study, have been established from a
simplified mathematical model that has not been validated by testing. The main focus of this thesis is that it is possible to enhance energy absorption in ROPS through careful proportioning of ROPS members irrespective of soil conditions.

![Energy versus Time Response for ROPS During Impact](image)

**Figure 5-14** Energy versus Time Response for ROPS During Impact

### 5.5 Detailed Numerical Study

To develop a more comprehensive understanding of the energy absorption capabilities of this ROPS, a more detailed numerical investigation was performed that involved adjusting the stiffness of the ROPS posts. To achieve the required stiffness variation between the models, post sizes of 120x250x10, 150x250x10, 200x250x10 and 240x250x10 were chosen. Each model was analysed using LS-DYNA for simulated impacts on firm slopes with inclinations of 15°, 30° and 45°.

#### 5.5.1 Velocity and Peak Deceleration Response

Each simulation involved movement of a rigid impacting surface into the side face of the ROPS model with an appropriate velocity that corresponded to the angle of inclination of the rollslope. The velocity versus time response for each model has been plotted at a node located at the centroid of the rigid surface as it comes into contact with the ROPS. Figure 5-15, Figure 5-16, Figure 5-17 and Figure 5-18 display the results for each simulation with varying rollslope inclination. Each graph
shows a linear reduction in the velocity of the rigid body with time as it is brought to rest by the ROPS. It is evident from careful examination of these graphs, that the rate of change of velocity with time is independent of rollslope angle for a given ROPS post stiffness.

Figure 5-15 Velocity versus Time K275-230x250x10

Figure 5-16 Velocity versus Time Response K275-190x250x10
The other interesting relationship that may be distinguished from each of the above mentioned graphs is the variation in contact time required to bring the rigid body to rest and hence dissipate the kinetic energy of the impact. This relationship has been
further clarified by plotting the variation in contact time versus rollslope inclination for varying ROPS post stiffness which has been displayed in Figure 5-19. Examination of this figure shows that an approximately linear relationship exists between the contact time required to bring the rigid surface to rest and the angle of inclination of the rollslope. In addition to this, it is also evident from this graph, that this established relationship is inversely proportional to the stiffness of the ROPS posts, or more simply the contact time increases with increasing rollslope angle, however, reduces with increasing ROPS post stiffness.

![Figure 5-19 Contact Time versus Rollslope Angle for varying ROPS Post Stiffness](image)

**Figure 5-19 Contact Time versus Rollslope Angle for varying ROPS Post Stiffness**

The variation in the peak deceleration response of the rigid surface with time was also monitored during each impact simulation. Figure 5-20, Figure 5-21, Figure 5-22 and Figure 5-23 present the results obtained for each simulation with reference to these variables. The relationship depicted from each of these graphs, shows that an initial fluctuating response took place that characterised by some large peak deceleration readings. Further to these initial fluctuations, the deceleration response appears to stabilise to a more constant result which gradually diminishes with time. The suggested reasoning behind this resulting response behaviour is believed to be due to the initial stiff elastic response displayed by the ROPS during the initial contact with the rigid surface. This stiff response resulted in the generation of large peak decelerations, which subsequently reduced as the stiffness of each ROPS model...
reduced during the plastic deformation process incurred by each ROPS during the impact.

Figure 5-20 Acceleration versus Time Response K275-240x250x10

Figure 5-21 Acceleration versus Time Response K275-200x250x10
Another not iceable trend that is vividly clear from each of these graphs, is the relationship that exists between the initial peak deceleration experienced by the rigid surface during the impact and the ROPS post stiffness. As expected, an increase in ROPS post stiffness results in an increase in the peak deceleration experienced by the
rigid surface. This relationship has been further clarified through the development of Figure 5-24.

![Figure 5-24 Initial Peak Deceleration Response of Rigid Surface for increasing Rollslope angle and ROPS Post Stiffness](image)

5.5.2 Load Deflection Response

The load deflection response has been plotted for each impact simulation in accordance with Figure 5-25, Figure 5-26, Figure 5-27 and Figure 5-28. The magnitude of the load for each time step was calculated through summation of the base reaction forces measured in the direction of the applied impact. Reference to each of these figures, indicates that the response behaviour exhibited by each model was similar, with a higher force and lower deflection demand taking place for the higher stiffness ROPS. This result was expected, as the collapse load for each ROPS frame is directly related to the post stiffness and the energy dissipated by the ROPS is able to take place with less deflection, as the force demand is higher. The other noticeable trend that also exists is the similarity in the shape of each graph irrespective of the angle of inclination of the rollslope. Whilst the energy demand placed on a ROPS increases with increasing rollslope inclination, it appears to have minimal influence on the initial part of the load deflection response for a given ROPS configuration. It appears that the influence of strain rate effects for the narrow
velocity range covered within the study is reasonably uniform as the impact velocity for these types of rollovers is quite low and the variation in the influence of strain rate effects between each rollover simulation is only minor.

Figure 5-25 Load Deflection Response K275-230x250x10

Figure 5-26 Load Deflection Response K275-190x250x10
5.5.3 Energy Absorption

It is well understood that the energy absorbed by a ROPS may be determined from the area beneath the load deflection response profile. This quantity of energy should approximately equate to the applied kinetic energy that has been imparted to the
ROPS by the rigid surface. As discussed previously, some minor variations may take place as some energy is dissipated through friction as the rigid surface comes into contact with the ROPS during the impact. The kinetic energy versus time and energy absorption versus time response profiles for each impact simulation have been plotted together on the same graph and are shown clearly in Figure 5-29, Figure 5-30, Figure 5-31 and Figure 5-32. As predicted, the two curves represent a reverse mirror image of one another with some minor variations taking place due to the reasons outlined above.

![Figure 5-29 Energy versus Time Response of ROPS for K275-230x250x10](image)

Figure 5-29 Energy versus Time Response of ROPS for K275-230x250x10
Figure 5-30 Energy versus Time Response of ROPS for K275-190x250x10

Figure 5-31 Energy versus Time Response of ROPS for K275-150x250x10
Current ROPS standards give designers little guidance on how to adequately proportion a ROPS to adequately meet their specified requirements. Chapter 4 has addressed the simplified plastic method of analysis that can be used to proportion a ROPS to meet the regulatory requirements of the current ROPS standard AS2294-1997. The ROPS configurations that have been studied analytically within this thesis, have suggested that a carefully proportioned ROPS that possesses sufficient stiffness about the lateral direction may be able to satisfy the requirements of the standard adequately. The term ‘may’ has been used wisely here, as it is extremely important that the ROPS members be proportioned so that they will have sufficient strength to withstand the subsequent vertical and longitudinal loading requirements of the standard. As discussed in chapter 4, designers and manufacturers commonly develop rollover protective structures that are excessively stiff in order to avoid premature failure and subsequent retesting. The reason for this is driven by economical constraints as the nature of ROPS standards currently do not permit the use of analytical testing procedures for the certification of a ROPS. Through carrying out the dynamic impact simulations on the above mentioned ROPS models, it has been discovered that increased ROPS stiffness leads to a shorter contact time and the
development of larger reaction forces and consequently the transfer of increased peak decelerations to the vehicle’s occupants.

It is well understood that each of these response parameters are undesirable and may jeopardise an occupant’s chances of survival during a rollover. Carney, (1993) has suggested that unacceptably high decelerations can be responsible for severe occupant injury during vehicle collisions. In addition to this, the use of an overly stiff ROPS will also result in the generation of more elastic rebound energy, which in the case of a rollover may lead to multiple revolutions of the vehicle after the initial impact. Lu and Yu, (2003) have suggested that during an impact, the recoverable elastic energy may lead to further injury to the occupants of vehicles as well as the structure that is being protected. They proposed a simple model based on the collision of a vehicle with an elastic spring. During such a collision, the spring would compress which would cause the vehicle to decelerate. During this time the kinetic energy of the impact would be transferred into stored elastic strain energy in the spring. Under a situation such as this, where no plastic deformation is able to take place, the elastic strain energy will be released once the maximum elastic deflection capability of the spring has been reached. During this time, the stored elastic strain energy will be converted back into kinetic energy that will result in the acceleration of the vehicle in the opposite direction. Under a situation such as this, the authors suggest that this deceleration followed by a re-acceleration in the opposite direction may cause severe injuries to the vehicle’s occupants. Details of this analogy are presented graphically in Figure 5-33.
This simplified model has been used to address the adequacy of the performance of each ROPS during the impact simulations. Figure 5-34 shows the variation in the quantity of elastic rebound energy that takes place the plastic moment capacity of the ROPS posts are increased. The term plastic moment capacity has again been re-introduced to quantify the stiffness of the ROPS posts. It is clearly evident from this graph that the amount of elastic rebound energy that is released during a rollover impact increases with increasing ROPS post stiffness and is more severe for smaller rollslope angles.
Figure 5-34 Elastic Rebound Energy versus ROPS Post Plastic Moment Capacity for increasing Rollslope angle

5.5.5 Dynamic Amplification

The quantity of energy absorbed by the K275 ROPS under the established dynamic loading conditions for each stiffness configuration that has been examined, was compared with the corresponding energy absorption capabilities of the same ROPS model that was subjected to static loads in chapter 4. The procedure used to compare the results involved determining the amount of energy absorbed by the structure for a given deflection at regular intervals and then comparing the ratio of the dynamic and static energy absorption. The mean dynamic amplification was then determined based on a suitable number of results evaluated for each rollslope angle and ROPS post stiffness. The results of this procedure which are shown in Figure 5-35 emphasize that the energy absorbed dynamically by each ROPS model has been amplified by as much as 25% from its corresponding static energy absorbing capabilities. The main reason for DAF is the increased energy input due to the input kinetic and not the strain rate effects. This has been addressed as follows: Reasons for this resulting dynamic amplification have been mainly attributed to the influence of inertia effects arising from the input kinetic energy as well as some strain rate effects arising from the velocity of the impact on the material properties of the steel members. These findings suggest that a much greater energy demand will be placed
on a ROPS during a dynamic rollover event as compared with energy absorption criteria present within current ROPS performance standards.

