Improving Skip Side Slipper Plate Design to Accommodate Higher Impact Bunton Force

by

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BScE, University of New Brunswick, 2015

A Thesis submitted in Partial Fulfillment of the Requirements for the Degree of

Master of Science in Engineering

In the Graduate Academic unit of Civil Engineering

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This thesis is accepted by the Dean of Graduate Studies

THE UNIVERSITY OF NEW BRUNSWICK

September, 2017

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Abstract

This research explored new ways to improve mine conveyance side slippers design, used to reduce the impact loads from the lateral movement resulting from misaligned mineshaft guiding system caused during the hoisting process. The research examined the dynamic behavior of conveyances using Comro design guideline (1990) and compared it against single degree of freedom analysis (SDOF). A soft material, rubber bearing pads, was added to the slipper design to reduce the total stiffness of the system, reducing the magnitude of the lateral impact force. Cotton duck pads, a type of bearing pad, were tested under strain rates of 0.001 to 200s⁻¹ to investigate their behavior under different strain rates and calculate a secant modulus of elasticity (300-400 MPa) to be used in the final slipper design. A new slipper was designed to increase the productivity of old and new mines. Design guidelines for the new slippers are presented in this thesis.
Acknowledgments

I hereby acknowledge and greatly appreciate the assistance of the following people and organizations:

- Dr. Alan Lloyd, for his guidance, advice and encouragement in supervising this work and proof-reading this document.

- Dr. Timo Tikka, for supervising the work and proof-reading this document.

- Natural Sciences and Engineering Research Council of Canada (NSERC), for funding the research.

- Stantec Consulting Ltd., for partially funding this research and providing the information required to conduct the research.

- Fabreeka International Inc., for providing the specimens needed to conduct the research.

- The Department of Civil Engineering, University of New Brunswick, for providing the necessary equipment for performing the tests.
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1 Chapter 1 Introduction to the problem

1.1 Introduction

Mining components are some of the most complex structures in the building field. Mineshafts are among the most important components of the infrastructure in deep mines as they are used for transporting people and materials to and from the mine core as well as lifting the raw materials to the surface. A steel system (steelwork) is used to divide the mineshaft area and provide guidance to the movement of conveyances between the surface and the mine core. This steelwork is comprised of horizontal buntons and vertical guides. The horizontal buntons are used to support the guides and transfer the loads of the hoisting process to the supports on the mineshaft walls. The guides are used to guide the vertical movement, while limiting the lateral movements, of the conveyances. This research focuses on the dynamic system where two guides guide the vertical movement (hoisting) of each conveyance. The guides are located on opposite sides of the conveyances. The steel guides and buntons are susceptible to multiple damaging mechanisms including fatigue damage, plastic deformation of conveyances and guide deraills. Impact forces on these components are expected to reach 2 to 3 times the gravity load of the conveyance in the form of horizontal impact (Krige 1983). In other words, the magnitude of impact forces can reach 2 to 3 times the weight of the loaded conveyances, which can be in excess of one hundred tons.

Problem statement:

The demand on mine conveyances is increasing as material is being transported to the surface in larger quantities than ever. Mines require higher mass conveyances moving
with higher hoisting speeds. Both contribute to potentially higher lateral impact forces (slam loads) from the conveyance onto the guide system. The magnitude of these slam loads needs to be reduced to mitigate potential damage to the guide rail system and allow continuous operation of mines.

1.2 Literature review

This section and Chapter 2 present an extensive background study of previous research investigating the topics of this thesis. It starts by explaining different mine shaft loads and the need for adding a rubber pad to reduce the impact load effect at the slipper location. Furthermore, it discusses a detailed study of elastomeric pad design as specified by the bridge AASHTO standards (2012).

1.2.1 Mining loads

A subsurface mine is generally divided into three components: Surface structures, a mineshaft, and a mine core. Surface structures organize the movement of conveyances hoisting people and materials from the mine core to the surface passing through the mineshaft. The mineshaft is the vertical tunnel connecting the surface to the mine core. The mineshaft is usually divided into multiple compartments. These compartments permit conveyances to move easily to transport people and materials from the mine to the surface. Each compartment contains at least two guides to guide a conveyance’s movement. These guides are supported by buntons anchored to the walls of the mineshaft. The buntons and guides are considered the steelwork of the mineshaft carrying the loads exercised by the hoisting process. The mine core is the active mine area where raw materials are extracted.
This research is only considering the loads exercised on the steel shaft system, guides and buntons, by the conveyances traveling between the surface and the core.

1.2.1.1 Permanent loads

South African National Standard, Design of structures for the mining industry SANS 10208-4 (2011) is a design code used in the mining industry worldwide. This design code divides the permanent load into:

- Self-weight of the steel work of the mineshaft.
- Rocks loads on the sides of a mineshaft supported by brow beam and sidewalls.
- Conveyor operating loads.

These loads are important in steelwork design but are not part of this research interest.

1.2.1.2 Dynamic loads

During the last few decades, researchers studied multiple mines to calculate the maximum bunton forces and deflections using load cells, strain gauges and computer programs (Krige 1986). The results of these investigations were compiled into a complete guideline to design mineshafts subjected to slam loads (Comro 1990). This guideline was also incorporated into the SANS 10208-4 (2011). In most cases of slam events, the rollers, located on the top of the skip, reduce the impact force exercised by the skip on the steelwork of the mine shaft. Over time, the rollers wear out and become inactive. In turn, the steel slippers mounted on both sides of the skip slam the steel guides directly without dissipating the force causing larger forces on the steelwork. These steel slippers are usually covered with a high-density polyethylene (HDPE) layer to reduce friction between
the slipper plate and the guide. Therefore, understanding the dynamic response of skip with inactive rollers is critical.

To calculate the magnitude of the forces on buntons and deflections of buntons, Comro (1990) transform the skip and the steel work system into a single degree of freedom system. This single degree of freedom system is formed using an effective mass of the conveyance computed about the impact location of the slipper at the leading end of the skip. The system also uses two springs in series representing the skip stiffness and the bunton stiffness. If the skip is very stiff, the total stiffness is considered to be only the bunton stiffness. Finally, the system has an initial velocity equal to impact velocity. This system will be discussed in detail in Chapter 2.

1.2.2 Elastomeric pads

To reduce the slamming force, a new slipper design could be developed using a combination of rubber type material and steel. The rubber material will reduce the effective stiffness of the system which in turn reduces the slamming force on the steel buntons. It will also play an important role for dissipating the dynamic behavior of the steel shaft. The hyper-elasticity of rubber is a useful property for damping dynamic loads. Rubber bearing pads have been used for multiple such applications including mitigating seismic loads and traffic loads on bridges (Roeder and Stanton 1983). Four types of rubber pads are specified in the American Association of State Highway and Transportation Officials (AASHTO): plain unreinforced elastomeric pads (PEP), Fiberglass-Reinforced Pad (FGR), Steel-Reinforced Elastomeric Bearing, and Cotton-Duck-Reinforced Pad (CDP).
To reduce the extreme impact loads, the new slipper design should provide a reasonable amount of load reduction, accommodate movement, and prevent premature fatigue failures or creep failures. As slippers are mounted on the sides of conveyances, the new design should have a rectangular shape to accommodate for the clearance between the guide and the slipper. Rubber bearing pads were found to be a realistic solution in this case for their capability in seismic base isolation and machine vibration control (Roeder and Stanton 1983) and their high allowable compressive capacity which can reach 20 MPa for fatigue loads (AASHTO 2012).

Roeder and Stanton (1983) provided a summary for different types of bearing pads and the gap between structural design and the manufacturers points of view of the elastomers bearing pads. They demonstrated two main types of rubber materials used in elastomeric pads: natural rubber or a synthetic rubber such as chloroprene. These rubber materials have a nonlinear stress-strain behavior affected by both temperature and strain rate. In general, the rubber pads stiffen under low temperatures and high strain rates.

1.2.2.1 Manufacturing of elastomeric pads

Raw rubber is a soft material with a very low stiffness. To resist higher stress with small deflection, the raw rubber should be reinforced by layers of reinforcements then vulcanized after bonding the reinforcement to the rubber (Roeder and Stanton 1983). This method secures a strong bond between the rubber and the reinforcement. During this procedure, some antiozonants, antioxidants, and fillers like oil and carbon black are added. These compounds affect the mechanical behavior of the rubber. The properties of the rubber can change from one manufacture to another, challenging structural engineers to design bearing pads (Roeder 2000). The structural engineer focuses mainly on the
mechanical properties of rubber pads including stiffness, strength and stress-strain relation of the bearing pad. On the other hand, the manufacturers’ focus is limited to achieve a certain range of hardness and elongation break (Roeder and Stanton 1983). Figure 1.2-1 shows an example of a rubber bearing pad.

CDPs are manufactured under the military specification MIL-C-882-E (1989). CDP consists of thin layers of elastomer and cotton duck fabric. Generally, CDP is stiffer and has higher compressive strength than other pad types (Roeder 2000).

![Elastomeric bearing pads](image)

**Figure 1.2-1** Elastomeric bearing pads (AASHTO 2012)

### 1.2.2.2 Different types of elastomeric pads

Elastomeric pads are generally classified by the type of the reinforcement used between the rubber layers. Four types of elastomeric pad specified in the AASHTO LRFD Design Specifications (2012) are; PEP, FGR, Steel-Reinforced Elastomeric Bearing, and CDP.

PEP is a plain rubber pad without any reinforcement layers. It is considered the least stiffened rubber pad in all the elastomeric pads. As the pad is formed completely
from rubber, the only force restraining the bulge of the pad is the friction between the rubber surface and the surface of the contacting material. This friction force depends on the type of material it is in contact with. The design of this type of pad is not permitted in some building codes and is penalized by others by reducing the effective mechanical properties used in design. (Roeder and Stanton 1983).

A steel reinforced bearing pad is formed of natural rubber or synthetic rubber with steel layers. The steel provides the pad increased in-plane stiffness by reducing the bulging area which reduces the total bulging of the pad (see Figure 1.2-2). The steel reinforcements restrain the pad’s lateral deflection, bulging, which increases compressive, shear, and rotation capacity (see Figure 1.2-3). In a reinforced pad, the shear stress in the pad transforms into a tensile stress in the reinforcement layers. As steel has high tensile capacity, few complications happened due to this tensile stress (Stanton and Roeder 1982).

Figure 1.2-2 Bearing Pads under compression: (a) PEP, (b) PEP with 2 surface in contact with a structural material, (c) PEP under compression, (d) reinforced pad under compression. (Stanton and Roeder 1982).
Fiberglass was used as reinforcing material in elastomeric pads. Fiberglass is much weaker than steel in tension which permit a better control of the strength of the bearing pad (Roeder and Stanton 1983).

Cotton duck pads (CDP) are produced in form of cotton duck material sheets, specified in the Military Specification MIL-C-882-E (1989), interlayered by layers of elastomer. Polytetrafluorethylene (PTFE) is used on the top of the pad as a sliding surface which accommodate for pad translation. CDP are known for their high stiffness relative to other bearing pad types. The hardness of this type of pad could reach 90 on a durometer shore A scale. High stiffness and hardness provide CDP with a higher compressive capacity and strength. Nevertheless, the thin layers of cotton duck deliver a higher rotational and shear stiffness which leads to a higher moments and edge capacities. These
higher rotational and shear stiffness also reduce the total shear and rotational deflections of the pads (Roeder 1999).

**1.2.2.3 Rubber pad properties and design criteria**

1.2.2.3.1 Hardness

Gent (1958) studied the relation between the hardness of natural rubber and modulus of elasticity. Gent derived a graph to describe this relation using the international rubber hardness (IRH) degrees and formulated in terms of deformation theory. This formula was compared with multiple experiments on natural rubber specimens (see Figure 1.2-4). The range of interest for the rubber hardness is between 50 and 70 Shore A hardness test.

![Figure 1.2-4](image1.png)

*Figure 1.2-4 Relation between the International Rubber Hardness (I.R.H.) and the elastic modulus of the natural rubber (Gent 1958).*
1.2.2.3.2 Compression

AASHTO (2012) limits the compressive stress of PEP to 0.8 ksi (5.5 MPa), FGP to 1 ksi (6.87 MPa), steel reinforced pad to 1.25 ksi (8.58 MPa) and CDP to 3 ksi (20.6 MPa). AASHTO describes the stress in PEP and FGP using the pad shape factor and the shear modulus. The shape factor of a pad is the ratio between the pad area and its perimeter area. Due to CDP manufacture procedures, the shape factor of CDP has a smaller effect on the behavior of the CDP compared to other pad types. Roeder (1999) compared the compressive behavior of CDP and the other pad types. CDP was found to have a higher stiffness and higher compressive strength. PEP had the lowest stiffness as it does not contain any reinforcements. Figure 1.2-5 shows the stress-strain curves for PEP, CDP, and steel reinforced elastomeric bearing pad.

As presented in Figure 1.2-5, the shape factor has a significant effect on both PEP and steel reinforced elastomeric pad. On the other hand, CDP experienced less effect on its compressive behavior compared to PEP and steel reinforced pads. In this research, the pad designed for the slipper plate has a high shape factor (5.4), as the pad’s area is very large, the same size as the slipper plate (200mm x 500mm), and the thickness of the pad is very small due to limited distance between the skip and the guide. The high shape factor increases the stiffness of PEP and steel reinforced bearing pads. Therefore, the research focused on testing CDP where the shape factor has smaller effect on the pad properties than PEP and steel reinforced pads. CDP was also selected due to higher compressive allowable stress (20.6 MPa) which will be essential while designing extreme impact cases.
1.2.2.3.3 Dynamic and fatigue

Lehman and Roeder (2005) presented the results of dynamic and fatigue compression tests for CDP samples. Their tests were designed for traffic loads. The test was conducted using 2 million cycles of stress levels equivalent to the heaviest truck loads with a loading rate of 1.5 Hz to avoid heat built up. This test evaluated the durability of the material for 50 to 100 years required for bridge life design. Two types of damage were observed: delamination of the cotton duck layer and oil secretion. The damage category
will be discussed in the following section. For samples tested to stress up to 7 MPa, the
damage was limited to oil secretion. For samples tested for maximum stresses of 14-21
MPa, the damage reached delamination of the cotton duck layers. The maximum stress
range had the most significant effect on the behavior and failure of the CDPs. To
maximize the pad durability, they recommended stress limit of 3 ksi (20.6 MPa).

1.2.2.4 Pads modes of failure

Lehman & Roeder (2005) categorized the test damage states of the CDP into 5
categories (see Figure 1.2-6):

- No damage (Figure 1.2-6(a))
- Secretion of oil or wax (Figure 1.2-6(b))
- Delamination of CDP Layer (Figure 1.2-6(c)).
- Fracture or cracking (Figure 1.2-6(d & e)).
- Internal split (Figure 1.2-6(f)).

Pads that were subjected to limited maximum strain or stress showed no damage. Oil
secretion from the pad occurred during long-term compression tests including the
dynamic compression tests with minimum stress of 7 MPa. For these first two categories,
CDPs were still functional. Pads, that were subjected to dynamic and uplift tests, showed
delamination of cotton duck layers. Diagonal failures were present due to high stress
compression or rotation tests. The last category, internal split, was observed in shear tests
with large shear deformations. Bearings that exhibited fractures or internal split damages
are required to be immediate replaced in the field.
Figure 1.2-6 CDPs failure states: (a) no damage, (b) oil secretion, (c) delamination, (d) diagonal failure due to compression, (e) diagonal failure due to rotation, and (f) internal slip (Lehman & Roeder 2005).

1.2.3 Conclusion

The previous CDP research focused on the compression and fatigue testing which describes the material behavior and compare it with other bearing pad types. This research will focus on the behavior of CDP under high impact loads. Examining this impact behavior will provide the information needed to design a flexible slipper to be mounted on the mine skip to reduce the dynamic behavior of the skip/guide system which will reduce the magnitude of impact load in the mineshaft.

1.3 Proposed research

The objectives of this study are as follows:
1. To assess the industry standard methodology of determining slam loads.

2. To determine an appropriate interface material that can be placed between mine skip slippers and rails to potentially reduce force transfer and mitigate slam loads.

3. To experimentally validate the static material properties of the interface material to ensure its appropriateness for use in the slipper design.

4. To experimentally determine the high strain rate behavior of the material to incorporate its dynamic material properties into the slipper design.

5. To simplify the material response into elastic equivalent single degree of freedom and equivalent static force procedure models for practical use in design of slippers.

6. To propose a simple design procedure for incorporating the interface material into slipper design.

The proposed research will consider a potash mineshaft located in Saskatchewan, Canada. The typical mineshaft configuration consists of two skips for material hoisting (A1, A2), a cage for people and equipment transportation to the mine location underground (C1), and a counter weight to balance the movement of the cage (B1) (see Figure 1.3-1).
An analysis of the mineshaft steelwork will be carried out to define the lateral loads developed in the steelwork as a result of the dynamic interaction between the conveyances and the conveyance guiding system. The dynamic interaction between conveyances and the shaft steelwork depends on multiple factors. These factors include the misalignment of the shaft guide, the bunton, skip, and guide stiffnesses, the mass of the skip, and the hoisting velocity. After analyzing the impact force, a bearing pad will be considered to reduce the impact force. This bearing pad will consist of a rubber-like material with low stiffness to change the dynamic response of the system and reduce the impact force at bunton location. In this mineshaft, a bearing pad is located originally on the slippers located on the sides of the conveyances (see Figure 1.3-2). An optimized design of the bearing pad will be explored with pad properties selected to reduce slam loads from the conveyance onto the rail system.
Unlike traditional bearing pads, the rubber, in this case, will not carry large gravity loads. The rubber will carry dynamic loads including cyclic and extreme rare impact loads. Cotton duck pads (CDP) will be used in this research at the slipper location. Due to the lack of information on the dynamic behavior and properties of CDP, CDP samples will be tested under static and dynamic loadings. This research will present a full design guideline for design bearing pads used to prevent rare and extreme impact loads in a mineshaft.

After the introductory information and literature review presented in this chapter, the thesis structure is as follows:

- Chapter 2 presents the dynamic properties of mine shaft systems and discusses dynamic analysis procedures.
• Chapter 3 describes the experimental procedure that was employed to test the static and dynamic material properties of the material added to the slipper to interface with mine shaft rails.

• Chapter 4 presents the experimental results of the procedure outlined in Chapter 3. The influence of different parameters is explored along with response of materials under different strain rates. Supplementary data from the experimental results is presented in Appendix C and Appendix D.

• Chapter 5 describes the proposed methods to incorporate the interface material into slipper design to reduce slam loads.

• Chapter 6 presents conclusions and suggestions for future research that stem from this work.

In addition to the above chapters, there are four appendices that present an example calculation of slam loads (Appendix A), the data acquisition system and experimental test monitoring process (Appendix B), and experimental data summaries from the static tests (Appendix C) and the dynamic tests (Appendix D).
2 Chapter 2 The dynamic behavior of mine shaft systems

This chapter illustrates an overview of the theoretical aspects of designing a shaft steelwork/skip system for dynamic loads. The chapter contains a summary of how the COMRO (1990) guideline have been completed though the last few years. The chapter is divided into multiple parts:

- Understanding the skip behavior in both active and inactive rollers scenarios.
- Comparing different commercial software used to analyze the dynamic behavior of the skip and mineshaft steelwork.
- Using Comro (1990) guideline to calculate the impact bunton force and maximum guide deflection.
- Developing a spreadsheet to calculate the maximum impact force and guide deflection of the approximated single degree of freedom system (SDOF) and compare its results to the guideline results.

After explaining the details of developing and understanding the dynamic behavior of the skip/steelwork system, the results will be used in designing the cotton duck pad (CDP) attached to the skip slippers. The new design takes into consideration the data gathered in the experimental results discussed in Chapter 4.

2.1 The skip dynamic behavior

This part describes the dynamic behavior of skips and shaft steelwork as described in Comro (1990). The skip-guide behavior is divided into two associated parts. The first is considering the rollers on the sides of the skip are fully effective. The second is
considering the rollers are not active and the impact occurs on the slippers mounted on both sides of the skip. In a real situation, both scenarios occur at the same time. The roller located at the top or the bottom of the skip start by deflecting until it is no longer active and the slipper on the side of the skip impact the guide causing the most harming effect on the mineshaft steelwork (see Figure 2.1-1). To simplify the analysis, Comro guideline provided some dynamic fundamentals to describe the theory behind the dynamic behavior of the skip/guide system.

![Diagram of roller and slipper mounted on the side of the skip](image)

**Figure 2.1-1:** Roller and slipper mounted on the side of the skip (COMRO 1990)

### 2.1.1 Roller-active configuration

Comro provided a study that can be used by readers to understand the dynamic behavior of skip/guide system in case of active-roller. This study is divided into five
systems starting from the simplest dynamic model and adding more complications in form of variables to reach a more realistic model.

2.1.1.1 System 1

This system considers the total mass of the skip as a dimensionless mass \( (m_s) \). This mass is traveling along a rigid guide without roller isolating the guide misalignment (see Figure 2.1-2). As the mass is translating with a constant velocity, the lateral acceleration is calculated using the following equation:

\[
\frac{d^2x}{dt^2} = \frac{d^2x}{dy^2} \left( \frac{dy}{dt} \right)^2 = d^2x \frac{v^2}{y^2}
\]

Equation 2-1

where \( x \) is the skip’s lateral displacement and \( y \) is the vertical displacement.

Figure 2.1-2: Skip model without rollers (COMRO 1990)

The force acting the guide, caused by the lateral movement of the skip is expressed as:
Force= mass* acceleration

\[ F = m \cdot v^2 \frac{d^2x}{dy^2} \]  
Equation 2-2

This equation illustrates that the acting force of the skip on the steelwork is quadratic related to the hoisting speed, the skip vertical speed. In other words, if the velocity of the skip increases, it will increase the impact force.

To evaluate the peak value of acceleration, a uniform cosine wave misalignment was selected to prevent the typical shaft peak misalignment (see Figure 2.1-3).

![Figure 2.1-3: Idealized guide misalignment (COMRO 1990)](image)

For a peak to peak amplitude of 30 mm, the maximum curvature is calculated as follows:
For hoisting speed equal to 15 m/s, the lateral acceleration of the skip could reach 0.5 g which result in a high impact force on the guide. Due to this high lateral force, rollers are mounted on sides of the skip. The following section will consider the rollers effect on system 1.

### 2.1.1.2 System 2

This system consists of skip mass, rigid misaligned guide, and roller represented in a linear elastic spring between the skip mass and the guide (see Figure 2-4).

![Diagram of System 2](image_url)

**Figure 2.1-4**: Skip model without rollers (COMRO 1990)

For this system, the guide misalignment profile is considered a bump in the form of a short duration misalignment with large amplitude to produce a harmonic vibration for the skip mass. The frequency of the system ($f_s$) could be calculated by:

\[
x = 15 \cos \frac{y \pi}{2500} = \frac{d^2x}{dy^2} = \frac{15\pi^2}{2500^2} \cos \frac{y \pi}{2500}
\]  

Equation 2-3
Figure 2.1-5 Guide bump and the harmonic response of the skip (COMRO 1990)

The system can reach resonance if the natural frequency of the system and excitation frequency are equal. When resonance occurs, the roller load increases without bound. To prevent this resonance situation, $\beta$ should be maintained less or more than one to reduce the dynamic response of the skip. The case of $\beta<1$ corresponds a small skip.
travelling slowly on stiff rollers over a guide with long misalignment waves. On the other hand, $\beta >> 1$ corresponds to a heavy skip on flexible rollers, high skip speed and short wavelength misalignment. Each of these approaches are considered in the design of shaft/steelwork, each has its own limitations.