![Graph showing Mean Dynamic Amplification of Energy Absorbed for Increasing ROPS Post Stiffness and Rollslope Angle](image)

Figure 5-35 Mean Dynamic Amplification of Energy Absorbed for Increasing ROPS Post Stiffness and Rollslope Angle

### 5.6 Development of Dynamic FE Model for Wa600 Wheeled Loader ROPS

The ROPS for the Wa600 wheeled loader that was studied under static loads in chapter 4 was subjected to a similar dynamic impact study using the same loading procedure that was employed for the K275 ROPS. LS DYNA was again used to conduct the numerical rollover simulations and to determine the resulting response behaviour for roll slope angles of 15°, 30° and 45°.

#### 5.6.1 Dynamic Loads for Wa600 Wheeled Loader

The estimated energy dissipation levels for the ROPS of the Wa600 were similarly approximated using the procedure presented by Watson (1967). Determination of the vehicle’s moment of inertia about its centroid was estimated using an idealised rectangular vehicle cross section which had a length \( a = 2.65 \text{m} \) and a width \( b = \)
Equation 5.1 and was determined based on the assumption that the vehicle’s centroid was located 1.4m from the ground. The Wa600 wheeled loader which has a gross vehicle weight of approximately 50t was calculated to have a moment of inertia about its centroid of approximately 57000 kgm$^2$ which was of similar magnitude to that of the K275 bulldozer and the resulting values adopted by (Cobb, 1976) for a 50t tractor. Figure 5-36 shows the rectangular approximation used to estimate the WA600’s centroidal moment of inertia.

![Figure 5-36 Rectangular Approximation of Wa600 Wheeled Loader for Moment of Inertia Calculation](image)

<table>
<thead>
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<th>Property</th>
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</thead>
<tbody>
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<td></td>
<td></td>
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</tr>
<tr>
<td>(1) $KE_A$ before impact</td>
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</tr>
<tr>
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</tr>
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</tr>
<tr>
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<td>J</td>
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<td>----------------</td>
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<td></td>
</tr>
<tr>
<td>(9) ?&lt;sub&gt;D&lt;/sub&gt;</td>
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<td>(10) Energy absorbed by soil</td>
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<td>(11) Energy absorbed by ROPS = (6)-(8)-(10)</td>
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<tr>
<td>(12) Translational velocity at impact</td>
<td>m/s</td>
<td>2.61</td>
</tr>
</tbody>
</table>

5.6.2 Energy Absorbed by Soil

(Kacigin and Guskov, 1968) proposed equation for the load deflection response of a hard clay surface was similarly employed and resulted in a load deflection response profile that is shown in Figure 5-37. The contact area between the ROPS and the ground was much larger for this ROPS, primarily due to its forwarding cross arm projection.

![Figure 5-37 Force deflection behaviour for hard clay soil](image)

5.6.3 Model Setup

Establishment of the model geometry and mesh definition was performed using MSC Patran. Once this had been completed, the LS-DYNA Preprocessor Femb was used to import this information and apply the additional parameters required to analyse the FE model successfully.
The model was again composed of two primary parts each of which have been distinguished by individual colour coding as shown in Figure 5-38. The zone shown in red represents the ROPS, whereas the green zone shows the extents of the rigid body used to deliver the kinetic energy of the impact.

### 5.6.4 Elements

The meshing procedure that was used for this model was similar to the one employed previously for the K275 ROPS. A 20mm mesh density was selected and the Hughes-Liu four node, large strain, shell element was chosen to simulate the response behaviour of the ROPS under the established dynamic loading conditions. The shell thicknesses that were implemented throughout the model were 10mm for the posts and 12mm for the beam. Figure 5-39 shows a graphical representation of the meshed ROPS.
5.6.5 Material Properties

The material model adopted for this ROPS was identical to the one used for the K275 ROPS. Strain rate effects were also included using the Cowper Symonds constitutive model with strain rate coefficients of $D = 950 \text{ s}^{-1}$ and $q = 4$.

5.6.6 Boundary Conditions

Fixed boundary conditions about all global degrees of freedom were applied to the base of each post of the Wa600 ROPS. In addition to this, the analysis was carried out under the influence of a global acceleration boundary condition applied to each node in the model which reflected the effects of gravity on the structure.
5.6.7 Loading Procedure

The loading of the ROPS was performed through a rigid impacting surface that was modelled as a rectangular plane of dimensions 640mm wide x 340mm high. The impacting body was meshed with a 40mm density using the Hughes-Liu shell element and assigned the LS-DYNA material type 20 MAT_RIGID. The impacting body was constrained globally about all translational and rotational degrees of freedom with the exception of the global X axis translation. This degree of freedom was unrestrained in order to enable the impacting body to translate in the direction of the applied initial velocity.

The mass of the Wa600 wheeled loader was spread evenly throughout the impacting surface through application of a 10mm surface thickness and an appropriate mass density. The rigid impacting body was given initial translational velocities of 2.61, 3.61 and 4.41 m/s in accordance with Table 5-3 so as to represent the impact velocities that would occur for roll slope angles of 15\(^\circ\), 30\(^\circ\) and 45\(^\circ\) respectively.

5.6.8 Contact definition

Contact was defined between the impacting surface and the ROPS using the LS-Dyna contact type AUTOMATIC_NODES_TO_SURFACE. The impacting surface was again set as the master surface, while the ROPS was selected as the slave surface. Static and dynamic coefficients of friction were again set to 0.6 and all other variables required by LS-DYNA for contact definition were set to their default values. Similar to the previous analyses for the K275 ROPS, there was no need for any self contact definition.

5.7 Results of Detailed Numerical Investigation

The following section presents the results of the detailed numerical investigation that was performed on the Wa600 ROPS. During this study, two parameters were varied which included the ROPS post stiffness posts and the inclination angle of the rollslope. Response parameters including the load deflection behaviour and energy absorbed by the ROPS as well as the peak deceleration and velocity of the rigid surface during each simulation were measured and have been plotted graphically. In total, nine impact simulations were performed on this ROPS.
5.7.1 Impact of ROPS with Rigid Surface

The technique used to transfer the kinetic energy of the rollover into the ROPS was identical to the one used previously for the K275 ROPS rollover simulation. For this particular ROPS which is classified as a two post cantilever type ROPS, there is a much larger contact area that will come into contact with the ground during impact. For this reason, the rigid surface used to transfer the impact kinetic energy into the ROPS was much larger than the one used previously for the K275 ROPS. Impact simulations were performed on the Wa600 ROPS for three different stiffness configurations that were obtained by adjusting the section properties of the posts. Figure 5-40 shows the deformed shape and the distribution of Von Mises stress throughout the ROPS for one of the impact simulations. It is evident from this figure that the ROPS has dissipated the applied kinetic energy through the formation of plastic hinges at the top and base of each post. This method of energy dissipation was similar to the one employed previously by the K275 ROPS and is characteristic of the collapse mechanism displayed by a fixed base framed structure under the action of a sideways applied load.

Figure 5-40 Maximum Von Mises Stress Distribution throughout ROPS during Impact
5.7.2 Velocity and Deceleration Response

The velocity versus time response for each dynamic simulation has been plotted in accordance with Figure 5-41, Figure 5-42 and Figure 5-43. The same procedure that was used previously to vary the stiffness of the ROPS was again employed, however, for this study only three different ROPS post configurations were considered. The angle of the rollslope was adjusted to simulate rollover impacts on firm slopes with inclinations of 15?, 30? and 45?. Figure 5-41, Figure 5-42 and Figure 5-43 show the velocity versus time response of the rigid body for a node plotted at the centroid of the rigid surface as it comes into contact with the ROPS during each simulation. The response displayed by each graph is similar to the one experienced previously for the K275 ROPS indicating a linear reduction in the velocity versus time profile of the rigid body as it is brought to rest by the ROPS. As observed for the K275 numerical study, it also appears that the rate of change of velocity with time is independent of rollslope inclination for a given ROPS post stiffness. The response shown for the 45? impact in Figure 5-43 suggests that the rigid surface was not brought to rest during the impact simulation. This result is correct, as the ROPS frame underwent severe plastic distortion during transfer of the kinetic energy from the rigid surface into the ROPS which resulted in subsequent structural collapse.

![Figure 5-41 Velocity versus Time Response of Rigid Surface for Wa600-200x250x10](image-url)
As expected it was observed from the numerical investigation that the contact time required to bring the rigid body to rest during each impact was purely dependent on
the stiffness of the ROPS frame. This result was in agreement with the findings presented for the K275 numerical study and has been shown graphically in Figure 5-44. Examination of this figure, suggests that contact time reduces with increased ROPS stiffness for increasing rollslope inclination, or more simply the stiffer the ROPS the smaller the contact time.

![Contact Time versus Rollslope Angle for Varying ROPS Post Stiffness](image)

**Figure 5-44 Contact Time versus Rollslope Angle for Varying ROPS Post Stiffness**

The variation in the peak deceleration of the rigid surface with time was also monitored during each impact simulation. Figure 5-45, Figure 5-46 and Figure 5-47 have been developed to capture the behaviour of this response variable. Each graph shows a fluctuating response with significant peak decelerations taking part in the initial phase of the impact where the ROPS exhibited a stiff elastic response. As discovered previously for the K275 simulations, the onset of plastic deformation resulted in a dramatic reduction in model stiffness which resulted in a gradual reduction in the deceleration response.
Figure 5-45 Acceleration versus Time Response of Rigid Surface for Wa600-200x250x10

Figure 5-46 Acceleration versus Time Response of Rigid Surface for Wa600-150x250x10
As observed previously, ROPS post stiffness can significantly influence the response behaviour of a rollover protective structure. The expected behaviour is that an increase in ROPS stiffness will lead to an increase in the peak decelerations transferred to the occupant of the vehicle and the generation of large force demands upon the ROPS as it attempts to dissipate the ensuing energy of the rollover. Figure 5-48 further clarifies this relationship by plotting the variation in initial peak deceleration with rollslope inclination for varying ROPS post stiffness. It is evident from this figure that the initial peak deceleration of the rigid surface as it contacts the ROPS, increases with both increasing rollslope angle and ROPS post stiffness.
The load deflection response of the ROPS has been plotted for each impact simulation in accordance with Figure 5-49, Figure 5-50 and Figure 5-51. The shape of the response profile for each impact condition is similar with some influence from strain rate effects coming into play as the rollslope inclination is increased and the rigid surface is projected into the ROPS with a higher velocity. It may also be observed from reference to these figures that the deflection experienced by the ROPS for the steeper roll slope angles is much greater as such an impact requires that the ROPS absorb a much larger quantity of energy. The peak load carrying capacity of each model as the stiffness was varied resulted in peak readings that were as much as 70% higher than the corresponding recorded collapse loads for the equivalent ROPS frames that were studied in chapter 4. The increased load carrying capacity was most pronounced for the lowest stiffness ROPS. Reasons for this can only be attributed to the influence of strain rate effects and the inertia effects created from the input kinetic energy. Despite the high peak readings for some of these models, the amplification of energy absorbed by each ROPS was much more reasonable and has been reported accordingly in the subsequent sections.
Figure 5-49 Load Deflection Response of ROPS for Wa600-200x250x10

Figure 5-50 Load Deflection Response of ROPS for Wa600-150x250x10
5.7.4 Energy Absorption

The energy absorbed by the ROPS which is approximately equivalent to the kinetic energy of the impact has been plotted for each simulation in accordance with Figure 5-52, Figure 5-53 and Figure 5-54. Examination of these graphs shows that the shape of each curve is similar with the kinetic energy representing the mirror reversed image of the energy absorbed as would be expected.
Figure 5-52 Energy versus Time Response of ROPS for Wa600-200x250x10

Figure 5-53 Energy versus Time Response of ROPS for Wa600-150x250x10
Figure 5-54 Energy versus Time Response of ROPS for Wa600-125x250x10

5.7.5 Elastic Rebound Energy

Figure 5-55 shows the variation in the quantity of elastic rebound energy that occurred for each simulation as the stiffness of the ROPS posts were increased. It is evident from this graph, that the amount of elastic rebound energy is proportional to the ROPS post stiffness and is most severe for impacts that take place on lower gradient rollslopes.
5.7.6 Dynamic Amplification

The amplification of energy absorbed dynamically by the Wa600 ROPS for each simulation was compared to the corresponding amount absorbed statically for each impact simulation. The results of this comparison have been presented graphically in accordance with Figure 5-56. Similarly to the previous study on the K275 ROPS, it is evident from this graph that mean dynamic amplifications of as much as 30% took place and were most severe for rollover impacts that took place with a lower stiffness ROPS. Reasons for the trend in ROPS post stiffness are attributed to an increased percentage of frictional energy dissipation that has occurred for the lower stiffness ROPS. This increase in energy dissipation through friction may be a result of the longer contact duration that occurred between the rigid surface and the ROPS during the impact. In addition to this finding, it is also evident from this graph that the mean dynamic amplification of energy absorbed by the ROPS increases with increasing rollslope angle. It is suggested, therefore, that the reason for this trend may be attributed to the influence of the strain rate sensitivity of the material and through the influence of inertia effects resulting from the input kinetic energy.

Figure 5-55 Elastic Rebound Energy versus ROPS Post Plastic Moment Capacity for increasing Rollslope angle
This chapter involved the use of finite element analysis to perform dynamic impact simulations on ROPS that would be characteristic of those encountered during a sideward rollover of an earthmoving vehicle on a firm slope of varying inclination. Dynamic loads for each simulation were developed based on a conservation of angular momentum approach and involved the idealisation of an overturning vehicle as a two-dimensional entity. The use of this approach resulted in the development of ROPS energy absorption criteria that exceeded the requirements of the current Australian standard by as much as 190%. This figure may seem alarming and may question the adequacy of the current Australian standard; however, it must be noted that the approach used is approximate and does not account for any further energy absorption by other parts or components of the vehicle. The exact philosophy behind the energy absorption principles in the standard are not clear which makes its accuracy difficult to question. It must be acknowledged also that Klose (1971) suggested that from testing, a ROPS was found to absorb only 20% of the kinetic energy of a rollover. Aside from accuracy of either method, the dynamic impact simulations showed that the methods of energy dissipation used by a ROPS were

Figure 5-56 Mean Dynamic Amplification of Energy Absorbed for Increasing ROPS Post Stiffness and Rollslope Angle

5.8 Summary of Chapter Findings
similar for both static and dynamic loading conditions. It was discovered that strain rate effects and inertia influences resulting from the input kinetic energy resulted in significantly higher peak loads for each ROPS frame studied. This increase in load was found to fluctuate in the initial load deflection response, however a more accurate assessment was made by establishing the amplification of energy absorbed. This figure for each of the ROPS configurations studied was found to be as much as 30%. by as much as 25% for the K275 rollbar ROPS and 30% for the Wa600 ROPS.

Further response parameters that were studied during the analysis were the relationship between peak decelerations and ROPS post stiffness. As expected the peak decelerations were found to be an increasing function of ROPS post stiffness and were also more pronounced for steeper rollslope inclinations. This result suggests that a stiffer ROPS may therefore be less desirable for occupant protection as opposed to a more flexible one. In addition to this the contact time required to dissipate the rollover energy was also studied and it was concluded that a stiff ROPS had a shorter period of contact which resulted in a higher level of elastic rebound energy, whereas a more flexible ROPS possessed a longer contact time and was therefore able to dissipate the rollover energy more effectively with much less elastic rebound energy. This finding also promoted the use of more flexible rollover protective structures for use as energy absorbing safety devices. The influence of the variation in surface characteristics on the energy absorbing capacity of a rollover protective structure was not studied in this chapter. All analysis results were presented for a surface profile that was representative of a hard clay soil. The next chapter studies the influence of surface properties on the energy absorbing capacity of a ROPS by adjusting the time of contact between the ROPS and the ground surface.
Chapter 6 Transient Pulse Loading

Chapter 5 involved subjecting a number of ROPS frames to simulated dynamic impact loads that were representative of the impact condition that would occur during a sidewards vehicle rollover. The modelling technique utilised for this study examined the response behaviour of a number of ROPS configurations for rollover impacts on firm slopes with varying inclinations. The results of the study indicated that ROPS post stiffness played an important roll in controlling the energy absorbing capacity of a ROPS. It was found that a ROPS proportioned based on the minimum static load provision of the ROPS standard AS2294-1997, could adequately sustain an initial sidewards impact on a firm rollslope with an inclination as high as 30° without DLV violation. In addition to this finding, the study also revealed that the use of an overly stiff ROPS, resulted in the generation of high peak decelerations and reaction forces that may be detrimental to the occupant’s chances of survival during the occurrence of a rollover.

To further enhance the understanding of the impact response of rollover protective structures, a more detailed dynamic study was performed that involved the use of transient pulse loads. It is well understood that the surface properties have an influence on the impact duration of a rollover. Based on this notion, the time duration of the impulse was varied in order to study rollover impacts that could take place on a variety of surface conditions. For this study, the calibrated K275 ROPS model that was examined earlier in chapters 4 and 5, was used.

6.1 Transient Pulses

In the field of vehicle crashworthiness, impact studies have shown that during frontal collisions, automotive vehicles are subjected to a force distribution that commonly takes the form of an impulse curve. Common impulse curves used for the evaluation of such events may be either haversine, halfsine, triangular or square in shape. The shape of the loading distribution that is placed on a ROPS during an impact arising from a sidewards rollover, is unknown and has received little attention by researchers in the open literature. In the absence of such information, an assumption has been
made that idealises the rollover impact as a transient half-sine pulse. The duration of this pulse is an additional parameter that has not been clearly defined by other researchers in this area. Some guidance, however, has been provided by the earthmoving machinery manufacturer Caterpillar, who, during the late 1960’s performed a series of full scale dynamic rollover tests on a variety of earthmoving machines on rollslopes of varying inclination and soil type. Visualisation of this video footage suggested that the contact time between the ROPS and the ground during a rollover may lie within the 100 to 300ms range. Based on the results of this study, a series of transient pulse loads with durations ranging between these limits were applied to the established FE model of the K275 ROPS using the explicit FE code LS-DYNA.