Figure 2.1-6: Skip travelling over a bumpy guide and its response (COMRO 1990)

2.1.1.3 System 3

The mineshaft steel structure normally dissipates the vibration effect; this dissipation is called damping (see Figure 2.1-7). The amplitude of the harmonic response of the skip should decrease over time. The amplitude equation (Equation 2-6) could be transformed to the following equation:

$$A_{rm} = \frac{\beta^2}{\sqrt{(1-\beta^2)^2 + (2\beta\xi)^2}} \times A_{rm}$$  \hspace{1cm} \text{Equation 2-6}$$

Where $A_{rm}$ is the amplitude of relative motion, $\xi$ is damping ratio, the ratio between rollers damping properties and critical damping. Critical damping is defined as the situation
where the vibration damps out without crossing zero displacement. Critical damping ($C_c$) is calculated by:

$$C_c = 2\sqrt{k_r m_s}$$  \hspace{1cm} \text{Equation 2-7}

At resonance, the dynamic amplification factor is calculated by:

$$\text{Dynamic Amplification} = \frac{1}{2\xi}$$  \hspace{1cm} \text{Equation 2-8}

For a regular roller, the damping ratio ranges between 2% and 5%. The dynamic amplification factor is multiplied by the acceleration calculated in the system can reach up to 12.5g.

\begin{figure}[h]
\centering
\includegraphics[width=0.5\textwidth]{figure.png}
\caption{Skip with damping going over a bump in a guide (COMRO 1990)}
\end{figure}

In fact, this enormous load does not happen in real situations. In systems 1, 2, and 3, the skip is modeled as a concentrated mass with a roller on one guide. These assumptions could be improved by assuming a skip length and two active rollers.
2.1.1.4 System 4

In this system, the length of the skip and rollers on both top and bottom will be taken into consideration. The system is considered to be two degrees of freedom system. The two degrees are then divided into two single degrees of freedom (SDOF) systems (see Figure 2.1-8). The first system is similar to system 3 model describing the lateral movement of the skip. The second describes the rotational behavior of the skip.

![Skip model with two rollers and the two single-degree of freedom representing it (COMRO 1990).](image)

**Figure 2.1-8:** Skip model with two rollers and the two single-degree of freedom representing it (COMRO 1990).

The two SDOF systems could happen separately. The translational behavior happens when the guide misalignment wavelength is equal to the skip height. The second is the rotational behavior of the skip which happens when the skip height is equal to one and half times the wavelength of guide misalignment. The effect translational misalignment could be calculated as the average of the misalignment at the two rollers.
Also, the rotational misalignment is the ratio between the difference in guide misalignment at the two rollers and the skip height as shown in Figure 2.1-9.

The rotational degree of freedom could be calculated using the following terms:

The mass moment of inertia about the center of gravity (c.g.):

\[ I = \frac{m_s H^2}{12} \]  
Equation 2-9

This formula assumes the mass is uniformly distributed along its length \( H \), which is much longer than the width of the skip. The rotational stiffness when 2 rollers are in contact with guide:

Figure 2.1-9: Effective misalignment for a general skip and guide configuration (COMRO 1990).
\[ k_\theta = 2 k_r \left( \frac{H}{2} \right)^2 = \frac{1}{2} k_r H^2 \]  

Equation 2-10

This formula assumes that the rollers are symmetrically placed with respect to the skip c.g. If the c.g. of the skip is not symmetrically placed, it is advised using a computer model to calculate the mass moment of the inertia and the rotational stiffness.

The guide misalignment profile is not usually in a waveform. This profile could be solved by using superposition of multiple waves (see Figure 2.1-10). The skip’s dynamic response for each of these waves could be solved separately and the results will be combined to form the expected dynamic response of the skip. This system is used in guidelines are based for the rollers-active case.

Figure 2.1-10: Example for dividing a random guide profile into multiple periodic waves (COMRO 1990).
Computer programs could be used to evaluate the dynamic response of the skip for each waveform and combine them to get the overall dynamic response of the skip.

Up to system 4, the dynamic behavior discussed was composed of a skip, a guide, and maximum of two rollers. The following system will discuss the dynamic behavior of the skip while considering two guides, one on each side of the skip.

2.1.1.5 System 5

This system contains the description of the skip system with two guides, one guide on each side profile and how this profile affects the behavior of the skip. Figure 2.1-11 shows two misaligned guides profile which could be divided into parallel misaligned guides and a gage profile.

![Diagram showing general misalignment of two guides profile](image)

*Figure 2.1-11: General misaligned two guides profile and the equivalent parallel and gage profiles (COMRO 1990)*

The gage variation was found not affecting the vibration of the system along with the analysis procedures of the dynamic behavior of the guide/skip system. On the other hand, the gage profile will affect the static force of the skip roller directly. This change of
roller force should be taken into consideration while designing the steelwork of the mineshaft.

The presence of a second guide will not affect the analytical complexity of the system. It is safe to assume that minimum of two rollers will be active with a preload on the guides. But, for design purposes, it is advised performing the analysis twice: once with only two active rollers and the second has four active rollers.

To develop a plot describing the peak dynamic response for the shaft, it is necessary to perform many dynamic simulations, each with a different skip natural frequency and damping values. This plot could be used to determine the roller force related to the unit mass of the skip. Figure 2.1-12 shows an example of the roller force shock spectrum at President Steyn Gold Mine, Number 4 Shaft.

![Chart showing roller force shock spectrum](chart.png)
This spectrum is commonly used to present the dynamic response of multiple shock and vibration in different industries.

As a conclusion, the roller’s low stiffness is considered the main reason for reducing the guide misalignment effect on the skip bouncing behavior. In the case of rollers failure or excess deflection, the slippers on both sides of the skip impact directly on the guides. Overall, to reduce the roller force, the designer could use, as discussed in system 2, one of two scenarios a small skip traveling slowly on stiff rollers over a guide with long misalignment waves or a heavy skip on flexible rollers, high skip speed, and short wavelength misalignment. Scenario 2 is more likely the situation in the mining operations.

### 2.1.2 Inactive-rollers configuration

In this model, the rollers mounted on the skip are considered not active. The slipper on the side of the skip impacts the mineshaft’s steelwork making the most harming effect on the steel guides. Figure 2.1-13 shows a sequence of events during severe slamming in a mine shaft. This diagram was developed using information from multiple experiments in which bunton forces were measured during the mining operation.
In Figure 2.1-13, The most harming slamming force could be described as follows: Initially, the skip contacts guide E closer to midspan (point A). Due to the low stiffness of the guide and high inertia of the skip, the skip’s top corner start to bend guide E. Reaching the bunton (point B), the skip starts to accelerate and bounce back hitting guide F. The maximum skip lateral velocity is reached at point B just before the top of the skip reaches the bunton. Depending on the specifications of the skip and steel work, the maximum bunton force can be either at bunton C or C’. These slamming loads could cause sudden failure to steelwork components, buntons and/or steel guides. To reduce these slam loads, the design guide focuses on two main aspects. The first is the skip roller system performance which is considered the first line of defense against these loads. The second is the steel work, guides, and buntons, which would carry the dynamic loads by
hitting the steel slippers located on both sides of the skip, in the case of roller wearing failure. A computer program called SLAM was created to calculate the slamming event as described above.

In general, the roller lower stiffness is considered the main reason for reducing the guide misalignment effect on the skip bouncing behavior. In the case of rollers tire out, the slippers on both sides of the skip impact directly on the guides. Overall, to reduce the lateral impact force, the designer could use flexible buntons and stiff guides. On the other hand, using stiff buntons and flexible guides will result in accommodating bigger slamming forces (Comro 1990).

Due to the complexity of the dynamic analysis, different researchers created computer programs to calculate the roller loads or slipper load. Each of the computer programs was checked by on-site tests using load cells and accelerometers mounted on the skip. In the following section, three programs will be discussed and compared as considered in Krige (1983).

2.2 Computer models.

As discussed previously, it is advised to model active and inactive rollers scenarios separately. The following three computer models were created to calculate the maximum roller force or the slamming force, created by the impact of the slipper on the guides. The first two models, DISC and SKIP II, discussed the active roller situation and SLAM studied the inactive-roller situation.
2.2.1 DISC

Dynamic behavior of the conveyance has the most damaging effect on the steelwork of the mineshaft. To predict the maximum roller force the guide, Krige (1983) developed a computer program (DISCs) and validated it with on-site measurements. Multiple variables were found to be crucial for the dynamic behavior of the conveyance:

2. Plumb and gauge misalignments
4. Hoisting speed.
5. Conveyance mass, height, the center of gravity height and stiffness at slippers.
7. Bunton spacing and stiffness.

Due to the high hoisting speed and high weight of the skip, guide’s misalignment occurs regularly. The misalignment of steel guide causes a bouncing behavior of the conveyance between the two steel guides located on both sides of the skip. There are three main reasons for guide misalignment:

1. Guide mismatch due to the difference of guides depths.
2. Off- vertical distance between two buntons due to the alignment of the guide according to a vertical surveyor line.

To created the equation of motion for the skip behavior, Krige created models for different parts of the system as follows:
2.2.1.1 Guide modeling

To model the guide, Krige (1983) used a semi-continuous elastically supported beam (see Figure 2.2-1). The guide stiffness is then measured using structural analysis program or equation 2-11. Combining the guide stiffness and the roller stiffness will result in calculating the effective spring stiffness to approximate the model into a SDOF model. The guide stiffness changes episodically while the roller moves from bunton to bunton. The following equation present the guide stiffness at any location in the center span of the guide, L/3 ≤ x ≤ 2L/3, calculation:

\[
K_g = \frac{b_3}{[(b_1 - b_2) + (38b_1 + 22b_2) a + (54 - 479b_1 - 175b_2) a^2}
\]

\[
- (108 - 1044b_1 - 630b_2) a^3 + (54 - 927b_1 - 1125b_2) a^4
\]

\[
+ (486b_1 + 972b_2) a^5 - (162b_1 + 324b_2) a^6]\]

Equation 2-11

where:  \(a = x/L, \ b_3 = 162EI/L^3, \ b_1 = 1/(b_3/K_b + 5), \ b_2 = b_1 \left(\frac{7/3}{b_3/K_b + 1/3}\right),\) and \(K_b\) is bunton stiffness.

The derivation of this equation was found in Krige (1983). By combining the guide stiffness and the wheel stiffness, the effective stiffness is calculated:

\[
K_e = \frac{K_g K_W}{K_g + K_W}
\]

Equation 2-12

This equation is built on the assumption that both stiffness are linear.
In the case of slipper contact, the stiffness of the roller will no longer be in active and the effective stiffness will be only the guide stiffness. This situation is called hard contact. The load-displacement is no longer linear (see Figure 2.2-2) but, linearity could be maintained by adjusting the relative guide location from the moving skip (w_{xx}). Therefore, the guide location is essential in changing the effective stiffness and modeling the guide.

As previously discussed, the guide misalignment is the main reason for the dynamic behavior. The reasons of guide misalignment could be divided into three parts:
• If the guides were joined based on aligning the back of the guide, the guide steel could be mismatched for manufacturing reasons.

• The guide is aligned according to a vertical surveying line with an off-vertical distance. This distance has some tolerance limits but can vary which affect the misalignment of the guide.

• Gage errors are the last reason of misalignment as discussed in system 5 in the previous section.

The last two reasons are uniformly distributed over the length of the guide between the two buntons.

Firstly, the skip wheels are located clear from the guides or with a preload on the guide. Then, the wheel starts to deflect the guide. The wheel load is calculated using the effective stiffness. Furthermore, hard contact occurs, the wheel is no longer active, and the slipper carries the slam load. Lastly, wheel irregularity plays a very important role to calculate the guide location (see Figure 2.2-3).
2.2.1.2 Conveyances modeling

The conveyance was modeled in one plane between the two guides located on both sides of the conveyance and divided into three rigid bodies. These three bodies are connected by the bridles, which has sway stiffness and damping properties. Each rigid body has its transitional degree of freedom and the same rotational degree of freedom. All vertical movements are not considered in the analysis. As a result, the model consists of four degrees of freedom, three transitional and one rotational (see Figure 2.2-4).

Figure 2.2-3: Effective guide Position (Kringe 1983)
Figure 2.2-4: DISC Conveyance model (Krige 1983).
For each degree of freedom, an equation of motion could be presented in its simplified form, neglecting the damping effect, as follows:

\[
M_1 \ddot{v}_1 + A_5 v_1 - K_{c3} v_2 - K_{c4} v_3 + A_6 \phi = 0
\]

\[
M_2 \ddot{v}_2 - K_{c3} v_1 + A_7 v_2 + 0 + A_8 \phi = 0
\]

\[
M_3 \ddot{v}_3 - K_{c4} v_1 + 0 + A_9 v_2 + A_9 \phi = 0
\]

\[
I_r \ddot{\phi} - A_1 v_1 + A_2 v_2 + A_3 v_2 + A_4 \phi = 0
\]\[Equation 2-13\]

where:  
A_1 = K_{c3} \frac{d_1}{2} + K_{c3} d_3 - K_{c4} d_4 - K_{c4} \frac{d_6}{2}

A_2 = K_{c1} \frac{d_1}{2} + K_{c2} \frac{d_1}{2} - K_{c3} \frac{d_4}{2} - K_{c3} d_3

A_3 = K_{c4} d_4 + K_{c4} \frac{d_6}{2} - K_{c6} \frac{d_6}{2} - K_{c6} d_6

A_4 = K_{c3} \frac{d_1}{2} d_3 + K_{c3} d_3^2 + K_{c4} d_4^2 + K_{c4} \frac{d_6}{2} d_4

A_5 = K_{c3} + K_{c4}

A_6 = K_{c3} d_3 - K_{c4} d_4

A_7 = K_{c1} + K_{c2} + K_{c3}

A_8 = K_{c3} d_3

A_9 = K_{c4} + K_{c5} + K_{c6}

A_{10} = K_{c4} d_4

K_e is the effective stiffness of the wheels and guides, K_c is the stiffness of the sway connections, d is the height of each mass or the distance from the c.g. of the middle mass, as shown in Figure 2.2-4, M is the mass of each skip part, I_i is the moment of inertia of the mass of each rigid body, v_i is the displacement degree of freedom of mass i, and \( \phi \) is the rotational degree of freedom of the total mass.
Using a step-by-step numerical method, the previous equations is used to solve for the wheel load and the natural frequency of the conveyance system.

2.2.2 Skip II

SKIP II is a computer model set up by Colorado School of mines, as described in Krige (1983). The program set up is similar to DISC program, but with some differences. These differences are listed as follows:

1. The guides are modeled as a simply supported between two rigid buntons. Comparing with DISC program, the guide stiffness is a function of the buntons stiffness.

2. The conveyance is modeled as a three-dimension rigid body. Which is different from DISC program as three connected masses in the plane of the two guides.

3. It also does not take into consideration the guide mismatch or wheel irregularities.

4. SKIP II also used a step-by-step numerical method.

By comparing the results of these two models, despite the different assumptions used, the results of both computer programs were similar for maximum bunton force and maximum guide displacement. The guide inertia was the same for both models. The naturals frequencies were different as SKIP II simply calculated the inverse of the period that the conveyance takes to travel between the two buntons. The wheel loads were comparable in both programs. Figure 2.2-5 presents the results of four different cases using both programs. The similarity in wheel loads produced by each of the programs indicates that the assumptions used in both programs are reasonable.
Figure 2.2-5: Comparing the results from DISC and SKIP II (Krige 1983).

2.2.3 SLAM

As discussed in the inactive-roller configuration earlier in this chapter, SLAM computer program was created to calculate the extreme slam load created by the slippers
located on the side of the skip. Detailed information about the program was found in COMRO (1990) mineshaft steelwork design guideline. The program integrates the equation of motion at a small time step to evaluate the forces on the steelwork during impact. Figure 2.2-6 presents five different slamming events for the same skip-guide configuration.

Each of the curves presented in the figure is associated with a different initial contact location on the guide. The first curve the initial contact occurred at 20% of the distance between the two buntons, the second is at 40% etc. The most critical event occurred when the skip was closest to the bunton. The last curve shows the impact at the bunton which is not as severe as just before the bunton event (80%).

Figure 2.2-6: Slam force diagram on different locations of the skip slipper on the steel guide (COMRO 1990).
To build the equation of motion, SLAM used an approximation method while reducing the skip-guide system to a single degree of freedom (SDOF) system. Some assumptions were made based on previously completed researched systems as discussed to this point in the chapter. To create this equivalent SDOF system, multiple parameters needed to be calculated such as effective mass, effective stiffness and effective applied force. This phenomenon will be explained in the design procedures of the mineshaft steelwork provided by the COMRO guideline (1990).

2.3 Design procedures

To facilitate the design procedure, shafts were categorized according to the frequency of mine maintenance. The maintenance includes realignment of guides and adjusting the rollers. The shafts are categorized as shown in Table 2.3-1.

**Table 2.3-1**: Shaft Category, Design and Performance Parameters (Comro1990 & SANS 10208-4).

<table>
<thead>
<tr>
<th>Maintenance Category:</th>
<th>With Rollers</th>
<th>Without Rollers</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>A Good</td>
<td>B Average</td>
</tr>
<tr>
<td>Maximum guide misalignment at bunton (ε)</td>
<td>5 mm</td>
<td>10 mm</td>
</tr>
<tr>
<td>Maximum guide gauge variation (Δa)</td>
<td>±3 mm</td>
<td>±5 mm</td>
</tr>
<tr>
<td>Decelerometer data</td>
<td>&lt; 5 m/s²</td>
<td>5-10 m/s²</td>
</tr>
<tr>
<td>Frequency of realignment</td>
<td>3 months</td>
<td>6 months</td>
</tr>
<tr>
<td>Slamming event frequency</td>
<td>5%</td>
<td>10%</td>
</tr>
</tbody>
</table>
In Table 2.3-1, shafts are divided into four different categories. Each category demonstrates the maintenance, realignment frequency and their effect on guide misalignment and slamming data. Comro suggests that designers should consider using a more conservative category while calculating the slamming load. The example provided in Appendix A use category C, with a maximum guide misalignment 20mm and gauge 10mm, resulting in the maximum expected slamming load for the mineshaft design.

2.3.1 Shaft steelwork design

After deciding the category of the shaft, the designer should design the shaft steelwork while taking into consideration that the shaft could:

- Minimize the probability of slam events.
- Does not fail due to fatigue or under extreme dynamic loads.
- Ensure that the skip would be guided at all times.

The following guideline provided by Comro (1990) describes the calculation procedures to ensure the design of a safe steelwork satisfying the previous criteria.

2.3.1.1 Steelwork stiffness

To predict the maximum bunton force and deflection, multiple parameters needed to be determined in the calculations related to every shaft steelwork and skip. Firstly, the steelwork stiffness, the effective stiffness, is essential for the calculation of the dynamic response of the system. The steelwork stiffness could be divided into two: stiffness at the guide midspan \( k_g \) and guide to bunton connection stiffness \( k_b \). Computer calculation is recommended to analyze the steelwork stiffness.

In the case of flexible skip, where the stiffness of skip is relatively small, the stiffness of the skip \( k_s \) could be used to estimate a new effective bunton stiffness. The
stiffness should be calculated at the slipper plate mount and in the plane of the slamming load. This effective bunton stiffness could be calculated by putting bunton stiffness $k_b$ and skip stiffness $k_s$ in series (Equation 2-14). On the other hand, if the skip stiffness is very high, the bunton stiffness will be used in the calculations and skip stiffness will not be considered in the effective stiffness calculation (see the analysis section of this chapter).

$$k_{be} = \frac{k_b k_s}{(k_b + k_s)} \quad \text{Equation 2-14}$$

As discussed in the inactive-roller configuration, it is clear in the previous formula that the guide stiffness is not considered in the effective stiffness. The guide stiffness would be considered in calculating the impact and the rebound velocities.

### 2.3.1.2 Effective skip mass

During slamming events, the skip’s inertia force act on the guide at the slipper location on the leading corner of the skip. There would be two degrees of freedom associated with the skip system. The first degree of freedom is the transitional degree of freedom acting on the center of gravity of the skip. The second degree of freedom is the rotation about the center of gravity. This behavior is similar to system 4.

The new effective skip mass is calculated as follows:

$$m_e = \frac{m_s I}{(I + m_s h^2)} \quad \text{Equation 2-15}$$

where: $m_e$ is effective skip mass during impact, $m_s$ is total skip mass, $I$ is the mass moment of inertia of the skip through the skip c.g. and about an axis normal to the plane between the guides, and $h$ is the vertical distance between the skip c.g. and the corner at
which \( m_e \) is effective or the distance between skip c.g. and the slipper located at the skip corner (see Figure 2.3-1).

![Figure 2.3-1 The Geometry of the mine skip and the location of the effective mass (COMRO 1990)](image)

2.4 Analysis

To facilitate the skip behavior analysis, COMRO (1990) provided a list of parameters that could be defined as follows:

The ratio between the bunton and the midspan guide stiffness (\( r_k \)) is used to facilitate the slam load calculation. This ratio is always bigger than 1 as the bunton is much stiffer than the guide.

\[
r_k = \frac{k_b}{k_g}
\]

Equation 2-16

In case of flexible skip, the bunton effective stiffness could be used instead of the bunton stiffness.
The maximum impact velocity (u) is calculated as shown in the following equation:

\[ u = \frac{2ev}{L} \]  

Equation 2-17

Where L is the distance between two buntons, v is the skip hoisting speed, and e is the maximum guide misalignment depending on the maintenance frequency (see Table 2.3-1). A ratio of rebound velocity to the impact velocity u could be also represented as \( r_u \).

To generalize the impact calculation, multiple non-dimensional factors could be introduced:

- \( \overline{k_b} = \) nondimensional steelwork stiffness at bunton
  
  \[ \overline{k_b} = \frac{k_{be}L^2}{m_ev^2} \]  
  
  Equation 2-18

- \( \overline{P} = \) nondimensional bunton force
  
  \[ \overline{P} = \frac{PL}{m_ev^2} \]  
  
  Equation 2-19

where: P is the bunton impact force.

- \( \overline{\delta_s} = \) nondimensional skip displacement
  
  \[ \overline{\delta_s} = \frac{\delta_s}{L} \]  
  
  Equation 2-20

where: \( \delta_s \) is the skip displacement relative to steelwork in meter.

- \( \overline{M} = \) nondimensional guide bending moment
\[ \bar{M} = \frac{M}{m_e v^2} \]  

Equation 2-21

where: \( M \) is the guide bending moment in N.m.

Knowing the steelwork stiffness ratio (\( r_k \)) and the non-dimensional stiffness of the steelwork bunton (\( \bar{k}_b \)), Figures A.2-1 to A.2-4 (see Appendix A) could be used to determine all relevant steelwork parameters. Note these figures are only for \( u/v = 0.005 \). For different \( u/v \), a correction factor should be applied to \( P, \delta_s \) and \( M = 2e/0.005L \). The example, provided in Appendix A, illustrates the procedure for designing the steelwork of a mine shaft without taking into consideration the roller and fatigue designs.

The slippers should also be designed for fatigue loadings. The designer should consider the fatigue loads based on the number of impact events. According to the Commentary on SABS 0208:4 (2001), the slipper plate impact loads occur 10% times of the hoisting cycles. In an average mineshaft, the impact event could occur 1,000 times per year.

2.5 Dynamic response using Newmark \( \beta \) method

Up to this point of this chapter, an extensive understanding for the dynamic behavior of the skip/steelwork system was presented. Also, the Comro (1990) guideline was used to evaluate the impact force in a skip system after approximating the system to a SDOF system with an effective mass, effective stiffness and an initial impact velocity. In this portion of the chapter, three different numerical methods will be presented to evaluate the dynamic behavior of the SDOF system as presented by Biggs (1964). One of these
numerical methods will be used to check the analysed example in Appendix A with Comro guideline.

2.5.1 Building the SDOF model

The single degree of freedom system, as presented in Figure 2.5-1, consists of an effective mass of the skip, effective stiffness of the bunton and skip, and initial velocity which will be considered as the impact velocity of the skip. For this system, the applied force has a very small time interval, thus, the force could be replaced by an impact velocity. The equation of motion could be presented as follows:

\[ m_e \ddot{y} + k_e y = f(t) \]  

Equation 2-22

where \( y \) is the mass displacement and \( \dot{y} \) is the mass acceleration.

![Diagram of SDOF model](image)

Figure 2.5-1: Single degree of freedom set up

The damping effect of the system is not considered as it does not affect the initial impulse of the system vibration. The applied force is considered equal to zero and replaced by the skip impact velocity \( u \) which could be calculated using Equation 2-16. The spring force \( k_e y \) is assumed to have a linear spring constant equal to the effective stiffness calculated using Equation 2-14. The effective mass could be calculated using Equation 2-15. The equation of motion as presented in Equation 2-21 could be solved using step by
step to determine the variation of displacement and the change in acceleration. Biggs (1964) presented three different methods to solve the equation of motion:

- Constant velocity or lumped-impulse.
- Linear acceleration method.
- Newmark β method.

The equation of motion could be solved step by step method starting at zero time when the initial velocity and displacement are known. The time scale is divided into small intervals which simplify the calculation of displacement from one interval to another.

### 2.5.2 Constant velocity or lumped-impulse

This analysis considers a constant acceleration between two time steps. Figure 2.5-2 presents the analysis of the displacement-time variation for a single degree of freedom.

![Figure 2.5-2: Numerical Integration using constant velocity method (Biggs 1964)](image)

The displacement $y^{(s)}$ is the mass displacement at time $s$ and $y^{(s-1)}$ is the displacement at time $s-1$ which were previously determined. The acceleration $\ddot{y}^{(s)}$ at time
s could calculated using the equation of motion. The displacement $y^{(s+1)}$ could be
determined using the following equation:

$$y^{(s+1)} = y^{(s)} + \dot{y}_{av} \Delta t$$

Equation 2-23

where $\dot{y}_{av}$ is the average velocity between time stations $s$ and $s+1$, and $\Delta t$ is the time steps
between the stations. $\dot{y}_{av}$ could be determined by:

$$\dot{y}_{av} = \frac{y^{(s)} - y^{(s-1)}}{\Delta t} + \ddot{y}^{(s)} \Delta t$$

Equation 2-24

The average velocity could be presented as the average between the velocity at
time $s-1$ and $s$ added to the increase in the velocity exercised by the acceleration of the
system in the time interval while assuming $\ddot{y}^{(s)}$ is the average acceleration over the
interval. Using these approximation, the curves into straight lines facilitate the calculation
of the displacement on the next time station. Substituting Equation 2-24 in 2-23:

$$y^{(s+1)} = 2y^{(s)} - y^{(s-1)} + \ddot{y}^{(s)} (\Delta t)^2$$

Equation 2-25

Although, Equation 2-25 is approximated, but it could present an accurate solution
by minimizing the time step ($\Delta t \rightarrow 0$). The solution in this situation approaches the exact
solution, however, the number of computations will increase. In most cases, to find an
accurate solution with less computation, the time step could be assumed equal to the one-
tenth of the natural period of the system.

In some cases, $y^{(s-1)}$ has no value as the analysis could start at $t = 0$ where
different approach should take place to progress with the analysis. Two different
procedures could be used. First, the acceleration could be assumed to be linear on the
first time interval. The displacement could be calculated:
\[ y^{(1)} = \frac{1}{6} \left( 2\dot{y}^{(0)} + \ddot{y}^{(1)} \right) (\Delta t)^2 \quad \text{Equation 2-26} \]

Second, the acceleration could be assumed constant:

\[ y^{(1)} = \frac{1}{2} \dot{y}^{(0)} (\Delta t)^2 \quad \text{Equation 2-27} \]

However, Equation 2-27 should be used if the dynamic problem does not have an external force. Therefore, the acceleration will be equal to zero. In case of problem instability or convergence, more accurate methods could be used. The following numerical methods satisfy greater precision for the same time step used in the constant velocity method.

### 2.5.3 Linear-acceleration method

This method is a small adjustment for the previous constant-velocity method by assuming a linear acceleration function (see Figure 2.5-3). If the acceleration is assumed to be linear between two time steps, the acceleration could be presented by:

\[ \ddot{y} = \ddot{y}^{(s)} + \frac{\ddot{y}^{(s+1)} - \ddot{y}^{(s)}}{\Delta t} (t - t^{(s)}) \quad \text{Equation 2-28} \]

The velocity at the same time interval:

\[ \dot{y} = \dot{y}^{(s)} + \dot{y}^{(s)} (t - t^{(s)}) + \frac{\dot{y}^{(s+1)} - \dot{y}^{(s)}}{2\Delta t} (t - t^{(s)}) \quad \text{Equation 2-29} \]

The velocity at \( s+1 \):

\[ \dot{y}^{(s+1)} = \dot{y}^{(s)} + \frac{\Delta t}{2} \left( \ddot{y}^{(s+1)} + \ddot{y}^{(s)} \right) \quad \text{Equation 2-30} \]

The displacement at \( s+1 \):
\[ y^{(s+1)} = y^{(s)} + \dot{y}^{(s)} \Delta t + \frac{(\Delta t)^2}{2} (2\ddot{y}^{(s)} + \ddot{y}^{(s+1)}) \]  

Equation 2-31

The velocity at s as presented in Equation 2-30:

\[ \dot{y}^{(s)} = \dot{y}^{(s-1)} + \frac{\Delta t}{2} (\ddot{y}^{(s)} + \ddot{y}^{(s-1)}) \]  

Equation 2-32

Equation 2-31 and 2-32 are the basis for the linear acceleration numerical method.

![Diagram](https://example.com/diagram.png)

**Figure 2.5-3** Linear-acceleration approximation (Biggs 1964)

As presented in Equation 2-31, in order to obtain \( y^{(s+1)} \), \( \ddot{y}^{(s+1)} \) should be obtained first. Nonetheless, \( \ddot{y}^{(s+1)} \) is also depending on \( y^{(s+1)} \). To solve this problem, the equation of motion should be used:

\[ \ddot{y}^{(s+1)} = \frac{F(t^{(s+1)})}{M} - \frac{k}{M} y^{(s+1)} \]  

Equation 2-33

By combining Equation 2-33 and 2-31:
\[ y^{(s+1)} = y^{(s)} + \frac{(\Delta t)^2}{3} \ddot{y}^{(s)} \Delta t + \frac{(\Delta t)^2}{6} \frac{F(t^{(s+1)})}{M} \frac{1 + (\Delta t)^2 k}{6 M} \]  

Equation 2-34

The newly developed equation along with Equation 2-32 presents a direct solution for \( y^{(s+1)} \). This method solves the dynamic problem more accurately.

### 2.5.4 Newmark β method

A multipurpose method developed by Newmark that could be adjusted to satisfy different problems. The equations created by Newmark:

\[ \ddot{y}^{(s+1)} = \ddot{y}^{(s)} + \frac{\Delta t}{2} (\dddot{y}^{(s)} + \dddot{y}^{(s+1)}) \]  

Equation 2-35

And \( y^{(s+1)} = y^{(s)} + \dot{y}^{(s)} \Delta t + \left( \frac{1}{2} - \beta \right) j^{(s)} (\Delta t)^2 + \beta j^{(s+1)} (\Delta t)^2 \)  

Equation 2-36

\( \beta \) could be selected depending the assumed acceleration in the specific time step. The value selected affects the convergence in every time step, the stability of the system, and the amount of accuracy. \( \beta \) value could be affected by the time step value. After investigating the method, the best results could be obtained if \( \beta \) is between \( \frac{1}{6} \) and \( \frac{1}{4} \) and \( \Delta t \) is between \( \frac{1}{6} \) and \( \frac{1}{5} \) of the shortest natural period.

### 2.5.5 Numerical integration by spreadsheet

A spreadsheet was developed using the information provided by Chopra (2006) to reformulate Newmark’s equations. Chopra modified Equation 2-35 to prevent iteration using incremental quantities as follows:

\[ \Delta y^{s} \equiv y^{s+1} - y^{s} \quad \Delta \dot{y}^{s} \equiv \dot{y}^{s+1} - \dot{y}^{s} \quad \Delta \ddot{y}^{s} \equiv \ddot{y}^{s+1} - \ddot{y}^{s} \]
\[ \Delta f^s \equiv f^{s+1} - f^s \]  
Equation 2-37

These incremental forms are used for the analysis of non-linear system. Substituting the incremental forms in Equation 2-35 & 2-36:

\[ \Delta y^{(s)} = (\Delta t) \dot{y}^{(s)} + (\gamma \Delta t) \ddot{y}^{(s)} \]
And
\[ \Delta y^{(s)} = \dot{y}^{(s)} \Delta t + \frac{(\Delta t)^2}{2} \ddot{y}^{(s)} + \beta \Delta \ddot{y}^{(s)} \Delta t^2 \]  
Equation 2-38

The acceleration incremental form is presented as:

\[ \Delta \ddot{y}^{(s)} = \frac{1}{\beta (\Delta t)^2} \Delta y^{(s)} - \frac{1}{\beta \Delta t} \dot{y}^{(s)} - \frac{1}{2\beta} \ddot{y}^{(s)} \]  
Equation 2-39

Substituting Equation 2-39 in Equation 2-38:

\[ \Delta \ddot{y}^{(s)} = \frac{\gamma}{\beta \Delta t} \Delta y^{(s)} - \frac{\gamma}{\beta} \dot{y}^{(s)} + \Delta t (1 - \frac{\gamma}{2\beta}) \ddot{y}^{(s)} \]  
Equation 2-40

Equation 2-39 and 2-40 could be substituted in an incremental equation of motion:

\[ m \Delta \ddot{y}^{(s)} + k \Delta y^{(s)} = \Delta f^{(s)} \]  
Equation 2-41

\[ \hat{k} \Delta y^{(s)} = \Delta \ddot{f}^{(s)} \]  
Equation 2-42

where

\[ \hat{k} = k + \frac{1}{\beta (\Delta t)^2} m \]  
Equation 2-43

And
\[ \Delta \hat{f}^{(s)} = \Delta f^{(s)} + \frac{1}{\beta \Delta t} m \hat{y}^{(s)} + \frac{1}{2 \beta} m \ddot{y}^{(s)} \]  

Equation 2-44

With \( \hat{k} \) and \( \Delta \hat{f}^{(s)} \) are calculated knowing the mass and the stiffness of the system and the initial velocity and acceleration at the beginning of the first time step. The incremental displacement is calculated:

\[ \Delta y^{(s)} = \frac{\Delta f^{(s)}}{\hat{k}} \]  

Equation 2-45

After calculating \( \Delta y^{(s)} \) using Equation 2-45, \( \hat{y}^{(s)} \) and \( \ddot{y}^{(s)} \) are be computed using Equation 2-39 and Equation 2-40. The acceleration of the next time step is calculated as follows:

\[ \ddot{y}^{(s+1)} = \frac{f^{(s+1)} - k y^{(s+1)}}{m} \]  

Equation 2-46

Two special cases for Newmark’s method are presented as follows:

- The average acceleration method by substituting \( \gamma = \frac{1}{2} \), \( \beta = \frac{1}{4} \).

- The linear acceleration method by substituting \( \gamma = \frac{1}{2} \), \( \beta = \frac{1}{6} \).

The following steps were used to build the stepping spreadsheet using Newmark’s method (Chopra 2011):

a) Initial calculations:
   i. Calculating the initial acceleration using Equation 2-46.
   ii. Select time step suitable for the system.
   iii. Calculating \( \hat{k} \) using Equation 2-43.
iv. Calculating: \( a = \frac{1}{\beta \Delta t} \) \( m \) and \( b = \frac{1}{2\beta} \) \( m \)

b) Calculations for each time step (s):

i. \( \Delta \tilde{f}^{(s)} = \Delta f^{(s)} + a \dot{y}^{(s)} + b \ddot{y}^{(s)} \)

ii. \( \Delta y^{(s)} = \frac{\Delta \tilde{f}^{(s)}}{k} \)

iii. \( \Delta \dot{y}^{(s)} = \frac{1}{\beta \Delta t} \Delta y^{(s)} - \frac{1}{\beta \Delta t} \dot{y}^{(s)} + \Delta t \left(1 - \frac{\gamma}{2\beta}\right) \ddot{y}^{(s)} \)

iv. \( \Delta \ddot{y}^{(s)} = \frac{1}{\beta (\Delta t)^2} \Delta y^{(s)} - \frac{1}{\beta \Delta t} \dot{y}^{(s)} - \frac{1}{2\beta} \ddot{y}^{(s)} \)

v. \( y^{(s+1)} = y^{(s)} + \Delta y^{(s)} \), \( \dot{y}^{(s+1)} = \dot{y}^{(s)} + \Delta \dot{y}^{(s)} \), \( \ddot{y}^{(s+1)} = \ddot{y}^{(s)} + \Delta \ddot{y}^{(s)} \)

c) Step b will be repeated for every time step.

The spreadsheet was built using the following assumptions:

- The effective mass calculated in Appendix A was used as the mass of the system.
- The effective bunton stiffness calculated in Appendix A was used as the system’s stiffness. The stiffness was considered as a constant stiffness.
- The initial velocity of the system was assumed equal to the impact velocity calculated in Appendix A.
- The initial acceleration, initial displacement, and the external force were assumed equal to zero as, at time zero, the skip and the bunton would act as a single system at the initial contact.
- To transfer Newmark’s method to linear acceleration method, the values of \( \gamma \) and \( \beta \) were used \( \frac{1}{2} \) and \( \frac{1}{6} \) respectively.
A sample of the spreadsheet, used to calculate the example in Appendix A, is presented in Figure 2.5-4.

![Spreadsheet Image](image)

**Figure 2.5-4** A Sample Calculation for the SDOF Spreadsheet

The spreadsheet presents the step numerical analysis for a single degree of freedom system. At time zero, as shown in Figure 2.5-4, the initial displacement and acceleration was assumed equal zero. The initial velocity was considered equal to the impact velocity calculated in Appendix A (0.05 m/s). The maximum bunton force (F max) calculated as the maximum acceleration that the system reach (y’’) multiplied by the effective mass (mₑ) was found equal 20,092 N. A small difference between the calculated bunton force in Appendix A (20,756 N) and the force calculated using the spreadsheet (20,092 N) as the result calculated in Appendix A was obtained using a graph presented by the Comro guideline. Figure 2.5-5 presents a graph of the bunton force harmonic behavior calculated by the spreadsheet. The maximum deflection (y max) calculated using the spreadsheet is equal to 3 mm. On the other hand, the guide maximum guide
deflection calculated in section 3.5 was equal to 5.6 mm. The reason behind these different values of deflection is: the maximum deflection in the spreadsheet was calculated at the bunton location but the guide maximum deflection was calculated using the SLAM program using the assumption in section 2.1.2 Inactive-rollers configuration and presented in Figure 2.1-13. The maximum guide deflection does not occur at the bunton location. Figure 2.5-6 presents a graph of the deflection harmonic behavior calculated by the spreadsheet.

![Graph](image.png)

**Figure 2.5-5** The bunton force harmonic response as calculated by the spreadsheet.
Figure 2.5-6 The deflection harmonic response as calculated by the spreadsheet.

The spreadsheet was built to compare the data calculated using the Comro guideline and estimate the dynamic behavior of the dynamic test explained in Chapter three. The damping behavior of the system is not considered as damping will not affect the initial impact force magnitude.

2.6 Conclusion

In this Chapter, the dynamic behavior of the skip/steelwork system was discussed including the factors affecting this behavior. These factors could be summarized as follows:

- The misalignment of the shaft guide.
- The bunton, skip, and guide stiffness.
- The mass of the skip and the hoisting velocity.
- The distance between two consecutive buntons.
In extreme cases, the bunton and skip stiffness are relatively high. This higher skip stiffness could be found in the balancing weight behavior (B1) presented in Figure A.1b-1. Where the balancing weight has higher stiffness than the skip (A1&A2) or man cage (C1). Also, the bunton stiffness is much higher as it is a fixed concrete bunton, less flexible than the cantilever bunton used in Skips A1&A2. The hoisting speed could be higher reaching 15 m/s, the distance between the buntons and the guide misalignment could, also, increase the impact velocity of the skip which would increase the bunton force. The mass of the skip has a great effect on the dynamic behavior. The skip mass could reach 130 tons.

This research was conducted to reduce the bunton force in extreme cases. One of the ways to conduct this reduction is by reducing the skip stiffness at the slipper location which will reduce the bunton effective stiffness and the bunton force. The skip stiffness could be reduced by adding a flexible slipper on the sides of the skip. This flexible slipper could reduce the effective bunton stiffness and the bunton impact force. The following chapters will discuss the new design of this flexible slipper including the dynamic test of the CDP material.
Chapter 3 Experimental procedure

The experimental program consisted of two testing regimes: Static compressive test and Impact test. This chapter provides details on the testing method and the equipment used to conduct both tests including the data acquisition systems.

3.1 Static compressive test

This test is designed to determine the compression stress-strain characteristics of rubber bearing pads under static compression or low strain rate loading. This research focuses on Cotton Duck Rubber Pad (CDP) as discussed in previous chapters. Samples used in this test was provided by a single manufacturer not taking into consideration other manufacturers.

3.1.1 Equipment description

This test measures the compressive stiffness of the CDP. Deflection is measured as the reduction of CDP thickness. The force is measured while applying a constant deflection rate for the CDP. The following equipment was used to perform the test:

3.1.1.1 Universal compression testing machine: Instron (see Figure 3.1-1):

The machine consists of a moving head and non-moving bottom. The moving head is used to apply the load on the CDP specimen while the non-moving bottom holds the specimen in place. The moving head contains a load cell with a capacity of 250 kN which was calibrated prior to the tests. The load cell was used to measure the load applied to the CDP specimen. The movement rate of the head was determined depending on the thickness of the CDP and the strain rate needed to test the effect of different strain rates on different rubber pad sizes. The test machine usually moves with the speed needed to
produce the specified strain rate. This movement could result in an initial impact between the machine and the specimen, as the machine starts moving before touching the specimen. To prevent applying an impact force to the specimen, a 100 N compressive force was applied to each specimen with a rate of 5 mm/min before starting the test. This will prevent unnecessary measurements at the beginning of the test so that the acquired data will describe the actual test of the specimen.

3.1.1.2 Defection gauges: 2 Linear variable differential transformers (LVDTs)

(see Figure 3.1-2):

The two LVDTs are used, to prevent the effect of plate tilting affecting the accuracy of measurements. The LVDTs are used to measure the deflection on both sides of the specimen and an average of the two gauges was calculated for use in the data analysis. The LVDTs were calibrated with a military height gauge (see Figure 3.1-3). Each LVDT has a maximum deflection of ± 50 mm.
LVDTs and Experimental set up

LVDTs calibration using a military height gauge
3.1.1.3 A high-speed camera (PFV ver 2.0) (see Figure 3.1-4)

This camera was used in the static test to check the displacement and compare the Digital Image Correlation (DIC) calculated deflection to the average deflection from the LVDTs data. This comparison was then used to check the accuracy of the camera to be used later for measuring the specimen deflection in the impact tests. After multiple trials, the most accurate displacement data (for this camera) was found using the following specifications:

- Minimum 1 m distance between the object and the camera for good focus and plane picture.
- Maximum frame rate: 5000 frames per seconds.
- Minimum resolution of 128×80 pixels. This resolution could vary depending on the frame rate.

Figure 3.1-4  High-speed camera and light source.
3.1.1.4 A source of light

The source of light provided a higher lux for the specimen to acquire video and pictures for DIC more effectively (see Figure 3.1-4).

3.1.2 Data acquisition

The Instron machine was connected to a computer program recording the data acquired by the load cell and both LVDTs. The computer software processed the data and sorted it in a table form for further analysis. The high-speed camera was connected to another computer to record the frames needed for analysis using digital image correlation (DIC) with a Matlab program. The use of DIC will be discussed extensively in the impact test portion of this chapter.

3.1.3 Samples preparation

In the static compression test, the half inch thick specimens were cut into square 40x40 mm² specimens using a band saw. The 12.7 mm thick specimens were bonded with epoxy resin (ASTM D-4236) to form different specimen thicknesses: 12.7, 25.4, and 50.8 mm (see Figure 3.1-5). These three sizes were selected to evaluate the shape factor’s effect on the strength of the CDP, as discussed in Chapter 1.

Figure 3.1-5 Three specimen sizes: 12.7, 25.4, and 50.8 mm
The following table presents the number of specimens used for every loading rate. These loading rates are used to examine the effect of different strain rate on the behavior of the CDP. Twenty-eight specimens were used to examine static compression properties of the material. The numeric system and the measured sizes of the specimens will be discussed in Chapter 4.

Table 3.1.3-1: Number of specimens used in every loading rate

<table>
<thead>
<tr>
<th>Pad Nominal size</th>
<th>Loading Rate</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>12 mm/min</td>
</tr>
<tr>
<td>40×40×12.8</td>
<td>4</td>
</tr>
<tr>
<td>40×40×25.4</td>
<td></td>
</tr>
<tr>
<td>40×40×50.8</td>
<td></td>
</tr>
</tbody>
</table>

For every loading rate and specimen thickness the expected strain rate could be calculated using Equation 3-1 and presented in Table 3.1.3-2:

Expected Strain Rate = loading rate × 60/specimen thickness   
Equation 3-1

Table 3.1.3-2 Loading rate and expected strain rate for each CDP thickness

<table>
<thead>
<tr>
<th>Specimen Thickness (mm)</th>
<th>Loading Rate (mm/min)</th>
<th>Expected Strain Rate (s⁻¹)</th>
</tr>
</thead>
<tbody>
<tr>
<td>12.7</td>
<td>12</td>
<td>15.75×10⁻³</td>
</tr>
<tr>
<td></td>
<td>80</td>
<td>105.00×10⁻³</td>
</tr>
<tr>
<td></td>
<td>200</td>
<td>262.50×10⁻³</td>
</tr>
<tr>
<td></td>
<td>400</td>
<td>524.90×10⁻³</td>
</tr>
<tr>
<td>25.4</td>
<td>200</td>
<td>131.20×10⁻³</td>
</tr>
<tr>
<td></td>
<td>400</td>
<td>262.50×10⁻³</td>
</tr>
<tr>
<td>50.8</td>
<td>400</td>
<td>131.20×10⁻³</td>
</tr>
</tbody>
</table>
These various specimen’s thicknesses and strain rates will provide a clearer understanding of the CDP behavior and properties, providing that will be used as a basis to be compared to the dynamic properties of the CDP.

### 3.1.4 Summary of the test methodology

The test procedure used is described as follows:

- The specimens were cut into the specified dimensions using a band saw as previously described in Section 3.1.3.
- The specimens were centered between the two LVDTs to maintain an accurate deflection recording. The specimens were centered inside the UCTM to maintain a well distributed load (see Figure 3.1-2).
- Using the computer program associated with the test machine, the loading rate, maximum load, and maximum deflection were specified to maintain a safe and effective test procedures.
- Before starting a test, the LVDTs and load cell were zeroed to acquire the data starting from zero points.
- The data from the load cell and the two LVDTs were recorded in tabular form during the test and saved for further analysis.
- The recorded data was used to construct stress-strain curves for each specimen and loading rate.