6.2 Determination of Pulse Variables

As discussed in chapter 5, angular momentum considerations may be used to assess the energy absorption requirements of a ROPS fitted to a vehicle that is undergoing a sideways rollover on a constant slope. Figure 6-1 which was developed in chapter 5 has been used to assist the reader with the interpretation of the derivations that follow.

![Figure 6-1 Vehicle Rollover Model](image)

The following fundamental expressions have been historically used in the field of dynamics to explain the principles of motion. These expressions in association with
their rotational equivalents, have been presented to assist the reader with the interpretation of the resulting pulse load derivations.

Newton’s second law of motion states that the force acting on a body is equal to the product of its mass multiplied by its acceleration

\[ F = ma \quad \text{Equation (6.1)} \]

where \( F \) represents force, \( m \) the mass and \( a \) the acceleration

Another fundamental term which may be introduced is linear momentum which represents the product of mass and velocity

\[ p = mv \quad \text{Equation (6.2)} \]

where \( p \) represents linear momentum, \( m \) the mass and \( v \) the velocity

For zero initial velocity and noting that \( v = at \) Newton's second law may be rewritten as:

\[ F = \frac{p}{t} \quad \text{Equation (6.3)} \]

The rotational equivalent of Newton's second law may be expressed in the form of:

\[ \tau = I \alpha \quad \text{Equation (6.4)} \]

where \( \tau \) represents torque, \( I \) the polar moment of inertia of the body about the centre of mass and \( \alpha \) the angular acceleration of the rotating body

Angular momentum represents the product of the moment of inertia of a body multiplied by its angular velocity and may be expressed by the following equation:

\[ L = I\omega \quad \text{Equation (6.5)} \]

where \( L \) represents the angular momentum of the body about the axis of rotation, \( I \) the moment of inertia of the body about the centre of mass and \( \omega \) the angular velocity of the body.
Noting that \( \frac{L}{t} \) for zero initial rotational velocity Newton’s second law may be rewritten as:

\[ \frac{L}{t} \text{ Equation (6.5)} \]

\( \tau \) represents torque which is equivalent to a force multiplied by its perpendicular distance from the point of rotation, this equation may be therefore rewritten as:

\[ F \cdot \frac{\tau}{D} \text{ Equation (6.6)} \]

where \( F \) represents the tangential force on the body, \( \tau \) the torque on the body and \( D \) the perpendicular distance about the point of rotation.

Based on these above mentioned derivations, and noting that the distance \( D \) is equal to \( BD \) from Figure 6-1 the following equation may be used to represent the impulse for the rollover condition established above:

\[ \tau D \int_{t_0}^{t_1} F dt \text{ Equation (6.7)} \]

Integrating this equation between the limits of \( t_1 \) and \( t_0 \) simplifies to the following expression:

\[ \int_{t_1}^{t_2} BD \int_{t_1}^{t_2} F dt \text{ Equation (6.7)} \]

noting that \( t_0 = 0 \) and the initial rotational velocity is zero, the following expression may be used to determine the intensity of the impulse force to be applied to the ROPS:

\[ F = \frac{I}{BDt_1} \text{ Equation (6.8)} \]

This formula may then be equated to the area of a half–sin curve as follows:
\[ F = \frac{P \cdot \frac{d}{2TB}}{TBD} \sin \frac{T}{T BD} \]  

Equation (6.9)

where \( T \) represents the total duration of the pulse and \( t \) the cumulative intermediate time step.

From this equation, selection of a number of different pulse durations will therefore enable the impact response of a ROPS on surface profiles with varying conditions to be assessed.

### 6.3 Model Setup

An FE model similar to the one for the K275 ROPS that was studied using dynamic impact loads in chapter 5 was developed and subjected to a series of dynamic pulse loads. The geometry and mesh definition for this model was identical to the model that was used for the chapter 5 impact study. For this investigation, a 150x250x10 RHS and a 250x250x12 RHS were used for the posts and beam of the ROPS respectively.

#### 6.3.1 Elements

The element type chosen for this analysis was the Hughes-Liu shell element with a mesh density of 20mm.

#### 6.3.2 Material Properties

Two material models were used to simulate the response of the ROPS under the applied pulse loads. The first model that was used was the LS-DYNA nonlinear material model MAT_PIECEWISE_LINEAR_PLASTICITY. This material model was applied to all of the elements of the ROPS except those that were within close proximity to the loading zone. For the loading zone region, an elastic strip was incorporated into the model to avoid excessive deformation of the ROPS material. Implementation of this strip involved specifying the LS-DYNA material model MAT_ELASTIC to all elements within this region. Similar to the previous dynamic impact study that was performed in chapter 5, the elastic material properties that are outlined below, were specified for this material model as well as for the elastic portion of the nonlinear material model.
Strain rate effects were incorporated into the plasticity material model using the Cowper Symonds relations with coefficients that were identical to those used previously in chapter 5.

6.3.4 Boundary Conditions
The boundary conditions applied to the model were identical to those used in chapter 5 and involved full restraint of the base perimeter nodes of each post about all global axis directions. The FE model was also assigned an acceleration field in the negative global Y axis direction to simulate the effects of gravity.

6.3.5 Loading
Loading of the ROPS involved application of a face pressure load to the outer surface of the top corner of the ROPS over a 250mmx250mm zone. The intensity of the face pressure was dependent on the time of contact used for the pulse load. The contact time of the pulse was varied between 100 and 300 ms to account for differing surface conditions that may be experienced by the ROPS during rollovers on rollslopes with inclinations ranging between 15° and 45°. Table 6-1 provides a summary of the peak force intensities that were applied to the K275 ROPS model during the numerical study. The shape of each force pulse has also been shown in accordance with Figure 6-2, Figure 6-3, Figure 6-4, Figure 6-5 and Figure 6-6 for rollslope angles of 15°, 30° and 45°. The FE model was subjected to each dynamic pulse load for the defined rollslope inclinations giving rise to a total of 15 FE simulations.

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Table 6-1 Peak Impulse Load Summary
Figure 6-2 100 ms Load Pulses

Figure 6-3 150 ms Load Pulses

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</table>
Figure 6-4 200 ms Load Pulses

Figure 6-5 250 ms Load Pulses
6.3.6 Contact Definition

For this analysis there was no need for any contact definition as loading of the ROPS was performed through the application of face pressures only.

6.3.7 Output Requests

The output requests from LS-DYNA were similar to those used previously in chapter 5 and involved monitoring the displacements of certain nodes located on the ROPS in association with the recording of the base reactions forces that were determined for the base perimeter nodes of each post. Energy data and visualisation output data were also recorded for each simulation.

6.4 Results of Numerical Study

The following section presents the results of the numerical study that was performed on this ROPS model for the impulse loads that have been outlined above. As discussed previously each model was subjected to a series of dynamic pulse loads in order to study the effect of vehicle rollovers on rollslopes with varying surface properties. The intensity of the load applied to the FE model was based on the established duration of the pulse with the operating mass of the vehicle kept constant.
for all pulse load calculations. Reference to Table 6-1 indicates that the intensity of the applied load was most severe for impacts that were of short duration. For certain pulse durations, the peak force intensity of the pulse was calculated to be well above the collapse load of the ROPS which resulted in premature failure. In order to account for this, a termination condition was introduced into each model. This condition allowed the analysis to be prematurely stopped once the zone of the DLV had been violated and the ROPS was no longer able to provide protection to the vehicle’s operator. For this particular ROPS configuration, this deflection limitation was set to 500mm. Each model was analysed for the load pulses described above and the load deflection behaviour of the ROPS was plotted accordingly and has been displayed in Figure 6-7, Figure 6-8 and Figure 6-9. Reference to each of these graphs shows that for short duration impulses, the zone of the DLV was impeded during each analysis as the 500mm DLV deflection limitation was reached which represented the deflection limit prior to DLV infringement. In general, each curve shows a response that is characteristic of the one experienced by a simple fixed base framed structure under a sideways applied static force. The interesting difference in this response is the second peak that takes place in the load deflection response of the model prior to reaching the DLV deflection limitation. This behaviour is believed to be a second order effect that has been characterised by the extensive plastic deformation sustained by the ROPS during this loading sequence in association with the influence of strain rate effects on the model taking place. In addition to these load deflection response profiles, the Von Mises stress distribution throughout the ROPS and the extent of the plastic deformation experienced by the ROPS for a 100ms impulse duration on a 30° rollslope have also been shown graphically in Figure 6-10. Similar to the previous dynamic study that was performed in chapter 5, plastic hinges have developed at the top and base of each post which can be clearly seen in this figure.
Figure 6-7 Load Deflection Response for 45° Rollslope

Figure 6-8 Load Deflection Response for 30° Rollslope
The energy absorbed by the ROPS for each pulse duration and the associated rollslope angles have been summarised in Table 6-2 and also graphically in Figure 6-11. Typically, the trend that may be depicted from each of these figures, is that the
short duration pulses which are characteristic of impacts that would occur on hard surfaces will result in significant plastic deformation that is characterised by a high force/deflection demand and a corresponding large amount of energy absorption. For large duration pulses that are characteristic of softer surface impacts, the force/deflection demand placed on the ROPS will be small and consequently the energy absorbing capacity of the structure will also be small. For events such as this, the soil will be forced to absorb more of the impact energy, whereas for impacts on firm surfaces, the ROPS will be forced to absorb a much larger proportion of the rollover energy.