### 3.2 Impact test

This test was designed to determine the compression force-deflection characteristics of rubber bearing pads under impact or high strain rate loading. This
research was conducted on the same Cotton Duck Rubber Pad (CDP) as discussed in previous chapters.

3.2.1 Summary of test method and equipment

This test measured the compressive stiffness of the rubber under impact. Deflection is measured as the reduction of rubber thickness.

3.2.1.1 Modified Charpy impact load test

Preparing the Charpy machine:

The Charpy impactor was originally used to test the material ductility under impact load as specified in the ASTM Standard Test Methods for Notched Bar Impact Testing of Metallic Materials (E23). In this test, a swinging U-shaped hammer strikes the sample in two locations causing the bending of the sample until fracture. The gauge records the amount of energy absorbed by the sample to measure its ductility. By adding a thick plate to the U-shaped hammer and modifying the part holding the sample (support), the Charpy machine was transformed to an impact machine.

Modifying the hammer:

The most important parameter for plate design is the plate’s stiffness. The stiffer the plate means less plate deformation and the impact event would only measure the properties of the rubber sample. The stiffness was calculated: in the case of using a 1” (25.4mm) thick plate is equal to 2.375x10^9 N/m and for a 2” (50.8mm) thick plate is equal to 1.900x10^10 N/m. Both steel plate thicknesses could be used to transform the U-shaped pendulum hammer to a surface impact tool without affecting the impact data. For this test, the 1” (25.4mm) plate was used. The hammer and plate arrangement used for the tests is shown in Figure 3.2-1.
The swinging hammer is used to apply the impact load on the rubber specimen. However, to keep the rubber specimen from moving, a non-moving support was added to the test mechanism.

Figure 3.2-1 The hammer and plate arrangement for impact measurement.

Non-moving support design:

A thick steel plate was mounted to a steel frame, bolted to the strong floor, to act as non-moving support for the impact test. The support design requirements were as follows:

a) The support was mounted to the strong floor so that the readings would not be affected by the frame vibrations.

b) The support was assumed to have a high stiffness, so the data measured during the impact test would only measure the CDP properties.

c) The load was applied perpendicular the specimen.

d) The support was adjustable to accommodate for the change in specimen thicknesses.

To satisfy the previous requirements, the support components were designed as shown in Figure 3.2-2 and described below:
a) To maintain a perpendicular load, without inclination, spherical washers were used to remove any bolt inclination errors (Figure 3.2-2a).

b) One 1 1/8” diameter bolt used to mount the steel plate to the steel frame fixed to the concrete ground (Figure 3.2-2b). The bolt was used to hold the load cell in place.

c) Two steel plates to increase the total stiffness of the support and prevent direct impact on the bolt, providing a smooth surface to mount the specimen. The mating surfaces of the two plates were surface ground to ensure a good transfer of load between the plates and reduce friction losses (Figure 3.2-2d).

d) Eight small bolts were used to hold the two plates together (Figure 3.2-2e and f).
Figure 3.2-2 Support design: (a) Spherical washers. (b) the washer, bolt, and first steel plate. (c) The hole in the steel plate so the bolt could be easily mounted on the steel frame. (d) Two steel plates surface ground to reduce friction between the plates. (e) Eight bolts to connect the two plates together. (f) Final look for the support and the space needed for the load cell and adjusting steel washers.
The modified Charpy machine, including the modified hammer and the support, is shown in Figure 3.2-3. The body of the Charpy machine (the green frame) was also mounted to the steel frame, supporting the non-moving portion of the test machine, to prevent the movement of the machine.

![Image of Charpy machine]

**Figure 3.2-3** Modified Charpy, Impact Testing Machine.

The Charpy machine provides two release positions for the pendulum hammer. The original impact energy for both positions, as provided by the impact gauge, were 169.4 joules and 406.7 joules. These values are irrelevant in the new configuration as there is a change in the hammer mass by adding the steel plate and also a change in the
support configuration (see Figure 3.2-4). The two release positions were used to provide different impact forces to measure the rubber properties under higher strain rates.

An accelerometer was mounted on the hammer’s plate to measure the acceleration of the hammer to calculate the applied impact force. To verify the data from the accelerometer, a load cell was located at the support. The following section will discuss these sensors in more details.

3.2.1.2 Two sensors: an accelerometer and a full bridge load cell:

A one dimension shock accelerometer designed to measure up to 5000 g (equivalent to 49050 m/s²) (see Figure 3.2-5), used for testing was calibrated by the manufacturer with a calibration factor of 0.263 mV/V/g. The accelerometer was mounted
to the back of the steel plate fixed on the Charpy hammer using a tapped hole to accept the threaded stud at the base of the accerometer. To calculate the impact force on the specimen, measured mass of the hammer and plate was multiplied by the acceleration data acquired during the impact event. The hammer total mass was equal to 35.7 Kg.

![Figure 3.2-5 One Dimension Shock Accelerometer.](image)

A customized full bridge load cell was used to check the data acquired by the accelerometer. The load cell was fabricated with four strain gauges mounted on the load cell to form a full bridge using a Wheatstone bridge configuration (see Figure 3.2-6).

![Figure 3.2-6 Wheatstone Full Bridge compression configuration (NI 2016).](image)
The externally applied voltage (Vex) of 10V used in this test. The data acquisition system would acquire the output voltage (Vo). Vo, in its turn, will measure the change in resistances, representing the strain gauges: R1, R2, R3, and R4. The total resistance of the system was initially equal to 120Ω. The load cell’s capacity was calculated as a function of the area and found equal to 320 KN. The load cell was calibrated using a universal compression testing machine. The calibration equation was found as follows:

\[ LC = 208295357 \text{ V} + 1404259 \]

Equation 3-2

where \( V \) is the acquired voltage from the load cell. Figure 3.2-7 shows the final load cell configuration. The overall test set up complete with the load cell, accelerometer and hammer is presented in Figure 3.2-8.
To accommodate the change in specimen sizes and to maintain the impact load perpendicular to the surface of the specimen, two steel washers one 25mm thick and the other 37.5 mm thick were fabricated (see Figure 3.2-9). These washers were located at the load cell’s location to adjust the support to maintain horizontal impact of the hammer on the specimen without having to move the Charpy green frame. The 25mm washer was used to adjust the support location of the 25.4 mm samples. The 37.5mm washer was used to adjust the location of the 12.7 mm samples. The washers have an extended edge to hold the load cell in place so that the load cell transfers the load completely from the plate through the washer to the support.
3.2.1.4 The high-speed camera (see Figure 3.1.3-4)

As previously described in the static compression test, the high-speed camera recorded images of different stages of the impact event. These images would be used in the DIC analysis.

3.2.1.5 A Source of light

The source of light provided a higher lux for the specimen to acquire video and pictures for DIC more effectively (see Figure 3.1-4).

3.2.2 Data acquisition

Two data acquisition systems were used in this test:

NIcDAQ-9172 series C (see Figure 3.2-10) was used as the data acquisition system for these tests. A NI9205 strain gauge card was used to monitor load through the load cell. A NI9237 card was used to monitor acceleration of the hammer. The data was recorded at sample rate of 50,000 samples/second for each measurement. More information is provided in Appendix B.

Figure 3.2-9: Two washers with thicknesses 25 mm and 37.5 mm.
A high-speed camera was used to monitor the side view of the specimen during impact response. The camera recorded 5000 frames/second. These recorded images were used to measure displacement response using digital image correlation. The digital image correlation was used to measure the CDP deflection. Digital image correlation is a white light technique based on comparing images before and after deformation of the sample by comparing the degree of white colour in each pixel of the picture. To facilitate the pictures comparison, a common method was used to change the colour of the specimen surface to form a black and white random pattern. This method is called speckling. The speckling procedures is discussed in the specimen preparation portion of this experimental program. A monochromatic camera could be used to capture the stages of the test forming an array of pixels for each picture. Detecting the movement of pixels was facilitated by defining a greyscale level for each pixel. An open source matlab program (NCORR V2) was used to calculate the movement of pixels in consecutive images to

Figure 3.2-10 NI cDAQ-9172 model and both NI9205 and NI9237 Cards.
calculate the deflection of the samples over time. For higher strain rates, the DIC’s high accuracy could be achieved by using a higher frame rate.

3.2.3 Samples preparation

Samples used for impact tests were prepared using the same method as for the static test. The result was three different specimen thicknesses: 12.7, 25.4, and 50.8 mm (see Figure 3.1-4). In the impact tests, speckling all samples was essential, creating a random pattern to facilitate the DIC analysis. This pattern could be produced by 2 different methods: 1- spray 2- airbrush. The sample was colored with a white color. Using a black spray or a brush containing black color, black dots was spread over the white surface of the specimen. The result should appear as presented in Figure 3.2-11.

![Random speckled surface](image)

**Figure 3.2-11 Random speckled surface.**

This pattern was developed by coloring the background with white and then use a black spray to form a pattern of a mixed black and white speckles. The pattern of the CDP rubber is not random, as it is formed with overlaying lines (see Figure 3.2-12). This lines pattern could affect the digital image correlation, affecting in tern the displacement measurement.
The following table presents the number of specimens used for both Charpy position. The total number of specimens used for the impact test was 24 specimens. The numeric system and the measured sizes of the specimens is discussed in Chapter 4.

These various specimen’s thicknesses and Charpy positions will give an extensive understanding of the CDP behavior and properties under impact (dynamic) loading. The following chapter will discuss the experimental results based on the methods used in this chapter.

Table 3.2.3-1 Number of specimens used for each Charpy position.

<table>
<thead>
<tr>
<th>Pad Nominal size</th>
<th>Charpy position 1</th>
<th>Charpy position 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>40×40×12.7</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td>40×40×25.4</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td>40×40×50.8</td>
<td>4</td>
<td>4</td>
</tr>
</tbody>
</table>

3.2.4 Summary of the test methodology

The test procedure is described by the following steps:

- The specimen was cut into the specified dimensions.
- The specimen was speckled by coloring the surface with white color and using a black spray to form a random pattern of black and grey dots.
- The specimen was mounted to the support plate perpendicular to the hammer
• The high-speed camera and the light was adjusted to acquire the needed pictures for analysis.

• The accelerometer and the load cell were connected to the NI cDAQ-9172 to acquire the data needed.

• The hammer was latched to either of the two positions.

• Both Data acquisition system were triggered at the same time.

• The programs were designed to acquire 10 seconds’ worth of data to reduce the time required for extracting the test data.

• The final acquired data was used to build a stress-strain curve for the loading and unloading for each specimen and Charpy location.
4 Chapter 4 Experimental Results

This chapter discusses the experimental results for the testing methods presented in chapter 3. Both methods were used to study the behavior of the Cotton Duck Pad (CDP) material under static compression loading and impact loading. The CDP properties, developed in this chapter, were used to design a new flexible skip slipper plate to reduce the bunton slam load force as discussed in Chapter 2. This Chapter is divided into two parts: Part one presents the results of 28 CDP samples static compression tests with different loading rates; Part 2 presents the results of 24 CDP samples impact tests with two different Charpy positions.

4.1 Static compression test results

The 2B static compression tests were conducted using a universal compression testing machine following the static test procedures presented in Chapter 3. The testing machine recorded the applied force measured using the load cell and the displacement with two LVDTs. The data from both LVDTs are used to calculate the average displacement. Before testing, the dimensions of each specimen were measured and recorded (see Table 4.1-1). Each dimension was calculated based on the average of three measurements. The average displacement, applied force, and the measured dimensions of the sample were used to develop stress-strain curves for each sample. Every four samples were used to create a lab report. Each lab report was used to describe the behavior of CDPs samples with different strain rate and thickness combination. The samples names, presented in Table 4.1-1, are described as: the letters show different samples for the same
test set and the number used to define different test set thicknesses and loading rates combination.

Table 4.1-1: Dimensions, test type, and loading rate for each sample for tests 1 to 7.

<table>
<thead>
<tr>
<th>Sample name</th>
<th>Average measured thickness (mm)</th>
<th>Average measured length (mm)</th>
<th>Average measured width (mm)</th>
<th>Area (mm²)</th>
<th>Test type</th>
<th>Loading rate (mm/min)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A-1</td>
<td>11.80</td>
<td>42.60</td>
<td>40.70</td>
<td>1733.82</td>
<td>Compression</td>
<td>12</td>
</tr>
<tr>
<td>B-1</td>
<td>12.24</td>
<td>40.60</td>
<td>39.85</td>
<td>1617.91</td>
<td>Compression</td>
<td>12</td>
</tr>
<tr>
<td>C-1</td>
<td>12.75</td>
<td>38.51</td>
<td>39.50</td>
<td>1521.15</td>
<td>Compression</td>
<td>12</td>
</tr>
<tr>
<td>D-1</td>
<td>12.52</td>
<td>40.70</td>
<td>38.80</td>
<td>1579.16</td>
<td>Compression</td>
<td>12</td>
</tr>
<tr>
<td>A-2</td>
<td>12.52</td>
<td>40.10</td>
<td>38.90</td>
<td>1559.89</td>
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<td>80</td>
</tr>
<tr>
<td>B-2</td>
<td>12.73</td>
<td>40.20</td>
<td>39.80</td>
<td>1599.96</td>
<td>Compression</td>
<td>80</td>
</tr>
<tr>
<td>C-2</td>
<td>12.80</td>
<td>40.90</td>
<td>39.90</td>
<td>1631.91</td>
<td>Compression</td>
<td>80</td>
</tr>
<tr>
<td>D-2</td>
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<td>Compression</td>
<td>80</td>
</tr>
<tr>
<td>A-3</td>
<td>12.35</td>
<td>39.80</td>
<td>40.60</td>
<td>1615.88</td>
<td>Compression</td>
<td>200</td>
</tr>
<tr>
<td>B-3</td>
<td>12.40</td>
<td>40.50</td>
<td>40.10</td>
<td>1624.05</td>
<td>Compression</td>
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</tr>
<tr>
<td>A-4</td>
<td>12.50</td>
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<td>40.50</td>
<td>1620.81</td>
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</tr>
<tr>
<td>B-4</td>
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<td>400</td>
</tr>
<tr>
<td>C-4</td>
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<td>40.75</td>
<td>1648.34</td>
<td>Compression</td>
<td>400</td>
</tr>
<tr>
<td>D-4</td>
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<td>40.70</td>
<td>1571.02</td>
<td>Compression</td>
<td>400</td>
</tr>
<tr>
<td>A-5</td>
<td>25.10</td>
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<td>40.20</td>
<td>1628.10</td>
<td>Compression</td>
<td>160</td>
</tr>
<tr>
<td>B-5</td>
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<td>40.70</td>
<td>40.10</td>
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</tr>
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<td>Compression</td>
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<td>40.20</td>
<td>1608.00</td>
<td>Compression</td>
<td>400</td>
</tr>
<tr>
<td>C-6</td>
<td>25.50</td>
<td>39.75</td>
<td>40.70</td>
<td>1617.83</td>
<td>Compression</td>
<td>400</td>
</tr>
<tr>
<td>D-6</td>
<td>25.10</td>
<td>40.35</td>
<td>40.20</td>
<td>1622.07</td>
<td>Compression</td>
<td>400</td>
</tr>
<tr>
<td>A-7</td>
<td>50.40</td>
<td>40.10</td>
<td>40.20</td>
<td>1612.02</td>
<td>Compression</td>
<td>400</td>
</tr>
<tr>
<td>B-7</td>
<td>50.35</td>
<td>40.50</td>
<td>40.20</td>
<td>1628.10</td>
<td>Compression</td>
<td>400</td>
</tr>
<tr>
<td>C-7</td>
<td>50.90</td>
<td>40.50</td>
<td>40.80</td>
<td>1652.40</td>
<td>Compression</td>
<td>400</td>
</tr>
<tr>
<td>D-7</td>
<td>50.30</td>
<td>40.30</td>
<td>40.10</td>
<td>1616.03</td>
<td>Compression</td>
<td>400</td>
</tr>
</tbody>
</table>

Detailed results for each test set are presented in lab reports provided in Appendix C. Each report present:
- Test description including the experimental set up as described in Chapter 3, loading rate and specimen dimensions.
- Test results for one sample including strain vs time for both LVDTs data (see Figure 4.1-1) and stress-strain curve for both LVDTs data (see Figure 4.1-2).

![Figure 3.2.4-1](image1.png)

**Figure 3.2.4-1** Strain rates from 2 LVDT and their average for sample A-1

![Figure 3.2.4-2](image2.png)

**Figure 3.2.4-2**: Strain rates from 2 LVDT and their average for sample A-1

- Stress-strain curve for all four samples and their average (see Figure 4.1-3).
4.1.1 Discussion & summary

The lab reports (see Appendix C) present the data for each test without comparing the tests results. In the following sections the effect of eccentric loading, comparison of the results, and DIC results for static test are presented.

4.1.1.1 Effect of eccentric loading

Due to eccentricity of loading for some samples, the difference between the displacement acquired by the two LVDTs was relatively large. The displacement at each edge of the specimen was calculated. 2% of the total displacement difference was allowed.
between the two edges of the CDP specimen, to minimize the rotation effect on CDP (Roeder 2000). If the difference exceeded 2% of the total displacement, the test was canceled and the specimen was replaced with a new specimen.

4.1.1.2 Samples failure

As previously discussed, the test program was designed to test the properties of the CDP material without failure. But, due to change in thickness and loading rates, some samples reached failure. Figure 4.1.1-1 presents examples of failures occurred during the test. In general, the failure occurred at stresses greater than 100 MPa and strains greater than 0.25 mm/mm. Only two modes of failure were observed: shear failure with an approximate angle of 45° and failure in the epoxy material used to bond the CDP 12.7 mm in fabricating larger sample thicknesses.

As shown in Figure 4.1.1-1, the CDP did not exercise any oil secretion as presented in Lehman (2005). But it exercised a diagonal failure as shown in the figure and some delamination.
4.1.2 Results comparison

To understand the behavior of the CDP material under static compression test, a comparison between the 7 conducted tests is presented in this section. Figure 4.1.2-1 presents the average stress-strain curves for each of seven sets of tests compiled from the compressive test data gathered from the lab reports. The calculated strain rate for each test, based on the average displacement readings from the two LVDTS, the measured thickness of the samples, and the time of the test, were used to compare the behavior of the material. Table 4.1.2-1 presents the calculated strain rate and the average $E$ calculated as the slope of the tangent line to the stress-strain curve at stresses of 20, 45, and 70 MPa. These stresses present the limits as provided by AASHTO standards (2012), discussed in Chapter 1. The maximum allowable stress for CDP required by AASHTO is 20MPa (3ksi), the minimum failure stress for CDP as required by AASHTO (2012) is 70 MPa (10ksi), and 45 MPa was used to improve the analysis of the CDP behavior.
Figure 4.1.2-1: Average Stress-strain curves compared for the 7 static compressive tests.

Table 4.1.2-1: Modulus elasticity calculated at 20, 45, & 70 MPa stress and strain rate.

<table>
<thead>
<tr>
<th>Test Number</th>
<th>CDP thickness (mm)</th>
<th>Tangent modulus of Elasticity, E (MPa) at stress</th>
<th>Strain rate (s⁻¹)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>20 MPa</td>
<td>45 MPa</td>
</tr>
<tr>
<td>1</td>
<td>12.7</td>
<td>246.21</td>
<td>384.87</td>
</tr>
<tr>
<td>2</td>
<td>12.7</td>
<td>295.62</td>
<td>395.87</td>
</tr>
<tr>
<td>3</td>
<td>12.7</td>
<td>301.69</td>
<td>400.96</td>
</tr>
<tr>
<td>4</td>
<td>12.7</td>
<td>302.78</td>
<td>394.95</td>
</tr>
<tr>
<td>5</td>
<td>25.4</td>
<td>315.48</td>
<td>411.42</td>
</tr>
<tr>
<td>6</td>
<td>25.4</td>
<td>324.45</td>
<td>406.76</td>
</tr>
<tr>
<td>7</td>
<td>50.8</td>
<td>261.47</td>
<td>391.74</td>
</tr>
</tbody>
</table>

The stress-strain curves, presented in Figure 4.1.2-1, shows the non-linear behavior of the CDP. Figure 4.1.2-2 presents the relation between E and the strain rates at stresses 20, 45, and 70 MPa. Tangent modulus (E) at stress equal 20 MPa was found approximately equal 300 MPa. Comparing the values of E at 45 MPa, the modulus of elasticity of the seven tests was found approximately equal 400 MPa. Followed by E at 70 MPa was approximately equal to 600 MPa. In general, the small changes in strain rates
did not affect major changes on the material properties. Additional CDP thicknesses should be tested to investigate the behavior and compare more CDP shape factors and strain rates.

Figure 4.1.2.2: The modulus of elasticity E values for the static test for each strain rate calculated at stresses 20, 45, & 70 MPa.

4.1.3 DIC results for static test

The high-speed camera was used in the static test to compare the displacement values provided by the DIC analysis with the data provided by the calibrated LVDTs. Multiple trials were conducted to measure the accuracy of the speckle sizes, the minimum distance between the camera and the speckled specimen, the frame rate, the size of the image in pixel, and their effect on the DIC analysis. The optimum accuracy was achieved by the following:

- A distance between the camera and the specimen equal to 1.8 m.
- A frame rate was equal to 5000 frames/s.
- An image size equal to 128x80 pixels
- The speckle size was adequate.

Figure 4.1.3-1 presents an example of the accuracy satisfied by the criteria listed above while comparing the displacement provided by the LVDTs and the displacement data obtained from the digital image correlation (DIC) analysis for sample A-7. The displacement obtained from the DIC analysis was similar to the average between the displacement obtained by the two LVDTs. Figure 4.1.3-2 presents the images of the CDP compression test as captured by the high-speed camera and used in the DIC analysis.

![Graph](image)

**Figure 4.1.3-1:** Comparison between the displacement data calculated from DIC analysis and the displacement data obtained from the LVDTs for sample A-7.
4.1.3 Conclusion

The compressive static test results, as presented in this portion of this chapter, described the behavior of the CDP under low strain rates. The small differences in strain rates did not affect the behavior of the CDP. Although, the change in the shape factor of the specimen affected the behavior of CDP as the 25.4 mm thick specimens showed higher stiffness than both 12.7 and 50.8 mm specimens. This behavior was unexpected, for this reason more research needs to be conducted to explore the shape factor effect on the CDP behavior. The DIC analysis, using the criteria presented in section 4.1.3, was successful. The same criteria were used in the impact test presented in the following part of this chapter.
4.2 Impact test results

The impact tests were conducted using a modified Charpy testing machine following the impact test procedures presented in Chapter 3. The accelerometer and the load cell were connected to a data acquisition system recording the applied and the reaction forces. The displacement was measured with the aid of the high-speed camera coupled with DIC analysis. The images were recorded by the high-speed camera using Photon FastCam Viewer ver 2 program. The recorded images were analysed using DIC to measure the CDP displacement. Before testing the dimensions of each specimen was recorded (see Table 4.2-1). The measured dimensions of the sample, the analysed displacement, and the recorded applied force were used to calculate stress-strain curves for each sample. A set samples were used to create each of six lab reports. Each lab report discusses the behavior of CDPs samples with different thicknesses and Charpy position combinations. The samples names, presented in Table 4.2-1, are described as: the letters show different samples for the same test set and the number used to define different test set thicknesses and loading rates combination.
Table 4.2-1: Specimens dimensions and Charpy positions for tests 8 to 13.