Table 6-2 Energy Absorption Levels during Pulse Loads

<table>
<thead>
<tr>
<th>Time (ms)</th>
<th>15° Rollslope</th>
<th>30° Rollslope</th>
<th>45° Rollslope</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>389945 J</td>
<td>392231 J</td>
<td>425365 J</td>
</tr>
<tr>
<td>150</td>
<td>367052 J</td>
<td>372772 J</td>
<td>396527 J</td>
</tr>
<tr>
<td>200</td>
<td>14811 J</td>
<td>361993 J</td>
<td>371378 J</td>
</tr>
<tr>
<td>250</td>
<td>5410 J</td>
<td>12811 J</td>
<td>363511 J</td>
</tr>
<tr>
<td>300</td>
<td>4163 J</td>
<td>5823 J</td>
<td>10641 J</td>
</tr>
</tbody>
</table>

The relationship between the energy absorbed by the ROPS and the pulse duration for varying rollslope inclinations have been shown graphically in Figure 6-11. This graph shows that the energy absorbed by the ROPS is a decreasing function of the pulse duration and highlights that much larger energy demands are placed on a ROPS during an impacts on a firm soil when compared to a corresponding impact on a softer soil. The other noticeable trend that is evident from this graph is a slight increase in the energy absorbing capacity of a ROPS for an impact occurring on a steeper rollslope. The reason for this increase may be attributed to the influence of inertia effects as a higher impact velocity is associated with a steeper rollslope. The influence of this parameter may also be distinguished through reference to Figure 6-7, Figure 6-8 and Figure 6-9. As shown by each of these graphs, the second peak in the load carrying capacity of the ROPS increases with increasing rollslope inclination further emphasising the possible influence of the strain rate sensitivity of the ROPS material.
6.5 Chapter Summary

A detailed impact study was performed that assessed the response behaviour of a K275 bulldozer ROPS to applied impulsive forces that were characteristic of those that may be experienced by a ROPS during impacts on ground surfaces with varying material characteristics. The established FE model was subjected to pulse loads with contact durations in the range of 100-300ms to represent a range of surface conditions varying from hard to soft. From the analysis it was discovered that shorter durations pulse loads resulted in complete collapse of the ROPS structure and consequently resulted in large energy absorption demands being placed on the ROPS. The influence of the strain rate sensitivity of the ROPS material was also found to have only a minor influence on the energy absorption capability of the ROPS, with a general trend indicating that steeper angle impacts resulted in slightly higher levels of energy absorption demand being placed on the ROPS. Long duration pulse loads were found to place lower energy absorption demands on the ROPS by forcing the ground surface to absorb more of the rollover energy. Finally this chapter has also demonstrated the power and feasibility of using analytical techniques for evaluating ROPS performance which was one of the aims of the project.
Chapter 7 Enhancement of Energy Absorption in ROPS

Rollover Protective Structures have to date relied on the permanent plastic deformation of their structural members to absorb the kinetic energy associated with a rollover impact. Under normal operating conditions, this form of energy absorption has been found to provide an acceptable level of protection to the occupants of the vehicle provided that the ROPS has been proportioned carefully to absorb the required amount of energy. The use of supplementary energy absorbing devices has received widespread attention in the automotive industry to assist with the protection of occupants during frontal and sideward collisions. The incorporation of such devices in a ROPS to enhance the level of protection afforded to the occupant during a rollover has, however, received minimal attention. This chapter examines the feasibility of including a cost effective supplementary energy absorbing device into the frame work of a rollover protective structure. The type of device that has been chosen for evaluation is a thin walled tapered tube that is known as a frusta. Through inclusion of this device it is envisaged that the energy absorption capabilities of a ROPS will be significantly enhanced and that there will be a reduction in the severity of the plastic deformation experienced by the ROPS during a rollover. In addition to this, the potential for further roll revolutions will also be reduced which will result in an increased level of safety for the vehicle occupants. The method used to study the feasibility of using such a device has involved extensive finite element analysis of a ROPS/ Frusta assembly using the FE modelling software LSDYNA V970.

7.1 Frusta Description

As discussed in chapter 2, a frusta is a thin walled tube that is commonly manufactured from mild steel, that is designed to absorb kinetic energy through axial crushing and buckling of its sidewalls. The numerical analysis of Frusta under dynamic impact loading has recently been addressed by researchers such as Mamalis (2000) and Nagel (2004). Nagel (2004) acknowledged that tapered frusta were one of the most desirable impact energy absorbers due to their relatively stable mean load deflection response under dynamic loading. In addition to this positive response characteristic, Nagel also suggested that frusta were able to withstand off axis oblique loading equally as well as axial loads. These two promising energy absorbing
characteristics, together with the frusta’s relatively inexpensive fabrication cost make this type of energy absorber a particularly attractive option for use in a rollover protective structure. For the purposes of this study, the reader is advised that the effectiveness of using a device such as this will be limited to impacts that occur on firm soil surfaces only.

7.2 Development of FE Model for K275 ROPS/ Frusta Assembly

A small thin walled tapered frusta was included in the FE model of the K275 ROPS that was developed and used in chapters 4 and 5. The base dimensions of the frusta were chosen to be 250mm x 250mm to suit the widest plan dimensions of the ROPS post. The height of the frusta was limited to 160mm so that it could be concealed by additional side plates that may form part of the roof structure of the ROPS for in practice aesthetic purposes. The initial wall thickness given to the frusta was 4mm, however, this parameter was varied to study the effect that it had on the system’s energy absorbing capabilities. A constant taper angle of 10° was applied to each face of the frusta in order to promote effective energy absorbing characteristics for oblique loading conditions. The frusta was located at the exterior corner junction between the posts and beam of the ROPS. The positioning of the frusta in this location meant that it would become the first point of contact during a sideways rollover of the vehicle and would therefore be able to absorb the maximum amount of energy. The numerical modelling technique that was used to simulate a sideways impact and hence assess the effectiveness of this supplementary device was similar to the one used previously in chapter 5. For this simulation, a rigid impacting surface was translated into the face of the ROPS/Frusta assembly with a constant impact velocity. The translational velocity of the impacting surface was later varied to simulate impacts that could occur on roll slopes with varying inclinations. Details of the ROPS/Frusta assembly are shown in Figure 7-1.
7.2.1 Model Setup

The geometry and mesh definition for the ROPS was taken from the previous model that was developed for the chapter 5 dynamic impacts studies. The geometry for the frusta was added to the model and meshed accordingly using a similar mesh density to the ROPS. The perimeter nodes linking the frusta to the side of the ROPS were equivalenced to enable the smooth flow of stress between the two segments. This modelling condition was used to represent the fillet welding process that would be used to attach the frusta to the ROPS for in practice use.

7.2.2 Elements

Similar to all previous dynamic studies, the Hughes-Liu shell element was chosen to simulate the structural behaviour for both the ROPS and frusta segments. The shell thicknesses that were used for the system were 10mm for the posts, 12mm for the beam and 4mm for the frusta. Details of the mesh definition for the assembly have been displayed graphically in Figure 7-2.
7.2.3 Material Properties

The material properties used for the ROPS and the impacting surface were identical to those employed previously in chapter 5. The material properties for the frusta were obtained from tensile testing performed by Nagel (2004) on test specimens taken from typical 4mm thick mild steel plate. Details of the true stress versus plastic strain relationship that was incorporated into the LS-DYNA material model for this element has been displayed graphically in Figure 7-3. In addition to these plastic material properties, the following additional elastic properties were employed for the frusta:

- $E=200000$ MPa
- $\mu=0.3$
- $\gamma=7850$ kg/m$^3$
7.2.4 Strain Rate Effects

The Cowper Symonds constitutive relation was used to model the influence of the strain rate sensitivity of the mild steel that was used in the fabrication of the ROPS/Frusta assembly. The Cowper Symonds coefficients D and q that were used previously in chapter 5 for the ROPS, were again employed, however, for the frusta materia, Cowper Symonds parameters $D = 6844 \text{ s}^{-1}$ and $q=3.91$ were selected based on information provided by Nagel (2004).

7.2.5 Boundary Conditions

The boundary conditions for the model were identical to those used previously in chapter 5 and involved restraining the translation and rotation of the base perimeter nodes of each post about the global X, Y and Z axes. In addition to this, a global vertical acceleration was applied to the model to simulate the influence of gravity.

7.2.6 Loading

The loading procedure for the model involved the transfer of kinetic energy from a rigid impacting surface into the ROPS/Frusta assembly. The rigid impacting surface was provided with the same geometric and mass properties as the model that was used in chapter 5. The impacting surface was projected into the assembly with translational velocities of 2.71, 3.37 and 3.94 m/s. These velocities corresponded to
the translational velocities that would be generated during the initial impact between
the assembly and the ground surface for a sideward rollover on firm surface with
inclinations of 15°, 30° and 45° respectively.

7.2.7 Contact Definition
The contact definition for the model involved the use of the LS-DYNA contact type
AUTOMATIC_NODES_TO_SURFACE for the contact between the rigid body and
the Frusta. To model the collapse mode displayed by the frusta during impact, the
contact type AUTOMATIC_SINGLE_SURFACE was employed. This contact
algorithm is a penalty based method which has the capability of modelling self
contact between each element as they touch each other during progressive collapse of
the member. Implementation of this contact definition is essential for capturing the
correct post yield response of the frusta during axial collapse. The static and dynamic
coefficient of friction for all contact types was set to 0.6.