<table>
<thead>
<tr>
<th>Sample Name</th>
<th>Average Measured Thickness (mm)</th>
<th>Average Measured length (mm)</th>
<th>Average Measured width (mm)</th>
<th>Area (mm²)</th>
<th>Test type</th>
<th>Charpy position</th>
</tr>
</thead>
<tbody>
<tr>
<td>A-8</td>
<td>12.16</td>
<td>40.25</td>
<td>41.20</td>
<td>1658.30</td>
<td>Impact 1</td>
<td></td>
</tr>
<tr>
<td>B-8</td>
<td>12.40</td>
<td>40.65</td>
<td>40.10</td>
<td>1630.07</td>
<td>Impact 1</td>
<td></td>
</tr>
<tr>
<td>C-8</td>
<td>12.27</td>
<td>39.72</td>
<td>39.85</td>
<td>1582.84</td>
<td>Impact 1</td>
<td></td>
</tr>
<tr>
<td>D-8</td>
<td>12.32</td>
<td>40.60</td>
<td>40.35</td>
<td>1638.21</td>
<td>Impact 1</td>
<td></td>
</tr>
<tr>
<td>A-9</td>
<td>12.47</td>
<td>39.75</td>
<td>40.05</td>
<td>1591.99</td>
<td>Impact 2</td>
<td></td>
</tr>
<tr>
<td>B-9</td>
<td>12.40</td>
<td>39.94</td>
<td>40.80</td>
<td>1629.55</td>
<td>Impact 2</td>
<td></td>
</tr>
<tr>
<td>C-9</td>
<td>12.45</td>
<td>40.37</td>
<td>41.20</td>
<td>1663.24</td>
<td>Impact 2</td>
<td></td>
</tr>
<tr>
<td>D-9</td>
<td>12.46</td>
<td>40.20</td>
<td>41.10</td>
<td>1652.22</td>
<td>Impact 2</td>
<td></td>
</tr>
<tr>
<td>A-10</td>
<td>25.02</td>
<td>40.80</td>
<td>39.60</td>
<td>1615.68</td>
<td>Impact 1</td>
<td></td>
</tr>
<tr>
<td>B-10</td>
<td>24.70</td>
<td>39.25</td>
<td>40.10</td>
<td>1573.93</td>
<td>Impact 1</td>
<td></td>
</tr>
<tr>
<td>C-10</td>
<td>24.70</td>
<td>39.35</td>
<td>39.70</td>
<td>1601.90</td>
<td>Impact 1</td>
<td></td>
</tr>
<tr>
<td>D-10</td>
<td>24.95</td>
<td>39.45</td>
<td>40.40</td>
<td>1593.78</td>
<td>Impact 1</td>
<td></td>
</tr>
<tr>
<td>A-11</td>
<td>24.70</td>
<td>39.90</td>
<td>40.25</td>
<td>1605.98</td>
<td>Impact 2</td>
<td></td>
</tr>
<tr>
<td>B-11</td>
<td>24.65</td>
<td>40.10</td>
<td>40.30</td>
<td>1616.03</td>
<td>Impact 2</td>
<td></td>
</tr>
<tr>
<td>C-11</td>
<td>25.10</td>
<td>40.10</td>
<td>40.20</td>
<td>1612.02</td>
<td>Impact 2</td>
<td></td>
</tr>
<tr>
<td>D-11</td>
<td>24.40</td>
<td>40.20</td>
<td>40.20</td>
<td>1616.04</td>
<td>Impact 2</td>
<td></td>
</tr>
<tr>
<td>A-12</td>
<td>50.24</td>
<td>39.30</td>
<td>40.00</td>
<td>1572.00</td>
<td>Impact 1</td>
<td></td>
</tr>
<tr>
<td>B-12</td>
<td>50.00</td>
<td>40.20</td>
<td>40.20</td>
<td>1616.04</td>
<td>Impact 1</td>
<td></td>
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<td>C-12</td>
<td>50.00</td>
<td>40.25</td>
<td>39.20</td>
<td>1577.80</td>
<td>Impact 1</td>
<td></td>
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<td>D-12</td>
<td>49.80</td>
<td>40.20</td>
<td>39.80</td>
<td>1599.96</td>
<td>Impact 1</td>
<td></td>
</tr>
<tr>
<td>A-13</td>
<td>50.02</td>
<td>40.20</td>
<td>40.30</td>
<td>1620.06</td>
<td>Impact 2</td>
<td></td>
</tr>
<tr>
<td>B-13</td>
<td>50.20</td>
<td>40.70</td>
<td>40.20</td>
<td>1636.14</td>
<td>Impact 2</td>
<td></td>
</tr>
<tr>
<td>C-13</td>
<td>50.10</td>
<td>40.30</td>
<td>39.80</td>
<td>1603.94</td>
<td>Impact 2</td>
<td></td>
</tr>
<tr>
<td>D-13</td>
<td>49.90</td>
<td>40.30</td>
<td>40.05</td>
<td>1614.02</td>
<td>Impact 2</td>
<td></td>
</tr>
</tbody>
</table>

Detailed results for each test set are presented in the lab reports provided in Appendix D. Each report provides:

- Test description including the experimental set up as described in Chapter 3, loading rate and specimen dimensions.

- Test results for one sample including stress-time plot for one sample including: accelerometer data, load cell data, and a noise removed plot for accelerometer data (see Figure 4.2-1) and strain-time plot (see Figure 4.2-2).
Figure 4.1.3-1 Stress-time plot for sample A-8 (including accelerometer data, load cell data, and a noise removed plot for accelerometer data)

Figure 4.1.3-2 Strain-time plot for sample A-8.

- Stress-strain curve for all four samples and their average (see Figure 4.2-3).
Figure 4.1.3-3 The Stress-Strain curves of the four samples tested under impact (first hammer position) and their average (S-8-Average)

- Observations and discussions for each test.

The lab reports (see Appendix D) present the data for each test without comparing the results of the tests. Additional tests were conducted to the samples C-9, D-9, C-11, D-11, C-13, and D-13. The purpose of additional tests was to compare the number and maximum stresses to fail different CDP thicknesses using the second position of the modified Charpy machine. The following sections will compare the results of the six tests conducted, discuss the sample failure modes including the additional tests conducted on
samples C-9, D-9, C-11, D-11, C-13, and D-13, review the errors affected the results, and provide some recommendations to improve the test and prevent these errors.

4.2.1 Results comparison

To understand the behavior of the CDP material under impact test, a comparison between the 6 conducted tests is presented in this section. Figure 4.2.1-1 presents the average stress-strain curves for all tests compiled from the data presented in the lab reports in Appendix D. The strain rate for each test, the displacement acquired from DIC analysis, the measured thickness of the samples, and the adjusted accelerometer data, were used to compare the stress-strain results and the modulus of elasticity (E). The ascending branch of the stress-strain data could be presented by a second-degree polynomial regression curves (see Figure 4.2.1-2). The tangent modulus of elasticity (E) was calculated as the derivative of the second-degree curve as presented in Table 4.2.1-1. Table 4.2.1-2 presents the calculated strain rates and the average E calculated as the slope of the tangent line to the stress-strain at stresses equal to 20, 45, 70, & 100 MPa. These stresses present the stress limits as provided by AASHTO standards (2012), discussed in Chapter 1. The maximum stress for CDP recommended by AASHTO is 20 MPa (3ksi), the minimum total strength for CDP as required by AASHTO (2012) is 70 MPa (10ksi), and 45 & 100 MPa were used to improve the understanding of the CDP behavior.
Figure 4.2.1-1: Stress-strain curves compiled for the 8 sets of impact test.

Figure 4.2.1-2: Second-degree polynomial regression of stress-strain data for tests 8 to 13.
Table 4.2.1-1: Stress second-degree polynomial, secant modulus of elasticity equation (E) and $R^2$.

<table>
<thead>
<tr>
<th>Test number</th>
<th>Stress function ($\sigma$) (Mpa)</th>
<th>Secant modulus of elasticity equation (E) (MPa)</th>
<th>$R^2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>8</td>
<td>$2144.5\varepsilon^2 + 16.565\varepsilon$</td>
<td>$2144.5\varepsilon + 16.565$</td>
<td>0.9969</td>
</tr>
<tr>
<td>9</td>
<td>$1869.7\varepsilon^2 - 5.3779\varepsilon$</td>
<td>$1869.7\varepsilon - 5.3779$</td>
<td>0.9999</td>
</tr>
<tr>
<td>10</td>
<td>$1674.9\varepsilon^2 - 10.47\varepsilon$</td>
<td>$1674.9\varepsilon - 10.47$</td>
<td>0.9994</td>
</tr>
<tr>
<td>11</td>
<td>$1967\varepsilon^2 - 11.557\varepsilon$</td>
<td>$1967\varepsilon - 11.557$</td>
<td>0.9995</td>
</tr>
<tr>
<td>12</td>
<td>$1845.8\varepsilon^2 + 23.151\varepsilon$</td>
<td>$1845.8\varepsilon + 23.151$</td>
<td>0.9985</td>
</tr>
<tr>
<td>13</td>
<td>$1306.4\varepsilon^2 + 95.018\varepsilon$</td>
<td>$1306.4\varepsilon + 95.018$</td>
<td>0.9976</td>
</tr>
<tr>
<td>Average</td>
<td>$1984.5\varepsilon^2 - 15.446\varepsilon$</td>
<td>$1984.5\varepsilon - 15.446$</td>
<td>0.9745</td>
</tr>
</tbody>
</table>

The tangent modulus of elasticity was obtained from the acquired stress strain data points. The calculated tangent E values, presented in Figure 4.2.1-1, shows the non-linear behavior of the CDP. Figure 4.2.1-3 presents the data from Table 4.2.1-2 and the relation between E and the strain rates at stresses 20, 45, 70 & 100 MPa. E at stress equal 20 MPa was found between a value of 340 to 420 MPa. This wide range for the modulus of elasticity values proves that the strain-rate play an important role in changing the CDP behavior. Modulus of elasticity (E) was generally increasing while the strain rate increased from 41.995 to 119.190 s$^{-1}$ followed by a reduction at strain-rate equal to 193.74 s$^{-1}$. Comparing the values of E at 45 MPa, the modulus of elasticity of the six tests was found between 480 to 620 following the same increase and decrease behavior as the behavior presented in the modulus of elasticity calculated at 20 MPa. Modulus of elasticity E at 70 MPa was approximately between 700 and 770 MPa following the same behavior. Finally, E at 100 MPa was between 830 and 860 for tests 9 and 11. In general, the change in strain rates affected the behavior of CDP in this impact test. Additional CDP
thicknesses should be tested to investigate the behavior and compare more CDP shape factors and strain rates.

Table 4.2.1-2: Tangent modulus elasticity at 20, 45, 70, & 100 MPa stress and strain rate.

<table>
<thead>
<tr>
<th>Test Number</th>
<th>CDP t (mm)</th>
<th>Tangent Modulus of Elasticity, E (MPa) at stress</th>
<th>Strain rate (s^{-1})</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>20 MPa</td>
<td>45 MPa</td>
</tr>
<tr>
<td>8</td>
<td>12.7</td>
<td>418.83</td>
<td>618.80</td>
</tr>
<tr>
<td>9</td>
<td>12.7</td>
<td>376.26</td>
<td>560.68</td>
</tr>
<tr>
<td>10</td>
<td>25.4</td>
<td>361.79</td>
<td>540.19</td>
</tr>
<tr>
<td>11</td>
<td>25.4</td>
<td>393.43</td>
<td>582.50</td>
</tr>
<tr>
<td>12</td>
<td>50.8</td>
<td>366.14</td>
<td>N/A</td>
</tr>
<tr>
<td>13</td>
<td>50.8</td>
<td>339.04</td>
<td>478.06</td>
</tr>
</tbody>
</table>

Figure 4.2.1-3 The modulus of elasticity E values for the static test for each strain rate calculated at stresses (20, 45, 70, & 100).

After presenting the effect of higher strain rates on the CDP’s modulus of elasticity, Figure 4.2.1-1 presented an overview of the effect of the reduction in stiffness
and the maximum stresses for the CDP samples. For the same Charpy position, same acting impact force, the maximum stress reduced while increasing the thickness of the CDP. For example, position 1, used for tests 8, 10, & 12, acquired maximum stresses were 80, 60, & 40 MPa for the CDP thicknesses 12.7, 25.4, & 50.8 mm respectively. For position 2, used for tests 9, 11, & 13, acquired maximum stresses were 150, 100, & 70 MPa for the CDP thicknesses 12.7, 25.4, & 50.8 mm respectively. In other words, using the same area and increasing the thickness of the CDP sample, under the same impact force, the maximum stress of the CDP will decrease and the stiffness of the system will reduce. Chapter 5 provides more detail about the new flexible slipper plate design.

4.2.2 Multiple impact tests and failure modes

This section provides data acquired during conducting multiple impact tests on samples C-9, D-9, C-11, D-11, C-13, and D-13. These extra tests were conducted to investigate how many impact tests would be required to fail different CDP thickness samples, the change in the CDP behavior after multiple impacts, and the different types of failure possible in multiple impact events. This section is divided into three portions each portion investigates the effect of multiple impact tests on different sample thicknesses.

4.2.2.1 Samples C-9 & D-9

Samples C-9 & D-9 were 12.7 mm thick and tested using the second Charpy position as discussed in Lab report 9 (see Appendix D). After completing the first impact test for each sample, the thickness of the tested samples was measured and a second impact test was conducted. The same procedure was repeated multiple times until the
failure of the samples. Table 4.2.2-1 presents the samples original thicknesses, the thicknesses after each impact test, and the maximum stress reached in each impact test. These stresses were calculated based on the maximum force acquired by the accelerometer and the original sample dimensions.

Table 4.2.2-1 displays the thicknesses and maximum stresses for each impact test for samples C-9 & D-9.

<table>
<thead>
<tr>
<th>Test Number</th>
<th>Sample C-9</th>
<th>Sample D-9</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test Number</td>
<td>CDP t (mm)</td>
<td>Max. stress (MPa)</td>
</tr>
<tr>
<td>0</td>
<td>12.45</td>
<td>157.524</td>
</tr>
<tr>
<td>1</td>
<td>11.83</td>
<td>173.156</td>
</tr>
<tr>
<td>2</td>
<td>11.50</td>
<td>129.266</td>
</tr>
</tbody>
</table>

After the first impact test, both samples showed some delamination in the cotton duck layers. The maximum stress was found around 155 MPa. The samples exercised 5% reduction of the original thickness. After the second impact, both samples showed some shear failures, increase of the maximum stress to 175 MPa, and another 2.5% reduction in the original thickness (see Figure 4.2.2-1c). Sample C-9 was tested a third time showing a reduction in the maximum stress to 130 MPa and a destruction of the sample (see Figure 4.2.2-1a&b).

**Figure 4.2.2-1** Failure modes for samples C-9 & D-9.
4.2.2.2 Samples C-11 & D-11

Samples C-11 & D-11 were 25.4 mm thick and tested using the second Charpy position as discussed in Lab report 11 (see Appendix D). After completing the first impact test for each sample, the thickness of the tested samples was measured and a second impact test was conducted. The same procedure was repeated multiple times until the failure of the samples. Table 4.2.2-2 presents the samples original thicknesses, the thicknesses after each impact test, and the maximum stress reached in each impact test. These stresses were calculated based on the maximum force acquired by the accelerometer and the original sample dimensions.

Figure 4.2.2-2 Failure modes for samples C-11, & D-11.
Table 4.2.2-2 the thicknesses and maximum stresses for each impact test for samples C-11 & D-11.

| Test Number | Sample C-11 | | | Sample D-11 | | |
|-------------|-------------|-------------|-------------|-------------|-------------|
| Test Number | CDP t (mm) | Max. stress (MPa) | CDP t (mm) | Max. stress (MPa) |
| 0           | 25.10       | 105.216     | 24.40       | 107.020      |
| 1           | 24.63       | 120.848     | 23.60       | 123.373      |
| 2           | 24.10       | 123.253     | 23.40       | 129.205      |
| 3           | 23.60       | 131.670     | 23.10       | 135.158      |
| 4           | 23.60       | 132.151     | 23.05       | N/A          |
| 5           | 23.45       | 134.208     | 22.90       | 127.582      |
| 6           | 23.35       | 131.791     | 22.80       | 134.556      |
| 7           | 23.30       | 137.502     | 22.70       | 135.278      |
| 8           | N/A         | N/A         | 22.60       | 136.059      |
| 9           | N/A         | N/A         | 22.50       | 172.675      |

After the first impact test, both samples showed some delamination in the cotton duck layers. The maximum stress was found around 105 MPa. The samples exercised 2% to 3% reduction of the original thicknesses to C-11 & D-11 respectively. From the second impact test to the sixth impact test, both samples showed more delamination failures, an increase of the maximum stress to reach 134 MPa, and another 5% reduction in the original thickness. Sample C-11 was tested a seventh time and the sample failed (see Figure 4.2.2-2 a, b, &c). Sample D-11 failed after the ninth impact test and the maximum stress reached 172 MPa (see Figure 4.2.2-2 d &e). In general, the maximum stress increases until failure then the stress drops dramatically but the sample carries load after failure.

### 4.2.2.3 Samples C-13 & D-13

Samples C-13 & D-13 were 50.8 mm thick and tested using the second Charpy position as discussed in Lab report 13 (see Appendix D). After completing the first impact test for each sample, the thickness of the tested samples was measured and a second
impact test was conducted. The same procedure was repeated ten times. For this extended test, only ten impact tests were conducted as the stresses were not high enough to fail the samples. Table 4.2.2-3 presents the samples original thicknesses, the thicknesses after each impact test, and the maximum stress reached in each impact test. These stresses were calculated based on the maximum force acquired by the accelerometer and the original sample dimensions. For some of the tests, the accelerometer did not acquire the data.

Table 4.2.2-3 the thicknesses and maximum stresses for each impact test for samples C-13 & D-13.

<table>
<thead>
<tr>
<th>Test Number</th>
<th>C-13</th>
<th>D-13</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>CDP t (mm)</td>
<td>Max. stress (MPa)</td>
</tr>
<tr>
<td>0</td>
<td>50.10</td>
<td>67.847</td>
</tr>
<tr>
<td>1</td>
<td>49.00</td>
<td>77.756</td>
</tr>
<tr>
<td>2</td>
<td>48.70</td>
<td>80.557</td>
</tr>
<tr>
<td>3</td>
<td>48.60</td>
<td>81.510</td>
</tr>
<tr>
<td>4</td>
<td>48.35</td>
<td>82.680</td>
</tr>
<tr>
<td>5</td>
<td>48.25</td>
<td>82.680</td>
</tr>
<tr>
<td>6</td>
<td>48.10</td>
<td>84.274</td>
</tr>
<tr>
<td>7</td>
<td>47.90</td>
<td>84.075</td>
</tr>
<tr>
<td>8</td>
<td>47.80</td>
<td>84.552</td>
</tr>
<tr>
<td>9</td>
<td>47.7</td>
<td>84.552</td>
</tr>
<tr>
<td>10</td>
<td>N/A</td>
<td>N/A</td>
</tr>
</tbody>
</table>

After the first impact test, both samples showed little delamination in the cotton duck layers. The maximum stress was found around 68 MPa. The samples exercised 2% reduction of the original thicknesses. From the second impact test to the last impact test, both samples showed more delamination failures, an increase of the maximum stress to reach 85 MPa, and another 3% reduction in the original thickness. In the last few tests, the samples did not experience big changes in their thicknesses and the maximum stress. In general, the maximum stress increases while the number of impacts increases.
4.2.3 Errors and recommendations to improve the test results

Multiple errors affected the test results. These errors are presented as follows:

- The data acquired by the accelerometer had unexpected noise (see Figure 4.2.3-1). The source of the noise could be a higher frequency mode produced by the hammer arm low stiffness or possible baseline shift of the output data for the accelerometer sensor (Soss 2008). This noise was reduced by adjusting the data by taking the average of multiple consecutive data points to form a single data point. Some of these noise errors were irrecoverable (see Figure 4.2.3-2). The reason for these irreparable noises was the movement of the wire connecting the accelerometer and the data acquisition system. Due to the low-quality images acquired by the high-speed camera, some data were difficult to perform the DIC analysis which canceled the tested specimen.

- The impact test was conducted using multiple impact events so that the pendulum impacted the samples multiple times as there were no arrest mechanism to prevent the hammer from swinging back and impacting the specimen until the force damps out. Figure 4.2.3-3 presents one series of impact forces exercised in one test in terms of acceleration vs time.

![Graph of Force vs Time](image)

**Figure 4.2.3-1** Example of recoverable accelerometer data noise.
Figure 4.2.3-2  Example of irrecoverable accelerometer data noise (high frequency and baseline shift error).

Figure 4.2.3-3  Multiple impact events in a single impact test.

The following recommendations could prevent the errors discussed previously and improve the test results:
• The noise presented in Figures 4.2.3-1&2 could be prevented by eliminating the movement of the wire connecting the accelerometer and the data acquisition system.

• The DIC analysis could be improved by improving the quality of the images, increasing the number of frames to be comparable to the number of data points acquired by the accelerometer, and improving the speckling sizes.

• The Charpy could be improved by adding an arrest mechanism to prevent multiple impact events.

• A more advanced trigger system should be used to trigger both high-speed camera and force acquisition systems at the same time. As discussed in Chapter 3 manual trigger was used in these tests.

4.2.4 Conclusion

In this portion of the Chapter, the impact test’s results described the behavior of the CDP under higher strain rates between 40 to 200 s$^{-1}$. The differences in strain rates affected the behavior of the CDP. The modulus of elasticity started by increasing reaching strain rate of 110 s$^{-1}$ then E reduced under strain rate of 192 s$^{-1}$. Although, the change in the shape factor of the specimen’s effect could not be identified on the behavior of CDP under higher strain rates. More research should be conducted to explore the shape factor effect on the CDP behavior including a wider range of strain rates. For the multiple impact tests presented in section 4.2.3, the CDP showed a remarkable behavior, if the stress was kept under 70 MPa. The 50.8 mm thick CDP resisted more than ten impact tests while showing a limited delamination in the pad, a limited increase in impact force, and a total of 5% of thickness reduction. More investigation needed to study the maximum number
of impact required to fail the CDP under several stresses. The pad showed a 2% reduction in thickness after the first test and a 2% reduction for the rest of the test, which shows that the first impact stretched the cotton duck layers causing a sudden increase in E and a sudden reduction in the original thickness.

4.3 Comparing static compression test results and impact test results

This Chapter reported the results for both static compression and impact tests for CDP and described its behavior providing a general overview on the change of modulus of elasticity while changing the strain rate. The results from both tests could be described by the following:

- The modulus of elasticity increases while increasing the strain rate (see Figure 2.3-1). In the figure, comparing E calculated at 20 MPa, the modulus of elasticity increased from a value of 300 MPa at strain rates 0 to 0.4 s\(^{-1}\) to values from 340 to 420 MPa at strain rates between 40 & 200 s\(^{-1}\). This increase is 15% to 40% increase in E depending on the strain rate.
- At 45 MPa, E increased from a value of 400 MPa at strain rates 0 to 0.4 s\(^{-1}\) to values from 480 to 620 MPa at strain rates between 40 & 200 s\(^{-1}\). This increase is 20% to 55% increase in E depending on the strain rate.
- At 70 MPa, E increased from a value of 600 MPa at strain rates 0 to 0.4 s\(^{-1}\) to values from 700 to 780 MPa at strain rates between 40 & 200 s\(^{-1}\). This increase is 17% to 30% increase in E depending on the strain rate.
- The maximum percentage increase in E is calculated equal to 55%.
Figure 4.2.4-1 E values at different strain rates calculated at 20, 45, & 70 MPa for static and impact tests.

The following chapters used the SDOF system created in chapter 2 and the test results presented in this chapter to build a guideline for describing the new skip slipper design. Using E calculated from the dynamic test and the maximum stresses for CDP provided by AASHTO Standards (20 MPa), the new skip slipper design could be designed to reduce the bunton impact force as discussed in Chapter 2.
5 Chapter 5 Developing a new slipper design

Using the dynamic analysis presented in Chapter 2, the bunton force or the impact force and the maximum guide displacement were determined using Comro guideline or the SDOF program. These variables are the key components to design the side slippers. In Chapter 4, cotton duck pads (CDP) were tested under impact to be used in the new slipper design. In this chapter, the new slipper design is developed including a design guide based on the information presented in both chapter 2 & 4. The chapter is divided as follows:

- The current slipper design.
- Developing the new slipper design using the SDOF program.
- Recommendations for modifying Comro guideline.
- Conclusion

5.1 Current slipper plate design

The slipper plates, presented in Comro guideline (1990) and SANS 10208:3 commentary (2001), are steel plates covered with High density polyethylene, to reduce friction, mounted on both sides of the conveyance to carry the acting lateral loads due to the misalignment of guides. These lateral loads can perform in two different directions: in the plane of guides or in perpendicular to this plane (see Figure 5.1-1). The magnitude of the lateral loads could be obtained using Comro (1990) or an equivalent SDOF program using numerical integration of the time history as presented in Chapter 2. Using this information, the steel could be designed to carry the impact loads without harming the body of the conveyance (see Figure 5.1-2).
These slippers do not affect the magnitude of the impact force acting on the conveyance or the bunton, rather they transfer the load directly to the steel work without
harming the skip body. For extreme cases, where the bunton and skip have very high stiffness, or the skip is moving with higher velocity, the slippers would transfer the high impact loads directly to the skip and the bunton which increases the steelwork needed to carry these large impact forces.

A material with less stiffness could be used in the slipper design to reduce these extreme impact forces by decreasing the stiffness of the system. The new slipper design, presented in this chapter, uses elastomer pads as a cushion to change the dynamic properties of the system, reduce the impact force and transfer a smaller portion of force and displacement to the skip and bunton system.

5.2 The new slipper design using the SDOF Newmark’s method

The new slipper design is based on creating a lower effective bunton stiffness by adding an elastomer pad with low stiffness to the effective bunton stiffness in series (see Figure 5.2-1). The final effective stiffness can be calculated by adding the pad stiffness to the original effective stiffness in series as presented in Equation 5-1.

\[ k_{be} + k_p = k_{be} \]

![Figure 5.2-1: The addition of the elastomer pad stiffness to the SDOF system presented in chapter 2 to form a new SDOF system.](image)
$k_{pbe} = \frac{k_{be}k_p}{k_{be}+k_p}$  \hspace{1cm} \text{Equation 5-1}

where: $k_{be}$ is the effective bunton stiffness, $k_p$ is the elastomer pad stiffness, and $k_{pbe}$ is the combined effective stiffness.

In this research, CDPs are recommended due to its higher overall compressive strength compared to plain unreinforced elastomeric pads (PEP), Fiberglass-Reinforced Pad (FGR), and Steel-Reinforced Elastomeric Bearing. The CDPs were tested under impact force as presented in chapter 3 & 4. The results of these tests were as follows:

- The stress-strain curve for impact is presented as a second-degree polynomial curve regression.
- The modulus of elasticity was calculated by dividing the stress equation by strain and a secant value of E at each strain (see Table 5.2.1-1).
- The dynamic amplification factor, comparing the static and dynamic test, was found equal to a maximum of 50% of the static compressive modulus of elasticity.
- Under impact loads, the CDP carried ten impacts of stress 70 MPa up to 85 MPa without showing any failure.

Using the results shown above, the SDOF program, created in chapter 2, can be used to obtain a new dynamic response including the calculating the maximum bunton force. Two methods were used in calculating the pad stiffness and using Equation 5-1 to calculate the new effective stiffness:

- The CDP stiffness function obtained by multiplying the pad modulus of elasticity’s function ($E_p$) by the ratio between the pad area (A) and thickness (t) (see Equation 5-2).
\[ k_p = \frac{E_p A}{t} \]  

Equation 5-2

- To facilitate the design procedures, constant value for \( E_p \) \((E^*)\) can be calculated using conservation of energy method, where the area under the pad stress-strain curve is equal to the area under the curve created by a constant value \( E^* \) at a given stress value.

5.2.1 Calculating the maximum bunton force and displacement using \( E_p \)

Table 5.2.1-1 presents the modulus of elasticity equations created for each dynamic test performed in chapter 4. Using Equation 5-2, \( k_p \) functions were calculated by the \( E_p \) functions presented in the table.

Table 5.2.1: \( E_p \) modulus of elasticity and pad stiffness \( k_p \) based on secant calculations, where \( u \) is the pad displacement and \( t \) is the pad thickness.

<table>
<thead>
<tr>
<th>Test number</th>
<th>Modulus of elasticity equation ((E_p)) (MPa)</th>
<th>Pad Stiffness ( k_p ) (N/m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>8</td>
<td>2144.5(\varepsilon) + 16.57</td>
<td>((2.145 \times 10^9 \frac{u}{t} + 16.57 \times 10^6) \frac{A}{t})</td>
</tr>
<tr>
<td>9</td>
<td>1869.7(\varepsilon) - 5.38</td>
<td>((1.870 \times 10^9 \frac{u}{t} - 5.38 \times 10^6) \frac{A}{t})</td>
</tr>
<tr>
<td>10</td>
<td>1674.9(\varepsilon) - 10.47</td>
<td>((1.675 \times 10^9 \frac{u}{t} - 10.47 \times 10^6) \frac{A}{t})</td>
</tr>
<tr>
<td>11</td>
<td>1967.0(\varepsilon) - 11.56</td>
<td>((1.967 \times 10^9 \frac{u}{t} - 11.56 \times 10^6) \frac{A}{t})</td>
</tr>
<tr>
<td>12</td>
<td>1845.8(\varepsilon) + 23.15</td>
<td>((1.846 \times 10^9 \frac{u}{t} + 23.15 \times 10^6) \frac{A}{t})</td>
</tr>
<tr>
<td>13</td>
<td>1306.4(\varepsilon) + 95.02</td>
<td>((1.306 \times 10^9 \frac{u}{t} + 95.02 \times 10^6) \frac{A}{t})</td>
</tr>
<tr>
<td>Average</td>
<td>1984.5(\varepsilon) - 15.45</td>
<td>((1984.5 \times 10^9 \frac{u}{t} - 15.45 \times 10^6) \frac{A}{t})</td>
</tr>
</tbody>
</table>

The \( k_p \) functions were used in the numerical integration spreadsheet presented in Chapter 2 to evaluate the effect of adding \( k_p \) in series to the bunton effective stiffness in
the example staged in Appendix A. The elastomer bearing pad stiffness \( (k_p) \) was calculated for every time step by using the previous step pad displacement \( (u_p) \) as shown in Equation 5-3.

\[
k_{p+1} = C_1 u_p + C_2 \quad \text{Equation 5-3}
\]

where \( k_{p+1} \) is the pad stiffness at a certain time step, \( C_1 \) & \( C_2 \) are constants depending on the area and thickness of the CDP the equation used from Table 5.2.1-1, \( u_p \) is the pad displacement in the previous time step. \( u_p \) was calculating using the following equation:

\[
u_p = u_i - u_{be} \quad \text{Equation 5-4}
\]

where \( u_i \) is the total displacement of the system and \( u_{be} \) is the displacement obtained by dividing the bunton force by the effective bunton stiffness at each time step.

Multiple trials were performed while changing the area and thickness of the CDP used to reduce the bunton force and guide displacement. For the example presented in Appendix A, a pad with the same size as the side slippers (500 mm × 200 mm) was used (see Figure 5.2-2). This pad size resulted in a negligible reduction in the impact force as the stiffness of the pad was relatively high (see Figure 5.2-4). After multiple trials, the most practical solution to reduce the area of the CDP without affecting the size of the slippers was by using CDP washers at the bolts locations of the original slippers (see Figure 5.2-2). The washer sizes and number could be determined by calculating the maximum impact force expected to be carried by the pad using the numerical integration spreadsheet, as presented in Chapter 2, or by using Comro guideline and the maximum allowable stress that the CDP could carry. The allowable compressive stress, provided by ASSHTOO 2012, is 3 ksi (20.3MPa). This allowable stress could be increased to 35 MPa,
as the impact events are rare to happen with the calculated magnitude. Also, The CDP could carry stress 70 MPa in impact without failure as presented in the test results chapter 4. It is recommended to use washer sizes provided by the manufactures.

![Figure 5.2-2: CDP mounted between the slipper plate and the skip (Pad and washer forms).](image)

To study the effect of changing the pad size on the bunton force and the bunton displacement, the following graphs (Figure 5.2-3&4) were created using the mine information provided in Appendix A and adding CDPs in series to the effective bunton stiffness while changing the thicknesses and the areas of the pads.

Comparing the results of adding the CDP to the system, the following was found:

- Reducing the area and increasing the thickness of the pad decreases the total pad stiffness which reduces the bunton force.
- In this example, the pad stiffness had to be reduced considerably to show a significant effect on the bunton force. Using larger areas and smaller pad thickness
was not effective. On the other hand, using washers with smaller area and larger thickness reduced the impact force.

- The percent reduction in bunton displacement was equivalent to the percent reduction in bunton force with a maximum of 38% reduction using six CDP washers of total area equal to 821 mm$^2$ and thickness equal to 50.8 mm (see Figure 5.2-4). The maximum stress of these washers was 15.2 MPa.

In figure 5.2-3&4: $f_p$ is the bunton force while the CDP is mounted to the slipper and $f_b$ is the bunton force without CDP.

![Graph](image)

**Figure 5.2-3** Ratio between the bunton forces before and after adding CDP compared by different pad thicknesses with the same area (821mm$^2$).
Figure 5.2-4 Ratio between the bunton forces before and after adding CDP compared by different pad areas with the same pad thickness (50.8 mm).

The data presented previously shows the effect of the CDP on the behaviour of the SDOF system with non-linear elastomer pad stiffness. The stiffness of the pad is desirable to be smaller than the effective bunton stiffness to significantly reduce the bunton force. Using a function to calculate the CDP stiffness is impractical. A single value of modulus of elasticity is needed to simplify the design process for the mine steelwork. The following section describes the equivalent secant value of modulus of elasticity ($E^*$) using conservation of energy method.

### 5.2.2 Calculating the maximum bunton force and displacement using secant value of modulus of elasticity ($E^*$)

It is common to convert a non-linear behaviour of a certain material (concrete for example) to a linear equivalent behaviour to simplify the design procedures. One of the common methods used to transform the non-linear material behaviour to an equivalent
linear behaviour is conservation of energy. The energy absorbed by the pad was calculated using integration of the polynomial equation to calculate the area under the stress-strain curve up to the maximum allowable stresses: 20 MPa by ASHTO 2012 and the recommended allowable stress 35 MPa. Two values of $E^*$ were calculated for each allowable stress. Each value of $E^*$ could be used to calculate the maximum bunton force and the maximum skip displacement. Figure 5.2-5 presents two values for $E^*$, calculated to be equivalent to the nonlinear behaviour up to 20 MPa allowable stress, 300 MPa to calculate the maximum skip displacement and bunton force and 150 MPa to calculate the maximum pad deflection. Figure 5.2-6 presents two values for $E^*$, calculated to be equivalent to the nonlinear behaviour up to 35 MPa allowable stress, 400 MPa to calculate the maximum skip displacement and bunton force and 200 MPa to calculate the maximum pad deflection. The area under the curves created by the $E^*$ values are equal to the area under the actual stress-strain curve of CDP while maintaining the same maximum stress (20 or 35 MPa) or maximum strain.
Figure 5.2-5: Actual stress-strain curve of CDP and the curves created by the equivalent values of $E^*$ for 20 MPa allowable stress.

Figure 5.2-6: Actual stress-strain curve of CDP and the curves created by the equivalent values of $E^*$ for 35 MPa allowable stress.
$E^*$ values are then used to calculate $k_p$ using Equation 5-2 which is then used to calculate $k_{pbe}$ using Equation 5-1. The new effective bunton stiffness could be then used to perform the dynamic analysis to calculate the maximum bunton force and maximum skip displacement. It is recommended to use the $E^*$ equivalent for the purpose of the analysis, bunton force or skip displacement.

Both non-linear $E_p$ and $E^*$ were compared using the SDOF integration spreadsheet and the results were equivalent depending on the value of the maximum stress. $E^*$ was then used to calculate the new effective bunton stiffness which later is used in calculating the maximum bunton force or skip displacement using Comro guideline.

5.3 The new slipper design using Comro guideline

The slipper design using Comro guideline could be performed as follows:

- Calculate the bunton force and skip displacement using SANS 10208-4: (2011). In this step, the bunton force and skip displacement should be calculated using the original information of the skip system. Including the effective mass and stiffness, the velocity and the maximum guide misalignment.

- Calculate the new rubber slipper area needed using the bunton force and a maximum allowable stress selected (20 or 35 MPa), using Equation 5-5:

$$A = \frac{F_b}{20 \text{ or } 35 \text{ MPa}}$$  
Equation 5-5

- Select pad thickness based on the available space between the guide and slipper and calculate $k_p$ using $E^*$ values depending on the response needed to be tested. In other words, calculate two different $k_p$ one used to calculate the maximum new bunton force and skip displacement and the other used to calculate maximum pad deflection.
• Calculate new \( k_{pbe} \) including \( k_p \) as a stiffness in series (Equation 5-1).

• Calculate the new bunton force & guide displacement using SANS 10208-4 (2011) and the new \( k_{pbe} \).

• Design steel work.

The following flow chart describes the steps needed to complete the slipper design and the new steel work design based on the new bunton forces and skip displacement.

Using this procedure, the design results were comparable to the results using integration spreadsheet applying non-linear and linear systems. Table 5.3-1 presents the design of the CDP using the methods previously discussed in this chapter.
Figure 5.3-1: Flowchart for the new slipper design.

Designing the new slipper

Calculate the bunton force & guide displacement using COMRO (1990)

Calculate the new rubber slipper area $A = \frac{F_b}{20 \text{ or } 35 \text{ MPa}}$ and pick CDP thickness

$\sigma = 20 \text{ MPa}$
$\sigma = 35 \text{ MPa}$

$E^* = 300 \text{ MPa}$ (skip displacement & Bunton force)
$E^* = 400 \text{ MPa}$ (skip displacement & Bunton force)

Calculate $k_p = \frac{E^* A}{t}$

Calculate $k_{pbe} = \frac{k_{be}k_p}{k_{be}+k_p}$

Calculate the new bunton force & guide displacement using Comro 1990 & $k_{pbe}$

Design steel work
Table 5.3-1: SDOF integration Spreadsheet results for bunton force, skip displacement, pad stress & pad deflection (for skip without pad and with 6 washers using Eq function and the equivalent E*) and the results using Comro guideline for the same washer sizes.

<table>
<thead>
<tr>
<th>Input</th>
<th>Without Pad</th>
<th>Comro (1990) $E^*=300$</th>
<th>Washer (MPa) $E_p$</th>
<th>Washer (MPa) $E^*=150$</th>
<th>Washer (MPa) $E^*=300$</th>
</tr>
</thead>
<tbody>
<tr>
<td>impact velocity (v)</td>
<td>0.05 m/s</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Time step ($\Delta t$)</td>
<td>0.0015 sec</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Effective mass (me)</td>
<td>23721.15 kg</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Washer Area</td>
<td>0.21 in$^2$</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Washer #</td>
<td>6</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Bunton Stiffness (kb)</td>
<td>26820000 N/m</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pad thickness (Tp)</td>
<td>50.8 mm</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Skip Stiffness (ks)</td>
<td>9100000 N/m</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Output</th>
<th>Without Pad</th>
<th>Comro (1990) $E^*=300$</th>
<th>Washer (MPa) $E_p$</th>
<th>Washer (MPa) $E^*=150$</th>
<th>Washer (MPa) $E^*=300$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Force (N)</td>
<td>20066</td>
<td>13046</td>
<td>12718</td>
<td>10292</td>
<td>12952</td>
</tr>
<tr>
<td>Stress (Mpa)</td>
<td>N/A</td>
<td>15.897</td>
<td>15.497</td>
<td>12.541</td>
<td>15.783</td>
</tr>
<tr>
<td>Skip displacement (m)</td>
<td>0.0030</td>
<td>0.0050</td>
<td>0.0020</td>
<td>0.0015</td>
<td>0.0019</td>
</tr>
<tr>
<td>Pad deflection (m)</td>
<td>N/A</td>
<td>0.0043</td>
<td>0.0042</td>
<td>0.0375</td>
<td></td>
</tr>
</tbody>
</table>

In Table 5.3-1, the results of using the $E_p$ function and $E^*=300$ MPa in the dynamic analysis and using SANS 10208-4 (2011) guideline were similar in maximum bunton force and maximum pad stress. The results were similar for the pad deflection between using $E_p$ function and $E^*=150$ MPa. The Skip displacement was similar for $E_p$ function and $E^*=300$ MPa, the total deflections of the system using SANS 10208-4 (2011) and $E^*=150$ MPa are equivalent.

5.4 Conclusion

In this chapter, the new slipper design was introduced. Using the information on CDP, tested in Chapter 3& 4, a pad was used between the slipper’s steel plate and skip to create a lower effective stiffness and reduce the bunton force. This pad stiffness was tested
using SDOF integration spreadsheet for non-linear behavior and the equivalent linear behavior. The results using both behaviours were similar. A guideline was introduced to use the new slipper design and Comro guideline to calculate the maximum bunton stiffness and skip displacement to design the mine steelwork.
6 Chapter 6 Conclusion

6.1 Conclusions

This research was conducted to improve the skip side slippers design. The new slipper design provides information about how to reduce slam loads in mineshafts of deep mines. Slam loads are highest when there is steel-to-steel contact between the slippers and the shaft rails. This hard contact has the most harmful effect on the steelwork of the mineshaft. The slam load is primarily influenced by the following:

- Misalignment of the shaft guide.
- The bunton, skip, and guide stiffness.
- The mass of the skip and the hoisting velocity.

The worst-case scenario for slam loads has a combination of high stiffness buntons and skips with high velocity hoisting along with high mass skips. In extreme cases, the mass of the skip may be up to 130 tonnes and the hoist speed up to 18m/s.

To increase the productivity of a mine, the mass and the velocity of the skip should be maximized. A high mass and high velocity system could result in high slam loads, requiring stronger steelwork for the mineshaft. This would likely also result in a stiffer mineshaft rail system, which has the effect of further increasing potential slam loads. An alternative to increasing capacity of the rails is to reduce the effective stiffness of the skip/guide system. This reduction in stiffness will reduce the maximum bunton force. One way to reduce stiffness is to provide an additional flexible material at the slipper location that acts in series with the skip and rail system. This flexible slipper reduces the effective bunton stiffness and therefore the bunton impact force.
Cotton Duck Pads (CDP), used as bearing pads in multiple applications, were tested to examine their behavior under static and dynamic compression loads. Chapter 4 reported the results for both types of tests for CDP and described the material with a focus on the change of modulus of elasticity with changing the strain rate. The results from both tests are described by the following:

- As expected, the modulus of elasticity increases with increasing the strain rate.
- The modulus of elasticity can be represented by a second-degree polynomial curve.
- CDP carries multiple impacts, with stresses between 70 and 85 MPa, without showing any noticeable failure.
- The maximum amplification factor of the modulus of elasticity due to high strain rate was found equal to 50%.

Using the results of the tests performed on CDP, the new slipper design was created using CDP between the slipper’s steel plate and the skip body. This new system reduced the effective bunton stiffness which reduced the maximum bunton force. To facilitate the new design procedures, a constant modulus of elasticity ($E^*$) was introduced to create a linear system instead of a non-linear system represented by a second-degree polynomial equation. Two limits for the maximum stress of CDP were introduced: the maximum allowable stress provided by AASHTO Standards (20 MPa) and a suggested maximum allowable stress based on the conducted tests (35 MPa). The designer has the choice to select any of these two values. The designer starts by finding the maximum bunton force from the analysis of the original system using SANS 10208-4 (2011). Using the calculated maximum force and maximum allowable stress, allows the designer to
select the area of the pad needed to keep the pad stress level below the allowable stress limit. Then, a pad thickness is selected depending on the allowed distance between the slipper plate and the guide surface. The pad stiffness \( k_p \) is calculated using the pad area and thickness previously obtained and \( E^* \) value: 300 MPa for the stress of 20 MPa and 400 MPa for the stress of 35 MPa. A new effective bunton stiffness is then calculated and a new bunton force is obtained using SANS 10208-4 (2011). The results of this method were compared to the results of using numerical integration of time history for a single degree of freedom system while using the non-linear pad behavior.

6.2 Suggestions for further research

This thesis provided a theoretical study for a new slipper design methodology. Further research needed could be presented as follows:

- Mount the new slipper of the side of an actual skip to compare the real-life force magnitude with the theoretically calculated bunton force.
- Test CDP for fatigue to measure the life span for this pad using the stress limits discussed in this thesis.
- Test more CDP sizes under static loading and impact to improve the understanding of the CDP behavior.
- Test more CDP samples while examining the temperature effect on its behavior using different loading rates.
References


- Cloth, Duck, cotton or cotton-polyester blend, synthetic rubber, impregnated and laminated, oil resistant (1989) MIL-C-882E


- NCORR v1.2 Open source 2D-DIC Matlab Software.


- Stantec (2013), Shaft Steel Testing- Sample No 1 Typical set – Plan & Details, Drawing No: D-JANSEN_2616_SK 0001, Saskatchewan, Canada.


Appendix
A  Example for calculating the bunton force using Comro (1990)

A.1  General information

Shaft configuration (see Figure A.1-1).

Shaft maintenance classification: C (default for design) (e= 20 mm)

Buntons : HSS 203 x 203 x16

Guides : HSS 203 x 203 x16

Conveyances: Compartments A1 and A2 – 22T Skip
               Compartments B1 – Balancing weight
               Compartments C1 – man cage

Bunton Spacing:  8 m

Hoisting speed:  10 m/s

Conveyance Data: Skip A1 (22.6 x 1.85 x 1.65)

Full mass:  94,400 kg

Empty mass:  42,400 kg

Stiffness ($k_s$):  9.1 KN/mm

Distance of skip c.g. from slipper plate:  11.3 m

Moment of inertia about global Z axis:  $4.19 \times 10^6$ kg m$^2$
A.2 Shaft steelwork design

Steelwork stiffness: $k_s = 2.49 \times 10^6$ N/m

$k_b = 26.82 \times 10^6$ N/m

But the skip is flexible $k_s = 9.1$ KN/mm

Bunton effective stiffness $k_{be} = \frac{k_b k_s}{k_b + k_s}$

$$k_{be} = \frac{26.82 \times 10^6 \times 9.1 \times 10^6}{26.82 \times 10^6 + 9.1 \times 10^6}$$

$$k_{be} = 6.79 \times 10^6$$ N/m

Effect mass $m_e = \frac{m_s l}{(l+m_s h^2)}$
\[ m_e = 23,721.15 \text{ kg} \]

Stiffness ratio \[ r_k = \frac{k_{be}}{k_g} \]

\[ = \frac{6.79 \times 10^6}{2.49 \times 10^6} \]

\[ r_k = 2.72 \]

Note: \( k_{be} \) is used instead of \( k_b \) because the skip is flexible.

Impact velocity ratio \[ \frac{u}{v} = \frac{2e}{L} \]

\[ = \frac{2(0.02)}{8} = 0.005 \]

\( \overline{k_b} : \) nondimensional steelwork stiffness at bunton

\[ \overline{k_b} = \frac{k_{be}L^2}{m_e v^2} \]

\[ = \frac{6.79 \times 10^6 \times 8^2}{23,721.15 \times 10^2} = 183.2 \]

Using the figures provided in Comro 1990 (Figure A.2-1 to A.2-4), the following was determined:

Ratio of rebound velocity to the impact velocity \( u \ (r_u) \)

\[ r_u = 1.3 \]
Figure A.2-1: Contour plot of rebound velocity ratio \( r_u \) (Comro 1990).