7.2.8 Output Requests
The output requests that were used to monitor the response behaviour of the
ROPS/Frusta assembly were identical to those in chapter 5 and are briefly
summarised below:
- DATABASE_BINARY_D3PLOT
- DATABASE_GLSTAT
- DATABASE_NODEOUT
- DATABASE_SPCFORC

7.3 Results of Numerical Study
The following section presents the results of the impact study that was performed on
the ROPS/Frusta assembly for a 30° rollslope and a frusta wall thickness of 4mm.
Following this discussion, the results of a parametric analysis are presented that
involved varying the wall thickness of the frusta as well as the rollslope angle and
studying the corresponding structural response of the system. It must be noted that all
of the results that are presented in the following sections have been plotted with
reference to a node that is located at the centroid of the rigid impacting surface.
7.3.1 Load Deflection Response under Impact

The exchange of kinetic energy from the rigid impacting surface into the combined ROPS/ Frusta assembly resulted in a load deflection response profile that has been shown in Figure 7-4. Examination of this figure gives rise to an initial fluctuating load deflection response in the response phase up to approximately the 100mm lateral deflection mark. After this phase, it is evident that the structure exhibited a more stable load deflection response until the maximum deflection of 350mm was reached. The initial phase of the load deflection response profile was characterised by the severe deformation experienced by the Frusta as it came into contact with the impacting surface. During this phase, the sidewalls of the Frusta began to buckle which resulted in a sudden drop in the load carrying capacity of the system. As the sidewalls began to fold up upon one another, however, the load carrying capacity of the system began to increase. This response went through two complete oscillations of loading and unloading until all of the energy absorbing capabilities of the frusta had been expended. Following this initial behaviour, the ROPS began to absorb the remaining energy available in the system and underwent plastic deformation that was characteristic of a typical ROPS frame subjected to a dynamic impact load. This behaviour was governed by the formation of plastic hinges at the top and base of each post as the structure reached its peak load carrying capacity. Details of the deformation mode experienced by the Frusta during the initial impact phase in association with the extent of the plastic deformation experienced by the ROPS have been shown in Figure 7-5 and Figure 7-6 respectively.
Figure 7-4 Lateral Load Deflection Response for K275 ROPS with 4mm Frusta and Rollslope angle of 30°

Figure 7-5 Von Mises Stress Distribution and Plastic Hinge Zones
7.3.2 Mean Load Deflection Response

The mean load deflection response may be used to assess the effectiveness with which a structure absorbs energy. The mean load response variable is essentially the total cumulative energy absorbed by the structure divided by the corresponding lateral deflection of the structure. A structure that exhibits a stable mean load deflection response that does not decline rapidly, will therefore exhibit a response that closely approximates that of a rectangle which is the ideal energy absorber response. Through examining the mean load deflection profile for the system shown which has been shown in Figure 7-7, it is evident that aside from the initial fluctuations, the structure has absorbed energy at a progressively increasing rate and may therefore be classified as an adequate energy absorbing system.
7.3.3 Velocity and Deceleration Response of Impacting Surface

The velocity versus time profile of the rigid body as it came into contact with the combined assembly has been presented in Figure 7-8. This graph shows that a contact time of approximately 210 ms was required to bring the rigid surface to rest during the simulation. This contact time was approximately 20 ms longer than the corresponding contact time without the frusta in place. In addition to this response profile, Figure 7-9 presents a plot of the peak deceleration versus time response behaviour of the rigid surface during the impact. It is evident from this figure that the deceleration of the rigid surface has fluctuated abruptly during the first 50ms of the impact. Peak decelerations experienced by the system during this phase reached levels as high as 2.5g. These measurements were approximately 30% lower than those experienced by the same ROPS that was examined earlier in chapter 5 without the inclusion of the Frusta. This result indicates that the frusta was able to reduce the peak decelerations developed during the impact.
7.3.4 Energy Absorption

Figure 7-10 shows the combined kinetic energy and energy absorption versus time response of the structure during the impact. As expected these two curves are a reverse mirror image of one another and are similar in shape and magnitude to the response profiles obtained for the same ROPS without the ROPS in place except for
a 20 ms lag created by the inclusion of the frusta. Careful examination of this figure indicates that the structure has absorbed energy at a progressively stable rate during the first 210 ms of the impact, as the velocity of the impacting surface was brought to rest. After this time was reached, there is a slight decrease in the energy absorption rate of the system as the stored elastic energy within the structure was recovered during rebounding of the rigid body from the surface of the structure. For this particular assembly of ROPS/Frusta stiffness, the amount of elastic recovery energy is very minimal, which indicates that the structure has dissipated most of the input kinetic energy through plastic deformation. This point is further clarified by examining the kinetic energy versus time profile for the system. Clearly, from examination of this figure, the combined assembly has absorbed the initial kinetic energy given to the rigid surface with very minimal elastic recovery taking place.

![Energy versus Time Response for 30° Rollslope Impact and 4mm Frusta](image)

**Figure 7-10** Energy versus Time Response for 30° Rollslope Impact and 4mm Frusta

### 7.4 Detailed Numerical Study

A detailed numerical investigation was carried out to determine the influence of critical parameters on the energy absorbing capabilities of the combined ROPS/Frusta system. For this particular analysis, no adjustments were made to the stiffness of the ROPS or the geometric definition of the frusta. The parameters that
were changed during this study, included the wall thickness of the Frusta which was varied between 4mm and 2mm and the translational velocity of the impacting surface.

Each model was run for a set frusta wall thickness at velocities of 2.71, 3.37 and 3.94m/s which corresponded to rollover impacts on roll slopes with inclinations of 15°, 30° and 45° respectively. A total of 9 FE models were run during this study in order to determine the feasibility of using a supplementary energy absorbing device to enhance the level of protection afforded to the occupants of heavy vehicles during a rollover. It must be noted that this numerical study involved examining the influence of only a limited number of parameters. For a more detailed explanation on the influence that other key variables have on the energy absorbing capabilities of thin walled rectangular tubes under impact loading, the reader is referred to research performed by Nagel (2004).

7.4.1 Load Deflection Response
The load deflection response behaviour of the system for varying rollslope inclinations has been plotted in accordance with Figure 7-11, Figure 7-12 and Figure 7-13 for a set velocity with varying frusta wall thickness. The trends displayed by each graph are similar with a fluctuating load deflection response in the initial phase as the frusta sidewalls buckle and come into contact with each other, followed by a more stable response as the ROPS begins to form plastic hinges to dissipate the remaining available kinetic energy. It appears from examination of each of these graphs that the thicker walled frusta is able to absorb a much greater amount of energy than its corresponding thinner walled counterparts under the same deflection. This result is expected as there is an increased quantity of material available for plastic deformation and subsequent energy absorption. The other noticeable trend that may be depicted from these graphs is the lagging effect that reducing the frusta wall thickness imposes upon the structural behaviour of the system. It is evident that as the wall thickness of the Frusta is reduced, its energy absorbing capability is also reduced which means that the ROPS must undergo further plastic deformation in order to absorb the remaining amount of kinetic energy. The increase in the plastic deformation sustained by the ROPS is characterised by an increase in the deflection of the structure which can be seen clearly in each of these figures.
Figure 7-11 Load Deflection Response for 45° Rollslope Impact

Figure 7-12 Load Deflection Response for 30° Rollslope Impact
The mean load deflection response for each model has been plotted for each set rollslope angle in accordance with Figure 7-14, Figure 7-15 and Figure 7-16. The overall response behaviour shown by each of these figures is very similar with a higher magnitude in mean load taking place as the frusta wall thickness is increased. This result was expected, as the increased wall thickness means that the frusta has a higher peak load carrying capacity and hence mean load carrying capacity. The other noticeable trend that may be depicted from each of these graphs, is that for each response there is little difference in the shapes and magnitudes of the load deflection response as the rollslope angle is increased. This behaviour differed from the findings presented by Nagel (2004) for a numerical study conducted on the frusta alone, however, it must be noted that the velocity range that has been addressed within this study is within a very narrow lower band and may therefore receive only minimal influence from the effects of strain rate.
Figure 7-14 Mean Load Deflection Response 45° Rollslope

Figure 7-15 Mean Load Deflection Response 30° Rollslope
### 7.4.3 Velocity and Peak Deceleration Response

The velocity versus time response for each model has been plotted for each set rollslope angle in accordance Figure 7-17, Figure 7-18 and Figure 7-19. The behaviour shown by each graph is very similar and reveals an almost linear reduction in velocity of the rigid body as it is brought to rest by the structure.

![Figure 7-16 Mean Load Deflection Response 15° Rollslope](image1)

![Figure 7-17 Lateral Velocity versus Time Response 45° Rollslope](image2)
The deceleration versus time response for each model has been plotted for each set rollslope angle in accordance with Figure 7-20, Figure 7-21 and Figure 7-22. The response displayed by each model was similar with an overall trend indicating that an increase in frusta wall thickness resulted in an increase in the initial peak deceleration experienced by the system. It is evident from this graph that the
rollslope angle appears to have had only a minor influence on the severity of the initial peak deceleration experienced by the rigid impacting surface. This trend is further exemplified through reference to Figure 7-23 which plots the variation in initial peak deceleration with frusta wall thickness experienced by the combined assembly for each impact. This graph shows a fairly even relationship between peak deceleration and frusta wall thickness for increasing rollslope angle.