Nondimensional bunton force \( \bar{P}_b = 0.07 \)

Bunton force:
\[
P = \frac{\bar{P} m_e v^2}{L}
\]

\[
= \frac{(0.07)(23,721.15)(10^2)}{(8)} = 20,756 \text{ N}
\]
Figure A.2-2: Contour plot of nondimensional bunton force $\overline{P_b}$ for $u/v = 0.005$ (Comro 1990).

Non-dimensional guide displacement $\overline{\delta_s} = 0.000667$

Guide displacement $\delta_s = \overline{\delta_s} \cdot L = (0.000667) \cdot 8 = 0.00533$ m
Figure A.2.3: Contour plot of nondimensional skip displacement $\delta_s$ for $u/v = 0.005$ (Comro 1990).

Nondimensional guide bending moment $\bar{M} = 0.011$

Guide bending moment $M = \bar{M} m_e v^2$

$$= (0.011) (23,721.15) (10^2) = 26,093.265 \text{ N.m}$$
Figure A.2-4: Contour plot of nondimensional guide bending moment $\bar{M}$ for $u/v = 0.005$ (Comro 1990).
B Data acquisition systems

B.1 Data acquisition system for accelerometer and load cell

The data acquisition system was used to acquire the forces, applied force measured using the accelerometer and the reaction force measured by the load cell. The impact event data was transmitted from the accelerometer and load cell as an electrical current to National Instrument Analog input cards. Two types of cards were used: NI 9205 and NI 9237.

- NI 9205 contains 16 Differential Analog Input channels. It could acquire ± 200 mV and up to ± 10 V. The maximum number samples acquired per second is 250,000 samples. This card was used to acquire the change in voltage in the accelerometer.

- NI 9237 is used to acquire Bridge Analog Input. It has a half- and full-bridge measurements built-in. The Card could acquire signals of ± 25mV/V and a maximum number of samples of 50,000 samples/second/channel. This card was used to measure the output voltage of the load cell.

The two cards were connected in series using the model NI cDAQ-9172 series C which helped acquiring data from both cards using the onboard clock at the same time intervals. The model was then connected to a computer using a high-speed USB 2. The computer was equipped with a software (Labview) providing control and access the model data including triggering the test, acquiring data in a tabulated form, and stopping the test. More details about building the Labview program and National Instruments equipment would be found in Labview help and NI guidelines.
B.2 Data acquisition system for displacement

The data acquisition system used for acquiring the deflection of the rubber specimen. This system consists of two stages; recording the frames captured by the high-speed camera during the impact event for every rubber sample and digital image correlation program to analyze the deflection of the rubber. A computer software (Photon FastCam Viewer ver 2) was used to record the frames needed. This software could be used to adjust the frame rate, shutter rate, image resolution, trigger the image recording, stop the record, and save the captured images in different formats (images or video extensions) (see Figure B.2-1). The images recorded by the camera and Photron FastCam viewer ver 2 was used in the DIC analysis.

Figure B.2-1: Photron FastCam viewer ver 2 used to record the high-speed camera’s images.
The digital image correlation (DIC) was conducted using an open source 2D-DIC Matlab Software NCORR V2. The following are the steps used to complete the DIC analysis using the NCORR V2 program:

(a) After capturing the images using the high-speed camera, the speckled images were recorded on the program as a reference image (the first recorded image before the impact event) and the impact event recorded images (see Figure B.2-2).

![Figure B.2-2: The reference image and the impact event images.](image)

(b) Drawing the region of interest: In this step, to reduce the analysis time, an area of interest is defined to specify the area of the image where the DIC should be performed (see Figure B.2-3).
Performing the DIC analysis: In this step, the program performs the analysis based on multiple parameters as: number of seeds, the greyscale level of the pixel picked for the analysis, and relating the images using step analysis. The product of this step is determining the movement of each pixel picked for the analysis through the entire series of images in terms of pixels.

After determining the movement of each pixel in the region of interest in pixel units, the pixels’ displacements were transformed to mm units using unit conversion tool. The unit conversion tool performs by defining a measured distance on the image, typically the thickness of sample before the test, to determine the number of pixels in one mm distance (see Figure B.2-4).
The result of the DIC analysis is presented in an array form of pixel displacements for each image in the region of interest (see Figure B.2-5). An average value of displacement was calculated for the most deflected column or row of pixel displacement values. As previously discussed, this method was used in the static compression test to be compared to the LVDTs’ data and ensure the accuracy of the method including the speckle size, resolution of the images, and the frame rate used.

Figure B.2-4: Identifying the unit conversion factor
More details about the DIC method used to acquire the displacement is found in the NCORR guidelines.

**Figure B.2-5:** Matlab Array of pixel displacements for image 23.

More details about the DIC method used to acquire the displacement is found in the NCORR guidelines.
C  Lab reports for static testing
**C.1 Lab report 1**

**Specimen Dimensions:**

![Specimen Dimensions](image)

**Figure C.1-1: Specimen Dimensions**

- **Material Type:** Cotton Duck Pad
- **Number of Specimens:** 4
- **Test Type:** Compression
- **Loading Rate:** 12 mm/min

**Descriptive notes:**

- The specimen was cut into the specified dimension using a band saw (see Figure C.1-1).
- The specimen was centered between the two LVDTs to maintain an accurate deflection recording.
- The loading rate was specified using the computer program associated with the machine.
- The data from the load cell and the 2 LVDTs were recorded using the same computer program.
- The specimen was centered inside the UCTM to maintain a well distributed load (see Figure C.1-2).
- The LVDTs are mounted on the steel plate that was used to distribute the load to the specimen.
- The machine is also equipped with a load cell with a capacity of 250 KN.
- A maximum displacement of 5 mm was set for the moving head to prevent overloading.

**Experimental Set up:**

![Experimental Set up](image)

**Figure C.1-2: (UCTM) set up.**

**Strain Rate and Stress-Strain Plots Sample A-1:**

**Figure C.1-3: Strain rates from 2 LVDT and their average for sample A-1**

**Figure C.1-4: Stress-strain curves for 2 LVDTs and their average for sample A-1**

**Date of test:** December 13th, 2016.

**Testing machine:** Universal compression testing machine (UCTM).
Observations and Discussion:

- The stiffness of the rubber increases while the applied load increases. The stress-strain relation is a non-linear relationship.
- The strain rate, shown as slope in Figure C.1-3, was relatively constant over the duration of the test due to the constant specified displacement base load rate. The average LVDT calculated strain rate = 14.1×10^{-3} s^{-1}.
- An average displacement was calculated from the data provided by the two LVDTs (see Figures C.1-3&4).
- The average stress-strain data will be then used to be compared with other strain rates and other rubber thicknesses.
- The provided Stress-strain curves, in Figure C.1-5, are calculated using the measured dimensions of each sample as specified in Table 4.1-1. Table C.1-1 shows calculated Stresses and strains of all four samples.

<table>
<thead>
<tr>
<th>Stress (MPa)</th>
<th>Strain:0.05</th>
<th>Strain:0.10</th>
<th>Strain:0.20</th>
<th>Strain:0.30</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stress: B -1</td>
<td>4.275</td>
<td>12.815</td>
<td>29.362</td>
<td>88.172</td>
</tr>
<tr>
<td>Stress: C -1</td>
<td>5.263</td>
<td>15.410</td>
<td>44.718</td>
<td>100.040</td>
</tr>
<tr>
<td>Stress: D -1</td>
<td>2.519</td>
<td>11.497</td>
<td>41.591</td>
<td>97.276</td>
</tr>
</tbody>
</table>
C.2 Lab report 2

Specimen Dimensions:

Figure C.2-1: Specimen Dimensions

Material Type: Cotton Duck Pad
Number of Specimens: 4
Test Type: Compression
Loading Rate: 80 mm/min

Date of test: December 13th, 2016.
Testing machine: Universal compression testing machine (UCTM).

Experimental Set up:

Figure C.2-2: (UCTM) set up.

Descriptive notes:

- The specimen was cut into the specified dimension using a band saw (see Figure C.2-1).
- The specimen was centered between the two LVDTs to maintain an accurate deflection recording.
- The loading rate was specified using the computer program associated with the machine.
- The data from the load cell and the 2 LVDTs were recorded using the same computer program.
- The specimen was centered inside the UCTM to maintain a well distributed load (see Figure C.2-2).
- The LVDTs are mounted on the steel plate that was used to distribute the load to the specimen.
- The machine is also equipped with a load cell with a capacity of 250 KN.
- A maximum displacement of 5 mm was set for the moving head to prevent overloading.

Strain Rate and Stress-Strain Plots Sample A-2:

Figure C.2-3: Strain rates from 2 LVDT and their average for sample A-2

Figure C.2-4: Stress-strain curves for 2 LVDTs and their average for sample A-2
Figure C.2-5: The Stress-strain curves of the four samples tested under 80mm/min loading rate and their average (S-2-Average)

Observations and Discussion:

- The stiffness of the rubber increased while increasing the applied force.
- The strain rate, shown as slope in Figure C.2-3, was relatively constant over the duration of the test due to the constant specified displacement base load rate. The average LVDT calculated strain rate = $79.7 \times 10^{-3}$ s$^{-1}$.
- An average displacement was calculated from the data provided by the two LVDTs (see Figures C.2-3&4).
- In Figure C.2-5, the stress-strain curve for samples B-2 & D-2 had unexpectedly stopped, as the moving head started applying the force after moving 1 or 2 mm without applying any load.
- The provided Stress-strain curves, in Figure C.2-5, are calculated using the measured dimensions of each sample as specified in Table 4.1-1. Table C.2-1 shows calculated Stresses and strains of all four samples.

Table C.2-1: Stresses at different specified strain for each of the four samples, calculated based on the measured dimensions of each sample.

<table>
<thead>
<tr>
<th>Stress (MPa)</th>
<th>Strain:0.05</th>
<th>Strain:0.10</th>
<th>Strain:0.20</th>
<th>Strain:0.23</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stress: A -2</td>
<td>6.569</td>
<td>18.077</td>
<td>50.403</td>
<td>64.243</td>
</tr>
<tr>
<td>Stress: B -2</td>
<td>5.331</td>
<td>17.386</td>
<td>52.698</td>
<td>67.983</td>
</tr>
<tr>
<td>Stress: C -2</td>
<td>3.502</td>
<td>15.359</td>
<td>49.420</td>
<td>64.021</td>
</tr>
<tr>
<td>Stress: D-2</td>
<td>4.927</td>
<td>14.204</td>
<td>50.064</td>
<td>62.216</td>
</tr>
</tbody>
</table>
C.3 Lab report 3

Specimen Dimensions:

![Specimen Dimensions](image)

Figure C.3-1: Specimen Dimensions

Material Type: Cotton Duck Pad
Number of Specimens: 4
Test Type: Compression
Loading Rate: 200 mm/min

Date of test: December 13\textsuperscript{th}, 2016.
Testing machine: Universal compression testing machine (UCTM).

Experimental Set up:

![Experimental Set up](image)

Figure C.3-2: (UCTM) set up.

Descriptive notes:
- The specimen was cut into the specified dimension using a band saw (see Figure C.3-1).
- The specimen was centered between the two LVDTs to maintain an accurate deflection recording.
- The loading rate was specified using the computer program associated with the machine.
- The data from the load cell and the 2 LVDTs were recorded using the same computer program.
- The specimen was centered inside the UCTM to maintain a well distributed load (see Figure C.3-2).
- The LVDTs are mounted on the steel plate that was used to distribute the load to the specimen.
- The machine is also equipped with a load cell with a capacity of 250 KN.
- A maximum displacement of 5 mm was set for the moving head to prevent overloading.

Strain Rate and Stress-Strain Plots Sample A-3:

![Strain Rate and Stress-Strain Plots](image)

Figure C.3-3: Stress-strain curves for 2 LVDTs and their average for sample A-3
Figure C.3-4: Strain rates from 2 LVDT and their average for sample A-3
Stress-strain curve for all four samples and average:

Figure C.3-5: The Stress-strain curves of the four samples tested under 12mm/min loading rate and their average (S-3-Average)

Observations and Discussion:
- The stiffness of the rubber increases while the applied load increases.
- The strain rate, shown as slope in Figure C.3-3, was relatively constant over the duration of the test due to the constant specified displacement base load rate. The average LVDT calculated strain rate = 202.8 \times 10^{-3} \text{ s}^{-1}.
- An average displacement was calculated from the data provided by the two LVDTs (see Figures C.3-3&4).
- In Figure C.3-5, the stress-strain curve for samples B-3 & D-3 had unexpectedly stopped, as the moving head started applying the force after moving 1 or 2 mm without applying any load.
- The provided Stress-strain curves, in Figure C.3-5, are calculated using the measured dimensions of each sample as specified in Table 4.1-1. Table C.3-1 shows calculated Stresses and strains of all four samples.

Table C.3-1 Stresses at different specified strain for each of the four samples, calculated based on the measured dimensions of each sample.

<table>
<thead>
<tr>
<th>Stress (MPa)</th>
<th>Strain:0.05</th>
<th>Strain:0.10</th>
<th>Strain:0.20</th>
<th>Strain:0.23</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stress: A -3</td>
<td>4.817</td>
<td>16.753</td>
<td>51.354</td>
<td>66.414</td>
</tr>
<tr>
<td>Stress: B -3</td>
<td>3.609</td>
<td>14.574</td>
<td>48.724</td>
<td>62.933</td>
</tr>
<tr>
<td>Stress: C -3</td>
<td>4.483</td>
<td>15.766</td>
<td>50.877</td>
<td>65.669</td>
</tr>
<tr>
<td>Stress: D -3</td>
<td>3.875</td>
<td>14.891</td>
<td>49.331</td>
<td>63.836</td>
</tr>
</tbody>
</table>
Specimen Dimensions:

![Specimen Dimensions](image)

**Figure C.4-1:** Specimen Dimensions

Material Type: Cotton Duck Pad
Number of Specimens: 4
Test Type: Compression
Loading Rate: 400 mm/min

Descriptive notes:
- The specimen was cut into the specified dimension using a band saw (see Figure C.4-1).
- The specimen was centered between the two LVDTs to maintain an accurate deflection recording.
- The loading rate was specified using the computer program associated with the machine.
- The data from the load cell and the 2 LVDTs were recorded using the same computer program.
- The specimen was centered inside the UCTM to maintain a well distributed load (see Figure C.4-2).
- The LVDTs are mounted on the steel plate that was used to distribute the load to the specimen.
- The machine is also equipped with a load cell with a capacity of 250 KN.
- A maximum Load of 160 KN was set for the moving head to prevent overloading.

**Experimental Set up:**

![Experimental Set up](image)

**Figure C.4-2:** (UCTM) set up.

**Figure C.4-3:** Strain rates from 2 LVDT and their average for sample A-4

**Figure C.4-4:** Stress-strain curves for 2 LVDTs and their average for sample A-4
Stress-strain curve for all four samples and average:

Figure C.4-4: The Stress-strain curves of the four samples tested under 400mm/min loading rate and their average (S-4-Average)

Observations and Discussion:
- The stiffness of the rubber increases while the applied load increases.
- The strain rate, shown as slope in Figure C.4-3, was relatively constant over the duration of the test due to the constant specified displacement base load rate. The average LVDT calculated strain rate = 403.2×10^{-3} s^{-1}.
- An average displacement was calculated from the data provided by the two LVDTs (see Figures C.4-3&4).
- In Figure C.4-5, the stress-strain curve did not stop as previous tests, due to changing the limit loading 160 KN instead of having a maximum head displacement equal to 5 mm.
- The provided Stress-strain curves, in Figure C.4-5, are calculated using the measured dimensions of each sample as specified in Table 4.1-1. Table C.4-1 shows calculated Stresses and strains of all four samples.

<table>
<thead>
<tr>
<th>Stress (MPa)</th>
<th>Strain:0.05</th>
<th>Strain:0.10</th>
<th>Strain:0.20</th>
<th>Strain:0.29</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stress: A -4</td>
<td>2.503</td>
<td>11.302</td>
<td>44.087</td>
<td>93.892</td>
</tr>
<tr>
<td>Stress: B -4</td>
<td>2.429</td>
<td>11.186</td>
<td>43.949</td>
<td>92.940</td>
</tr>
<tr>
<td>Stress: C -4</td>
<td>2.071</td>
<td>10.109</td>
<td>41.211</td>
<td>86.935</td>
</tr>
<tr>
<td>Stress: D -4</td>
<td>1.725</td>
<td>8.651</td>
<td>38.606</td>
<td>81.860</td>
</tr>
</tbody>
</table>
C.5 Lab report 5

Specimen Dimensions:

![Specimen Dimensions](image)

Figure C.5-1 Specimen dimensions

<table>
<thead>
<tr>
<th>Specimen Dimensions:</th>
</tr>
</thead>
<tbody>
<tr>
<td>25.4 mm</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Material Type:</th>
<th>Cotton Duck Pad</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of Specimens:</td>
<td>4</td>
</tr>
<tr>
<td>Test Type:</td>
<td>Compression</td>
</tr>
<tr>
<td>Loading Rate:</td>
<td>160 mm/min</td>
</tr>
</tbody>
</table>

Descriptive notes:

- The specimen was cut into the specified dimension using a band saw (see Figure C.5-1).
- The specimen was centered between the two LVDTs to maintain an accurate deflection recording.
- The loading rate was specified using the computer program associated with the machine.
- The data from the load cell and the 2 LVDTs were recorded using the same computer program.
- The specimen was centered inside the UCTM to maintain a well distributed load (see Figure C.5-2).
- The LVDTs are mounted on the steel plate that was used to distribute the load to the specimen.
- The machine is also equipped with a load cell with a capacity of 250 KN.
- A maximum displacement of 9 mm was set for the moving head to prevent overloading.

Strain Rate and Stress-Strain Plots Sample A-5:

![Strain Rate and Stress-Strain Plots](image)

Figure C.5-3: Strain rates from 2 LVDT and their average for sample A-5

Figure C.5-4: Stress-strain curves for 2 LVDTs and their average for sample A-5

Date of test: December 13th, 2016.

Testing machine: Universal compression testing machine (UCTM).

Experimental Set up:

![UCTM set up](image)

Figure C.5-2: (UCTM) set up.

Specimen Dimensions:

- 25 mm
Stress-strain curve for all four samples and average:

Figure C.5-5: The Stress-strain curves of the four samples tested under 160mm/min loading rate and their average (S-5-Average)

Observations and Discussion:
- The stiffness of the rubber increases while the applied load increases.
- The strain rate, shown as slope in Figure C.5-3, was relatively constant over the duration of the test due to the constant specified displacement base load rate. The average LVDT calculated strain rate = 72.7\times10^{-3} \text{ s}^{-1}.
- An average displacement was calculated from the data provided by the two LVDTs (see Figures C.5-3&4).
- In Figure C.5-5, the stress-strain curve for samples C-5 had, unexpectedly, stopped, as the moving head started applying the force after moving 1 or 2 mm without applying any load.
- The provided Stress-strain curves, in Figure C.5-5, are calculated using the measured dimensions of each sample as specified in Table 4.1-1. Table C.5-1 shows calculated Stresses and strains of all four samples.

Table C.5-1 Stresses at different specified strain for each of the four samples, calculated based on the measured dimensions of each sample.

<table>
<thead>
<tr>
<th>Stress (MPa)</th>
<th>Strain:0.05</th>
<th>Strain:0.10</th>
<th>Strain:0.15</th>
<th>Strain:0.2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stress: A -5</td>
<td>8.500</td>
<td>23.993</td>
<td>41.548</td>
<td>64.480</td>
</tr>
<tr>
<td>Stress: B -5</td>
<td>7.948</td>
<td>23.744</td>
<td>41.663</td>
<td>65.398</td>
</tr>
<tr>
<td>Stress: C -5</td>
<td>7.063</td>
<td>22.141</td>
<td>38.548</td>
<td>N/A</td>
</tr>
<tr>
<td>Stress: D -5</td>
<td>7.201</td>
<td>22.189</td>
<td>40.058</td>
<td>63.598</td>
</tr>
</tbody>
</table>
Specimen Dimensions:

Material Type: Cotton Duck Pad
Number of Specimens: 4
Test Type: Compression
Loading Rate: 400 mm/min

Descriptive notes:
- The specimen was cut into the specified dimension using a band saw (see Figure C.6-1).
- The specimen was centered between the two LVDTs to maintain an accurate deflection recording.
- The loading rate was specified using the computer program associated with the machine.
- The data from the load cell and the 2 LVDTs were recorded using the same computer program.
- The specimen was centered inside the UCTM to maintain a well distributed load (see Figure C.6-2).
- The LVDTs are mounted on the steel plate that was used to distribute the load to the specimen.
- The machine is also equipped with a load cell with a capacity of 250 KN.
- A maximum Load of 160 KN was set for the moving head to prevent overloading.

Strain Rate and Stress–Strain Plots Sample A-6:

Date of test: December 16th, 2016.
Testing machine: Universal compression testing machine (UCTM).

Experimental Set up:

Non-moving bottom
Moving head
Steel plate
2 LVDTs

Figure C.6-1: Specimen dimensions

Figure C.6-2: (UCTM) set up.

Figure C.6-3: Strain rates from 2 LVDT and their average for sample A-6
Figure C.6-4: Stress-strain curves for 2 LVDTs and their average for sample A-6
Stress-strain curve for all four samples and average:

Figure C.6-5: The Stress-strain curves of the four samples tested under 400mm/min loading rate and their average (S-6-Average)

Observations and Discussion:
- The stiffness of the rubber increases while the applied load increases.
- The strain rate, shown as slope in Figure C.6-3, was relatively constant over the duration of the test due to the constant specified displacement base load rate. The average LVDT calculated strain rate = 171.7×10^{-3} s^{-1}.
- An average displacement was calculated from the data provided by the two LVDTs (see Figures C.6-3&4).
- In Figure C.6-5, the stress-strain curve for did not stop as previous tests, due to changing the limit loading 160 KN instead of having a maximum head displacement equal to 9 mm.
- The provided Stress-strain curves, in Figure C.6-5, are calculated using the measured dimensions of each sample as specified in Table 4.1-1. Table C.6-1 shows calculated Stresses and strains of all four samples.

Table C.6-1 Stresses at different specified strain for each of the four samples, calculated based on the measured dimensions of each sample.

<table>
<thead>
<tr>
<th>Stress (MPa)</th>
<th>Strain:0.05</th>
<th>Strain:0.10</th>
<th>Strain:0.15</th>
<th>Strain:0.23</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stress: A -6</td>
<td>8.520</td>
<td>24.651</td>
<td>42.480</td>
<td>84.416</td>
</tr>
<tr>
<td>Stress: B -6</td>
<td>9.426</td>
<td>25.292</td>
<td>42.880</td>
<td>85.152</td>
</tr>
<tr>
<td>Stress: C -6</td>
<td>10.121</td>
<td>26.290</td>
<td>44.099</td>
<td>87.521</td>
</tr>
<tr>
<td>Stress: D -6</td>
<td>8.090</td>
<td>23.746</td>
<td>42.514</td>
<td>84.968</td>
</tr>
</tbody>
</table>
**C.7 Lab report 7**

**Specimen Dimensions:**

![Specimen dimensions](image)

**Material Type:** Cotton Duck Pad

**Number of Specimens:** 4

**Test Type:** Compression

**Loading Rate:** 400 mm/min

**Date of test:** December 16th, 2016.

**Testing machine:** Universal compression testing machine (UCTM).

**Experimental Set up:**

![Experimental set up](image)

**Descriptive notes:**

- The specimen was cut into the specified dimension using a band saw (see Figure C.7-1).
- The specimen was centered between the two LVDTs to maintain an accurate deflection recording.
- The loading rate was specified using the computer program associated with the machine.
- The data from the load cell and the 2 LVDTs were recorded using the same computer program.
- The specimen was centered inside the UCTM to maintain a well distributed load (see Figure C.7-2).
- The LVDTs are mounted on the steel plate that was used to distribute the load to the specimen.
- The machine is also equipped with a load cell with a capacity of 250 KN.
- A maximum displacement of 5 mm was set for the moving head to prevent overloading.

**Strain Rate and Stress-Strain Plots Sample A-7:**

![Strain rates and Stress-strain plots](image)

**Figure C.7-3:** Strain rates from 2 LVDT and their average for sample A-7

**Figure C.7-4:** Stress-strain curves for 2 LVDTs and their average for sample A-7
Observations and Discussion:
- The stiffness of the rubber increases while the applied load increases.
- The strain rate, shown as slope in Figure C.7-3, was relatively constant over the duration of the test due to the constant specified displacement base load rate. The average LVDT calculated strain rate = 119.6×10⁻³ s⁻¹.
- An average displacement was calculated from the data provided by the two LVDTs (see Figures C.7-3&4).
- In Figure C.7-5, the stress-strain curve for did not stop as previous tests, due to changing the limit loading 150 KN instead of having a maximum head displacement equal to 5 mm.
- The provided Stress-strain curves, in Figure C.7-5, are calculated using the measured dimensions of each sample as specified in Table 4.1-1. Table C.7-1 shows calculated Stresses and strains of all four samples.

Table C.7-1 Stresses at different specified strain for each of the four samples, calculated based on the measured dimensions of each sample.

<table>
<thead>
<tr>
<th>Stress (MPa)</th>
<th>Strain: 0.05</th>
<th>Strain: 0.10</th>
<th>Strain: 0.20</th>
<th>Strain: 0.28</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stress: A -7</td>
<td>4.400</td>
<td>14.832</td>
<td>45.744</td>
<td>90.007</td>
</tr>
<tr>
<td>Stress: B -7</td>
<td>4.352</td>
<td>14.656</td>
<td>44.924</td>
<td>88.747</td>
</tr>
<tr>
<td>Stress: C -7</td>
<td>4.294</td>
<td>14.042</td>
<td>42.540</td>
<td>83.282</td>
</tr>
<tr>
<td>Stress: D -7</td>
<td>5.182</td>
<td>15.073</td>
<td>46.368</td>
<td>91.434</td>
</tr>
</tbody>
</table>
D Lab Reports for Impact Testing
D.1 Lab report 8

Specimen Dimensions:

![Image of specimen dimensions]

Figure C.7.1: Specimen Dimensions

Material Type: Cotton Duck Pad
Number of Specimens: 4
Test Type: Impact
Charpy Position: First

Date of test: March 1st, 2017.

Experimental Set up:

![Image of MCIM set up]

Figure D.1.1: MCIM set up.

Descriptive notes:

- The specimen was cut into the specified dimensions using a bandsaw (see Figure D.1-1).
- The specimen was centered inside the MCIM to maintain a well-distributed load (see Figure D.1-2).
- The specimen was speckled to facilitate the Digital Image Correlation (DIC) analysis, acquired by a high-speed camera located 5 meters from the specimen.
- The high-speed camera’s frame rate used was 5000 frames/s, which resulted in sufficient image numbers and pixels’ sizes for the DIC analysis.
- The 35.7 kg hammer is equipped with an accelerometer with a capacity of 5000g. The accelerometer has a calibration factor of 0.263 mV/V/g.
- The support is equipped with a load cell with a capacity of 320 KN. The load cell was calibrated using a universal compression testing machine.
- The data from the load cell and the accelerometer were recorded using a national instrument data acquisition system (50,000 samples/sec).

Table D.1-1 Stresses at different specified strain for each of the four samples calculated based on the measured dimensions of each sample from inbound response

<table>
<thead>
<tr>
<th>Stress (MPa)</th>
<th>Strain:0.050</th>
<th>Strain:0.100</th>
<th>Strain:0.150</th>
<th>Strain:0.175</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stress: A -8</td>
<td>4.620</td>
<td>27.430</td>
<td>61.296</td>
<td>77.045</td>
</tr>
<tr>
<td>Stress: B -8</td>
<td>2.926</td>
<td>18.907</td>
<td>49.812</td>
<td>66.162</td>
</tr>
<tr>
<td>Stress: C -8</td>
<td>4.972</td>
<td>22.832</td>
<td>53.710</td>
<td>70.748</td>
</tr>
<tr>
<td>Stress: D -8</td>
<td>7.321</td>
<td>23.330</td>
<td>49.940</td>
<td>63.322</td>
</tr>
</tbody>
</table>

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Figure D.1-2: Stress-Time plot for sample A-8 (including accelerometer data, load cell data, and a noise removed plot for accelerometer data)

Figure D.1-3: Strain-Time plot for sample A-8 and images acquired from the high-speed camera for DIC analysis
Observations and Discussion:

- The applied load was calculated using the acquired accelerometer data (acc), the hammer mass, and the calibration factors. The reaction force was also calculated using the calibration factor of the load cell data (LC).
- The calculated loads were plotted for every sample as shown in Figure D.1-3.
- It was noticed that during the impact event the acquired accelerometer data contained some noise. To remove this noise, an adjusted loading curve was calculated (see Figure D.1-3).
- It was also noticed that there was a delay of 1ms between the applied load (acc) and the reaction load (LC). This delay was due compression shock wave through the material and the steel plate.
- The stiffness of the rubber increased while the applied load increased.
- The loading strain rate changed over the impact time shown as parabola in Figure D.1-4. The data provided in the figure was acquired using DIC analysis. The average calculated strain rate based on maximum strain and the time taken to reach this maximum for the four samples = 119.19 s^{-1}.
- The provided Stress-Strain curves, in Figure D.1-5, were calculated using the measured dimensions of each sample as specified in Table 4.2-1. Table D.1-1 shows calculated stresses of all four samples at different specified strains.

Figure D.1-4: The Stress-Strain curves of the four samples tested under impact (first hammer position) and their average (S-8-Average)
D.2 Lab report 9

Specimen Dimensions:

![Specimen Dimensions](image)

Figure D.1-5: Specimen Dimensions

Material Type: Cotton Duck Pad
Number of Specimens: 4
Test Type: Impact
Charpy Position: Second

Descriptive notes:
- The specimen was cut into the specified dimensions using a bandsaw (see Figure D.2-1).
- The specimen was centered inside the MCIM to maintain a well-distributed load (see Figure D.2-2).
- The specimen was speckled to facilitate Digital Image Correlation (DIC) analysis, acquired by a high-speed camera located 5 meters from the specimen.
- The high-speed camera’s frame rate used was 5000 frames/s, which resulted in sufficient image numbers and pixels’ sizes for the DIC analysis.
- The 35.7 kg hammer is equipped with an accelerometer with a capacity of 5000g. The accelerometer has a calibration factor of 0.263 mV/V/g.
- The support is equipped with a load cell with a capacity of 320 KN. The load cell was calibrated using a universal compression testing machine.
- The data from the load cell and the accelerometer were recorded using a national instrument data acquisition system (50,000 samples/sec).

Date of test: March 1st, 2017.

Experimental Set up:

![Experimental Set up](image)

Figure D.2-1: MCIM set up.

Table D.2-1 Stresses at different specified strain for each of the four samples calculated based on the measured dimensions of each sample from inbound response

<table>
<thead>
<tr>
<th>Stress (MPa)</th>
<th>Strain:0.05</th>
<th>Strain:0.10</th>
<th>Strain:0.15</th>
<th>Strain:0.25</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stress: A -9</td>
<td>5.507</td>
<td>14.744</td>
<td>36.635</td>
<td>104.700</td>
</tr>
<tr>
<td>Stress: B -9</td>
<td>4.734</td>
<td>16.475</td>
<td>35.153</td>
<td>97.975</td>
</tr>
<tr>
<td>Stress: C -9</td>
<td>6.754</td>
<td>24.597</td>
<td>53.146</td>
<td>135.001</td>
</tr>
<tr>
<td>Stress: D -9</td>
<td>7.875</td>
<td>21.881</td>
<td>46.153</td>
<td>119.927</td>
</tr>
</tbody>
</table>
**Stress-Time Plot for Sample A-9:**

![Stress-Time Plot](image1)

*Figure D.2-2: Stress-Time plot for sample A-9 (including accelerometer data, load cell data, and a noise removed plot for accelerometer data)*

**Strain-Time Plot for Sample A-9 and camera images:**

![Strain-Time Plot](image2)

*Figure D.2-3: Strain-Time plot for sample A-9 and image acquired by the high-speed camera for DIC analysis*
Observations and Discussion:

- The applied load was calculated using the acquired accelerometer data (acc), the hammer mass, and the calibration factors. The reaction force was also calculated using the calibration factor of the load cell data (LC).
- The calculated loads were plotted for every sample as shown in Figure D.2-3.
- It was noticed that during the impact event the acquired accelerometer data contained some noise. To remove this noise, an adjusted loading curve was calculated (see Figure D.2-3).
- It was also noticed that there was a delay of 1ms between the applied load (acc) and the reaction load (LC). This delay was due compression shock wave through the material and the steel plate.
- The stiffness of the rubber increased while the applied load increased.
- The loading strain rate changed over the impact time shown as parabola in Figure D.2-4. The data provided in the figure was acquired using DIC analysis. The average calculated strain rate based on maximum strain and the time taken to reach this maximum for the four samples = 193.74 s⁻¹.
- The provided Stress-Strain curves, in Figure D.2-5, were calculated using the measured dimensions of each sample as specified in Table 4.2-1. Table D.2-1 shows calculated stresses of all four samples at different specified strains.

Figure D.2-4: The Stress-Strain curves of the four samples tested under impact (first hammer position) and their average (S-9-Average)
Date of test: March 3rd, 2017.


Specimen Dimensions:

![Figure D.3-1: Specimen Dimensions](image)

Material Type: Cotton Duck Pad
Number of Specimens: 4
Test Type: Impact
Charpy Position: First

Descriptive notes:

- The specimen was cut into the specified dimensions using a bandsaw (see Figure D.3-1).
- The specimen was centered inside the MCIM to maintain a well-distributed load (see Figure D.3-2).
- The specimen was speckled to facilitate the Digital Image Correlation (DIC) analysis, acquired by a high-speed camera located 5 meters from the specimen.
- The high-speed camera’s frame rate used was 5000 frames/s, which resulted in sufficient image numbers and pixels’ sizes for the DIC analysis.
- The 35.7 kg hammer is equipped with an accelerometer with a capacity of 5000g. The accelerometer has a calibration factor of 0.263 mV/V/g.
- The support is equipped with a load cell with a capacity of 320 KN. The load cell was calibrated using a universal compression testing machine.
- The data from the load cell and the accelerometer were recorded using a national instrument data acquisition system (50,000 samples/sec).

Table D.3-1 Stresses at different specified strain for each of the four samples calculated based on the measured dimensions of each sample from inbound response

<table>
<thead>
<tr>
<th>Stress (MPa)</th>
<th>Strain:0.050</th>
<th>Strain:0.100</th>
<th>Strain:0.150</th>
<th>Strain:0.175</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stress: A -10</td>
<td>4.202</td>
<td>17.048</td>
<td>37.046</td>
<td>48.589</td>
</tr>
<tr>
<td>Stress: B -10</td>
<td>2.279</td>
<td>15.624</td>
<td>37.715</td>
<td>50.440</td>
</tr>
<tr>
<td>Stress: C -10</td>
<td>2.422</td>
<td>14.304</td>
<td>37.908</td>
<td>51.284</td>
</tr>
<tr>
<td>Stress: D -10</td>
<td>4.605</td>
<td>16.699</td>
<td>35.2501</td>
<td>46.579</td>
</tr>
</tbody>
</table>
Figure D.3-3: Stress-Time plot for sample A-10 (including accelerometer data, load cell data, and a noise removed plot for accelerometer data)

Figure D.3-4: Strain-Time plot for sample A-10 and images acquired by the high-speed camera for DIC analysis
Observations and Discussion:
- The applied load was calculated using the acquired accelerometer data (acc), the hammer mass, and the calibration factors. The reaction force was also calculated using the calibration factor of the load cell data (LC).
- The calculated loads were plotted for every sample as shown in Figure D.3-3.
- It was noticed that during the impact event the acquired accelerometer data contained some noise. To remove this noise, an adjusted loading curve was calculated (see Figure D.3-3).
- It was also noticed that there was a delay of 1ms between the applied load (acc) and the reaction load (LC). This delay was due compression shock wave through the material and the steel plate.
- The stiffness of the rubber increased while the applied load increased.
- The loading strain rate changed over the impact time shown as parabola in Figure D.3-4. The data provided in the figure was acquired using DIC analysis. The average calculated strain rate based on maximum strain and the time taken to reach this maximum for the four samples = 79.77 s⁻¹.
- The provided Stress-Strain curves, in Figure D.3-5, were calculated using the measured dimensions of each sample as specified in Table 4.2-1 Table D.3-1 shows calculated stresses of all four samples at different specified strains.

![Stress-Strain curves for all four samples and their average](image-url)
Date of test: March 3rd, 2017.

Specimen Dimensions:

![Image](image1.png)
Figure D.4-1: Specimen Dimensions

Material Type: Cotton Duck Pad
Number of Specimens: 4
Test Type: Impact
Charpy Position: Second

Experimental Set up:

![Image](image2.png)
Figure D.4-2: MCIM set up.

Descriptive notes:

- The specimen was cut into the specified dimensions using a bandsaw (see Figure D.4-1).
- The specimen was centered inside the MCIM to maintain a well-distributed load (see Figure D.4-2).
- The specimen was speckled to facilitate the Digital Image Correlation (DIC) analysis, acquired by a high-speed camera located 5 meters from the specimen.
- The high-speed camera’s frame rate used was 5000 frames/s, which resulted in sufficient image numbers and pixels’ sizes for the DIC analysis.
- The 35.7 kg hammer is equipped with an accelerometer with a capacity of 5000g. The accelerometer has a calibration factor of 0.263 mV/V/g.
- The support is equipped with a load cell with a capacity of 320 KN. The load cell was calibrated using a universal compression testing machine.
- The data from the load cell and the accelerometer were recorded using a national instrument data acquisition system (50,000 samples/sec).

Table D.4-1 Stresses at different specified strain for each of the four samples calculated based on the measured dimensions of each sample from inbound response

<table>
<thead>
<tr>
<th>Stress (MPa)</th>
<th>Strain:0.05</th>
<th>Strain:0.10</th>
<th>Strain:0.15</th>
<th>Strain:0.23</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stress: A -11</td>
<td>2.939</td>
<td>15.088</td>
<td>38.632</td>
<td>90.837</td>
</tr>
<tr>
<td>Stress: B -11</td>
<td>6.623</td>
<td>23.472</td>
<td>47.582</td>
<td>101.435</td>
</tr>
<tr>
<td>Stress: C -11</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
</tr>
<tr>
<td>Stress: D -11</td>
<td>3.169</td>
<td>17.599</td>
<td>44.290</td>
<td>105.620</td>
</tr>
</tbody>
</table>
**Stress-Time Plot for Sample A-11:**

![Stress-Time Plot](image)

Figure D.4-3: Stress-Time plot for sample A-11 (including accelerometer data, load cell data, and a noise removed plot for accelerometer data)

**Strain-Time Plot for Sample A-11 and camera images:**

![Strain-Time Plot](image)

Figure D.4-4: Strain-Time plot for sample A-11 and image acquired by the high-speed camera for DIC analysis
Observations and Discussion:

- The applied load was calculated using the acquired accelerometer data (acc), the hammer mass, and the calibration factors. The reaction force was also calculated using the calibration factor of the load cell data (LC).
- The calculated loads were plotted for every sample as shown in Figure D.4-3.
- It was noticed that during the impact event the acquired accelerometer data contained some noise. To remove this noise, an adjusted loading curve was calculated (see Figure D.4-3).
- It was also noticed that there was a delay of 1ms between the applied load (acc) and the reaction load (LC). This delay was due compression shock wave through the material and the steel plate.
- The stiffness of the rubber increased while the applied load increased.
- The loading strain rate changed over the impact time shown as parabola in Figure D.4-4. The data provided in the figure was acquired using DIC analysis. The average calculated strain rate based on maximum strain and the time taken to reach this maximum for the four samples = 109.44 s\(^{-1}\).
- The provided Stress-Strain curves, in Figure D.4-5, were calculated using the measured dimensions of each sample as specified in Table 4.2-1 Table D.4-1 shows calculated stresses of the three samples at different specified strains.
- The result of sample C-11 was not recorded by the high-speed camera.
D.5 Lab report 12

Specimen Dimensions:

Material Type: Cotton Duck Pad
Number of Specimens: 4
Test Type: Impact
Charpy Position: First

Figure D.5-1: Specimen Dimensions

Date of test: March 3rd, 2017.

Experimental Set up:

• The specimen was cut into the specified dimensions using a bandsaw (see Figure D.5-1).
• The specimen was centered inside the MCIM to maintain a well-distributed load (see Figure D.5-2).
• The specimen was speckled to facilitate the Digital Image Correlation (DIC) analysis, acquired by a high-speed camera located 5 meters from the specimen.
• The high-speed camera’s frame rate used was 5000 frames/s, which resulted in sufficient image numbers and pixels’ sizes for the DIC analysis.
• The 35.7 kg hammer is equipped with an accelerometer with a capacity of 5000g. The accelerometer has a calibration factor of 0.263 mV/V/g.
• The support is equipped with a load cell with a capacity of 320 KN. The load cell was calibrated using a universal compression testing machine.
• The data from the load cell and the accelerometer were recorded using a national instrument data acquisition system (50,000 samples/sec).

Figure D.5-2: MCIM set up.

Table D.5-1 Stresses at different specified strain for each of the four samples calculated based on the measured dimensions of each sample from inbound response

<table>
<thead>
<tr>
<th>Stress (MPa)</th>
<th>Strain:0.05</th>
<th>Strain:0.10</th>
<th>Strain:0.12</th>
<th>Strain:0.14</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stress: B -12</td>
<td>3.988</td>
<td>19.659</td>
<td>28.066</td>
<td>36.170</td>
</tr>
<tr>
<td>Stress: C -12</td>
<td>6.653</td>
<td>24.362</td>
<td>31.805</td>
<td>38.347</td>
</tr>
<tr>
<td>Stress: D -12</td>
<td>4.882</td>
<td>20.251</td>
<td>28.056</td>
<td>36.417</td>
</tr>
</tbody>
</table>
Stress-Time Plot for Sample A-12:

Figure D.5-3: Stress-Time plot for sample A-12 (including accelerometer data, load cell data, and a noise removed plot for accelerometer data)

Strain-Time Plot for Sample A-12 and camera images:

Figure D.5-4: Strain-Time plot for sample A-12 and image acquired by the high-speed camera for DIC analysis
Observations and Discussion:

- The applied load was calculated using the acquired accelerometer data (acc), the hammer mass, and the calibration factors. The reaction force was also calculated using the calibration factor of the load cell data (LC).
- The calculated loads were plotted for every sample as shown in Figure D.5-3.
- It was noticed that during the impact event the acquired accelerometer data contained some noise. To remove this noise, an adjusted loading curve was calculated (see Figure D.5-3).
- It was also noticed that there was a delay of 1ms between the applied load (acc) and the reaction load (LC). This delay was due compression shock wave through the material and the steel plate.
- The stiffness of the rubber increased while the applied load increased.
- The loading strain rate changed over the impact time shown as parabola in Figure D.5-4. The data provided in the figure was acquired using DIC analysis. The average calculated strain rate based on maximum strain and the time taken to reach this maximum for the four samples = 41.955 s\(^{-1}\).
- The provided Stress-Strain curves, in Figure D.5-5, were calculated using the measured dimensions of each sample as specified in Table 4.2-1. Table D.5-1 shows calculated stresses of all four samples at different specified strains.

**Figure D.5-5:** The Stress-Strain curves of the four samples tested under impact (first hammer position) and their average (S-12-Average)
Date of test: March 3rd, 2017.


Specimen Dimensions:

![Specimen Dimensions](image)

Figure D.6-1: Specimen Dimensions

- Material Type: Cotton Duck Pad
- Number of Specimens: 4
- Test Type: Impact
- Charpy Position: Second

Experimental Set up:

![MCIM Set up](image)

Figure D.6-2: MCIM set up.

Descriptive notes:

- The specimen was cut into the specified dimensions using a bandsaw (see Figure D.6-1).
- The specimen was centered inside the MCIM to maintain a well-distributed load (see Figure D.6-2).
- The specimen was speckled to facilitate the Digital Image Correlation (DIC) analysis, acquired by a high-speed camera located 5 meters from the specimen.
- The high-speed camera’s frame rate used was 5000 frames/s, which resulted in sufficient image numbers and pixels’ sizes for the DIC analysis.
- The 35.7 kg hammer is equipped with an accelerometer with a capacity of 5000g. The accelerometer has a calibration factor of 0.263 mV/V/g.
- The support is equipped with a load cell with a capacity of 320 KN. The load cell was calibrated using a universal compression testing machine.
- The data from the load cell and the accelerometer were recorded using a national instrument data acquisition system (50,000 samples/sec).

Table D.6-1 Stresses at different specified strain for each of the four samples calculated based on the measured dimensions of each sample from inbound response

<table>
<thead>
<tr>
<th>Stress (MPa)</th>
<th>Strain:0.050</th>
<th>Strain:0.10</th>
<th>Strain:0.15</th>
<th>Strain:0.20</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stress: A -13</td>
<td>6.268</td>
<td>21.992</td>
<td>43.439</td>
<td>68.400</td>
</tr>
<tr>
<td>Stress: B -13</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
</tr>
<tr>
<td>Stress: C -13</td>
<td>6.339</td>
<td>24.291</td>
<td>46.110</td>
<td>67.596</td>
</tr>
<tr>
<td>Stress: D -13</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
</tr>
</tbody>
</table>
Figure D.6-3: Stress-Time plot for sample A-13 (including accelerometer data, load cell data, and a noise removed plot for accelerometer data)

Figure D.6-4: Strain-Time plot for sample A-13 and image acquired by the high-speed camera for DIC analysis
Observations and Discussion:

- The applied load was calculated using the acquired accelerometer data (acc), the hammer mass, and the calibration factors. The reaction force was also calculated using the calibration factor of the load cell data (LC).
- The calculated loads were plotted for every sample as shown in Figure D.6-3.
- It was noticed that during the impact event the acquired accelerometer data contained some noise. To remove this noise, an adjusted loading curve was calculated (see Figure D.6-3).
- It was also noticed that there was a delay of 1ms between the applied load (acc) and the reaction load (LC). This delay was due compression shock wave through the material and the steel plate.
- The stiffness of the rubber increased while the applied load increased.
- The loading strain rate changed over the impact time shown as parabola in Figure D.6-4. The data provided in the figure was acquired using DIC analysis. The average calculated strain rate based on maximum strain and the time taken to reach this maximum for the four samples = 64.006 s⁻¹.
- The provided Stress-Strain curves, in Figure D.6-5, were calculated using the measured dimensions of each sample as specified in Table 4.2-1. Table D.6-1 shows calculated stresses of all four samples at different specified strains.
- The result of sample B-12 and D-12 were not recorded by the high-speed camera. The accelerometer data for sample A-12 experienced some unexpected noise at the end of the test.
CURRICULUM VITAE

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Conference Presentations: NONE