Figure 7-20 Lateral Deceleration vs Time Response 45° Rollslope

Figure 7-21 Lateral Deceleration vs Time Response 30° Rollslope
7.4.4 Effect of High Peak Decelerations on Vehicle Occupants

It is well understood that an impact event that takes place over a very short time period will result in the generation of a large resultant impact force. The generation
of this force arises from the need to change momentum \( mv \) during a collision. Examination of Newton’s second law which states that \( F = Ma \) or in more simplistic terms \( F = \frac{mv}{t} \), shows that the magnitude of a force generated during an impact is inversely proportional to the duration of the impact. Based on this equation, it is evident that a large contact time will result in the generation of small accelerations and subsequently small forces, whereas a small contact time will result in the generation of large accelerations and large forces. The transfer of large peak decelerations to the occupants of vehicles has been targeted by researchers such as Carney (1993) as one of the key sources for occupant injury sustained during vehicle impacts. Whilst little published information is available on peak deceleration transfer to occupants during heavy vehicle rollover impacts, it is believed that a reduction in such levels will result in the promotion of an increased level of occupant protection. The incorporation of a supplementary energy absorbing device in the form of a thin walled tapered frusta, has been found to substantially reduce the initial peak decelerations experienced by a ROPS during the rollover of a heavy vehicle. These findings suggest that the inclusion of such a device will be beneficial in providing an increased level of operator safety. Figure 7-24 shows the % reduction in the initial peak decelerations experienced by the rigid impacting surface for varying frusta wall thickness. The reader is made aware that these graphs have been plotted as percentage reduction of the decelerations experienced by the corresponding model without the frusta in place. It is evident from this graph that for a given ROPS stiffness, the frusta will reduce the peak decelerations substantially and that the magnitude of the reduction will be most favourable for thinner walled frusta. Based on these findings it is evident that the inclusion of such a device is beneficial and that a balance between maximising energy absorption and reducing peak decelerations may be an essential component for the design of energy absorbers suitable for use on ROPS.
7.4.5 Energy Absorption

The energy absorption versus time and kinetic energy dissipation versus time relationships were developed for each model and have been plotted in accordance with Figure 7-25, Figure 7-26 and Figure 7-27. The response shown by each graph for each rollslope inclination is similar with a higher magnitude of energy absorption taking place with an increase in rollslope inclination. This relationship is to be expected, as the kinetic energy increases due to the higher impact velocity associated with the steeper rollslope. For each model, the energy absorbed by the structure, is approximately a reverse mirror image of the input kinetic energy that was applied to the rigid impacting surface.
Figure 7-25 Energy versus Time Response of ROPS for 45° Rollslope Impact

Figure 7-26  Energy versus Time Response of ROPS for 30° Rollslope Impact
7.4.6 Energy Absorbed by Frusta

It is evident from the above mentioned graphs that the inclusion of a thin walled tubular supplementary energy absorption device has proven to be beneficial in reducing the amount of plastic deformation needed to absorb the kinetic energy of a rollover. This device for firm surface impacts, has also be found to reduce the severity of the peak decelerations transferred to the vehicle occupants by increasing the contact time between the ROPS and ground surface during an impact. In order to establish the optimum performance use of including such a device within a ROPS, the quantity of dynamic energy absorbed by the Frusta has been plotted as a percentage of the total dynamic energy absorbed by the complete ROPS/Frusta assembly for increasing rollslope inclination. The method used to determine the amount of energy absorbed by the Frusta was based on the principles of superposition. From inspection of the shape of the load deflection response for each model it was possible to determine the area beneath the load deflection response up until the point of engagement of the ROPS. From visual inspection of Figure 7-28, it is evident that a frusta with a thicker wall is capable of absorbing a greater proportion of the total dynamic energy. The other interesting relationship that is apparent from this figure, is that the energy absorbing capability of a frusta is greater for impacts that occur on lower gradient rollslopes. These findings suggest that the inclusion of a
supplementary energy absorber such as the Frusta will be more beneficial for lower velocity minor impact conditions.

![Frusta Wall Thickness vs Dynamic Energy Absorbed](image)

**Figure 7-28** Percentage of Dynamic Energy Absorbed versus Frusta Wall Thickness for Increasing Rollslope Angle

### 7.5 Chapter Summary

This chapter presented the results of a numerical investigation that investigated the feasibility of including a thin walled supplementary energy absorbing device into the framework of a typical two post rollover protective structure. The energy absorbing device chosen for examination was a thin walled rectangular tube known as a Frusta. A series of impact models were run using the explicit FE code LS-DYNA that involved adjusting the wall thickness of the frusta and studying the structural response for rollover impacts of varying inclination. The results of the investigation showed that a frusta was well suited to reducing the severity of the initial peak decelerations transferred to the occupants of the vehicle by increasing the contact time during the impact. The effectiveness of a Frusta to reduce such decelerations was found to be most pronounced for the frusta wall thickness of 2mm.

The other relationship that was discovered from the investigation was that a frusta was able to absorb as much as 25% of the input dynamic energy of the rollover. This amount of energy absorption was found to be greatest for the thicker walled frusta which had more area available for plastic deformation. In addition to this it was also
discovered that the effectiveness of a frusta as an energy absorber was most enhanced for impacts on lower gradient rollslopes. Overall, the inclusion of a supplementary energy absorbing device in the form of a thin-walled frusta was found to be a cost effective solution that promoted an enhanced level of occupant safety by reducing the amount of plastic deformation sustained by the ROPS during a rollover as well as reducing the severity of the peak decelerations transferred to the vehicle cabin during such an event.
Chapter 8 Summary and Conclusions

Rollover Protective Structures have to date been seen as the most viable method for providing protection to occupants during the rollover of heavy vehicles on construction and mining sites. These types of safety devices which are commonly fabricated from mild steel hollow sections rely predominantly on plastic deformation of their members in order to absorb some or all of the kinetic energy of the roll. For the case of earthmoving vehicles, the current performance standards both in Australia and worldwide employ simplified static testing methods to assess the capabilities of a ROPS. This form of ROPS certification is destructive and can be extremely expensive, particularly for the case of large vehicles. The use of analytical methods to assess the performance of ROPS has been prohibited by current performance standards primarily due to a lack of fundamental research information available on the nonlinear response of safety structures such as ROPS. In the light of these views, this project which has been a joint venture between QUT and the Industry Partner Robert Bird and Partners has aimed to rectify this situation by developing comprehensive research information that may be used to develop an analytical procedure for the assessment of ROPS.

In order to study the behaviour of ROPS a detailed experimental and analytical program was undertaken. The experimental program involved the use of static testing techniques to assess ROPS behaviour and to provide a base model for the establishment of an analytical method for ROPS evaluation. Finite element analysis formed the basis for the analytical program and involved subjecting established analytical models of the tested ROPS to both static and dynamic loadings. Implementation of this combined analytical and experimental approach enabled a calibrated analytical ROPS assessment procedure to be established that could lessen the need for expensive full scale testing. The developed procedure was able to replicate the five loading sequences required by the current Australian standard for earthmoving machinery protective structures AS2294.2-1997. Two procedures were initially evaluated the first of which was a load controlled method that was guided by a unit load time history function. This procedure was adequate for the evaluation of ROPS that underwent only minor plastic deformation during loading. The second
method which involved the use of an enforced displacement, avoided convergence problems and was found to be more stable for highly nonlinear problems and hence the most desirable method for analytical ROPS assessment.

The established displacement controlled solution method was applied to three different two post ROPS configurations. Two of the three models were also tested experimentally which enabled a calibration process to take place. For each model it was found that correlation between analytical and experimental results were excellent under the lateral loading phase which represents the most crucial loading condition of the standard. For the other two loading phases namely those in the vertical and longitudinal directions, it was found that the finite element analysis under-predicted the resulting displacements. Reasons for this lack of correlation between analytical and experimental results was attributed to the inability of the FE plasticity material model to handle the influence of load reversal effects such as the Baushinger effect as well as the possible inherent residual stresses incurred in the members from the heavy welding of the members during fabrication.

Further detailed finite element analysis of two posts ROPS subjected to static loads led to the development of simplified design guidelines that would enable ROPS designers to quickly proportion ROPS members to adequately meet the requirements of the standard. The term MP which represents the plastic moment capacity of a structural member was introduced. For a set global ROPS geometry, a range of plastic moment capacities were assessed by adjusting the section properties of each of the posts of the ROPS. For the two post ROPS configurations that were assessed, a simple collapse load model which assumed the formation of plastic hinges at the top and base of each post was able to be used to assess the maximum structural capacity of a ROPS. This method was also tested analytically and it was discovered that a carefully proportioned two post ROPS was able to adequately meet the performance requirements of the Australian standard provided that it possessed a collapse load that was equivalent to that of the minimum lateral loading provision of the standard. In addition to this it was also discovered that the energy absorption capability of a ROPS was an increasing function of the plastic moment capacity of its posts. In general for the stiffness configurations that were addressed, it was found that the
response of a ROPS to the applied vertical and longitudinal loading was approximately linear. Based on this study, it was recommended that two post ROPS be carefully proportioned to form plastic hinges at the top and base of each post for the lateral loading sequence whilst possessing sufficient capacity to withstand the subsequent vertical and longitudinal loading phases. In general it can be concluded that the simplified collapse load approach is an adequate design method for the assessment and proportioning of ROPS members provided that the estimated collapse load is at least equivalent to the minimum provision of the standard. It is advised that ROPS members do not be proportioned to possess collapse loads that are greater than 2 times this value as this may result in the chassis system of the vehicle being forced to absorb the energy which may also lead to mount failure and other adverse effects.

Dynamic loading was undertaken analytically in order to further understand the structural response behaviour of ROPS and to assess the adequacy of the provisions of the current Australian standard. Establishment of appropriate dynamic loads is difficult as the interaction between a rolling vehicle and the ground surface is a complex problem that is difficult to predict mathematically and consequently there is little published information in this area. In the absence of further information a procedure developed by Watson (1967) which involved a conservation of angular momentum approach was applied to the case of a two post ROPS attached to a rolling earthmoving vehicle. This procedure estimated the energy to be absorbed by a ROPS during a sideways vehicle overturn to be as much as 200% greater than the energy level required by the standard under static loading conditions. In considering this large differential between the two results it must be noted that Watson’s procedure is an approximate method that does not consider any energy absorption by other parts of the vehicle. Numerical analysis of the same ROPS models that were studied under static loads was performed using the explicit Finite element analysis package LS-DYNA. Numerical simulations were performed for varying rollslope angles and ROPS post stiffnesses. Results of the study suggested that dynamic amplification of the energy absorbed by the ROPS for a given deflection level exceeded the corresponding amount absorbed statically by as much as 30%. Reasons for this amplification were attributed to the strain rate sensitivity of the ROPS.
material, and the influence of inertia effects due to the input kinetic energy. These findings suggested that a much greater energy demand was placed on a ROPS during a dynamic rollover event as compared with the energy absorption criteria present within current ROPS performance standards. Further response parameters that were studied during each analysis were the relationships that existed between peak decelerations and ROPS post stiffness. As expected the peak decelerations were found to be an increasing function of ROPS post stiffness and were also more pronounced for steeper rollslope inclinations. This result suggested that a stiffer ROPS may therefore be less desirable for occupant protection as opposed to a more flexible one. In addition to this the contact time required to dissipate the ensuing rollover energy was also studied and it was concluded that a stiff ROPS had a shorter period of contact which resulted in a higher level of elastic rebound energy, whereas a more flexible ROPS possessed a longer contact time and was therefore able to dissipate the rollover energy more effectively with much less elastic rebound energy. This finding also promoted the use of more flexible rollover protective structures for use as energy absorbing safety devices. In addition to these findings, it was also discovered that for the range of ROPS stiffnesses examined, the high energy demands placed on the ROPS could successfully be absorbed without violation of the zone of the DLV for rollslope inclinations up to 30°. This finding suggested therefore that for the range of ROPS configurations and stiffnesses that were examined, the code procedure was adequate. From the analysis it was discovered that shorter durations pulse loads resulted in complete collapse of the ROPS structure and consequently resulted in large energy absorption demands being placed on the ROPS. The influence of the strain rate sensitivity of the ROPS material was also found to have only a minor influence on the energy absorption capability of the ROPS, with a general trend indicating that steeper angle impacts resulted in slightly higher levels of energy absorption demand being placed on the ROPS. Long duration pulse loads were found to place lower energy absorption demands on the ROPS by forcing the ground surface to absorb more of the rollover energy.

The enhancement of the energy absorbing capabilities of a ROPS was also studied through the inclusion of a supplementary energy absorbing device in the form of a thin walled tapered tube known as a frusta. A series of impact models were run using
the explicit FE code LS-DYNA that involved adjusting the wall thickness of the frusta and studying the structural response for rollover impacts of varying inclination. The results of the investigation showed that a frusta was well suited to reducing the severity of the initial peak decelerations transferred to the occupants of the vehicle by increasing the contact time during the impact. The effectiveness of a Frusta to reduce such decelerations was found to be most pronounced for the frusta wall thickness of 2mm. The other relationship that was discovered from the numerical investigation was that a frusta was able to absorb as much as 25% of the input dynamic energy of the rollover. This amount of energy absorption was found to be greatest for the thicker walled frusta primarily due to the greater area of material available for plastic deformation. In addition to this it was also discovered that the effectiveness of a frusta as an energy absorber was most enhanced for impacts on lower gradient rollslopes. Overall, the inclusion of a supplementary energy absorbing device in the form of a thin-walled frusta was found to be a cost effective solution that promoted an enhanced level of occupant safety by reducing the extent of plastic deformation sustained by the ROPS during a rollover as well as reducing the severity of the peak decelerations transferred to the vehicle cabin during such an event.

8.1 Implications of Research

Overall this thesis has demonstrated that analytical methods involving the use of finite element analysis may be used as a cost effective design tool to lessen the need for expensive destructive full scale testing. Some design guidelines that involve simple collapse load formulae have been provided to assist ROPS designers in proportioning member sizes to adequately meet the requirements of the standard. It is suggested that a carefully proportioned two post ROPS will have sufficient capacity to meet the requirements of the standard. It is recommended that a value possibly in the range of 1.5 to 2 times this requirement may be the correct answer to develop a solution that possesses an ideal balance between strength and stiffness. Further research will be required on this topic, however, in order to establish this ideal stiffness range as there are a number of controlling factors that are associated with the vehicle type and ROPS configuration. Dynamic analysis using explicit finite element analysis has highlighted that the energy absorbed by a ROPS will be amplified during the first impact of a sideways rollover.
whilst based on a simplified model have also shown that a much larger energy demand will be placed on a ROPS during a dynamic rollover event when compared to the energy absorption criteria present within the ROPS standard AS2294.2-1997. This finding may suggest that the standard is unconservative, however, as indicated by Klose (1969) it has been shown that other parts of the vehicle assist in absorbing energy during a rollover, which is a characteristic that may well have been accounted for in the development of the standard, hence leading to a lower quantum of energy absorption for ROPS certification. Finally the inclusion of a supplementary energy absorbing device within the framework of a ROPS has also been shown to provide beneficial response behaviour that will lessen the energy demand placed on the ROPS during a rollover as well as increasing the level of safety provided to the vehicle occupants.

8.2 Suggestions for Further Work

This project has examined the structural response of a number of two post rollover protective structures under both static and dynamic loading conditions. Extensive knowledge has been developed on the nonlinear response of safety structures which has led to the development of an analytical modelling procedure that may lessen the need for expensive full scale testing. Simplified design guidelines that will enable ROPS designers to adequately proportion the members of two post ROPS have also been developed and verified using finite element analysis. Whilst both the analytical and experimental phases of this project have been detailed some simplifying assumptions have been made and further work may need to be performed in order to verify some of the outstanding issues that exist in ROPS evaluation. Details of the suggestions for further work are outlined below.

Most of the work that was performed analytically in this thesis has assumed fixed based connections for the ROPS attachment to the vehicle chassis. This assumption was made in order to simplify the modelling procedures that were used for the numerical studies. It is suggested that further research be performed in this area as well as studying the influence of modelling the vehicle chassis and suspension system.
The major portion of this project has been concerned with the response behaviour of either two post rollbar type ROPS or cantilever type ROPS. It is suggested that further experimental and analytical studies take place on four post ROPS which are a common form of safety device used for rollover protection.

Whilst extensive efforts were made to establish the sources of the derivation for the empirical formulae present within AS2294.2-1997 and other similar performance standards, currently the exact origins and mathematical derivations remain unknown which makes clarification and verification procedures difficult. It is therefore suggested that further research be performed that will enable these formulae to be clarified.

It was discovered within this project that a ROPS that was carefully proportioned could adequately meet the requirements of the standard provided that it possessed a collapse load that was at least equal to the minimum lateral loading provision of the standard. It is suggested that a good energy absorbing structure will lie somewhere within the range of 1 to 2 times this minimum lateral loading level with optimum performance possibly around the 1.5 level. It is suggested that the design of overly stiff ROPS that possessed collapse loads greater than two times this level may be inappropriate and cause possible failure of the ROPS mounting systems as well as adverse dynamic effects on the vehicle occupants. It is suggested that further research be performed to verify a more acceptable and clearly defined collapse load range that will further assist ROPS designers.

Derivation of appropriate dynamic loads was difficult and consequently a simplified procedure was used. The dynamic impact modelling procedure using LS-DYNA was also simplified as it employed rigid bodies to transfer a predetermined amount of kinetic energy into the ROPS. Currently the computational requirements to perform a complete dynamic rollover simulation of an entire vehicle equipped with a ROPS are too extensive and therefore impractical. It is suggested that as the computational power
increases further research be performed in this area in order to develop a more accurate representation of a rollover condition.

The use of supplementary energy absorbing devices that can be incorporated into the frame of the ROPS is also an area that requires further investigation. This thesis has shown promising results for the use of thin walled tapered tubes as energy absorbing devices. It is recommended that the feasibility of other types of energy absorbing devices also be studied. It is also recommended that alternative to ROPS be further explored with the aim of promoting the highest level of safety possible to the vehicle occupants.
References


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Behaviour of Roll Over Protective Structures

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Citation

Heavy vehicles used in rural, mining and construction industries often operate on sloping terrain and have a high centre of gravity and hence are prone to accidental roll overs. These vehicle have a steel frame called a Roll Over Protective Structure (ROPS) attached over the cabin of the operator to protect him during such rolls. The role of this safety device is to absorb some of the kinetic energy of the rollover, whilst maintaining a survival zone for the operator. Design and evaluation techniques in the current Australian and international standards involve destructive testing and are not suitable for local conditions. Certification of ROPS by a more economical means using analytical modelling techniques is currently not permitted. This is due to lack of knowledge and research information on the behaviour of these structures in the post yield region and their energy absorption capacity.

Brian’s thesis addressed this issue using extensive computer simulation techniques supported by experimental testing. The research findings have enhanced our understanding on ROPS behaviour and will be used to improve energy absorption and safety and to facilitate the development of computer based techniques for design and evaluation that may lessen the need for destructive full scale testing.