

Assessment of the Effects of Belt Conveying Parameters on the Performance of Unidirectional Idlers

Bryan Moore

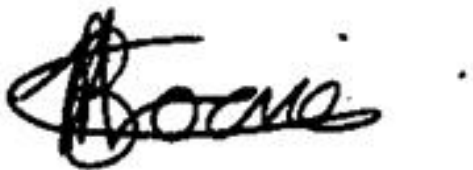
A dissertation submitted to the Faculty of Engineering and the Built Environment, University of the Witwatersrand, Johannesburg, in fulfilment of the requirements for the degree of Master of Science in Mechanical Engineering.

Johannesburg, August 2020

Declaration

I declare that this dissertation is my own, unaided work, except where otherwise acknowledged. It is being submitted for the degree of Master of Science in Mechanical Engineering in the University of the Witwatersrand, Johannesburg. It has not been submitted before for any degree or examination at any other university.

Signed this 13th day of August 2020

A handwritten signature in black ink, appearing to read 'Bryan Moore', with a large, stylized initial 'B'.

Bryan Moore

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Abstract

Despite their popularity as an inclined conveyor belt runaway preventing device, very little research into the performance of unidirectional idlers currently exists. Considering the catastrophic repercussions of a belt runaway, as well as the high cost of unidirectional idlers, the importance of correctly specifying the required quantity of these devices may be appreciated. As such, this study determined the relationship between significant belt conveying factors and the unidirectional idler resistive forces, so as to determine the proportion of idlers that are required to be unidirectional.

In order to realise this objective, a range of experiments was planned using Design of Experiment (DOE) techniques. These experiments were designed to determine the resistive forces that develop between unidirectional rollers and a conveyor belt being pulled over these rollers in the *reverse* direction (to represent a belt slipping down an incline). These experiments were executed using the unidirectional roller (UDR) Test Rig, which was used to vary the levels of inclination angle, troughing angle, idler pitch, roller type, roller diameter, belt tension and material load. Analysis of the subsequent results revealed that *material load*, *troughing angle* and *idler pitch* have the most significant effect on unidirectional idler braking. In addition, it was determined that although contact friction affects unidirectional idler performance, the presence of non-friction resistances (such as belt sag) strongly influence the resistive force at belt slip. To expand on these findings, an additional friction study of various roller and conveyor belt material pairs was conducted using a Jenike Direct Shear Tester (JDST). The findings from this study echoed those of the UDR Experiments – that HDPE rollers provide the least resistance to belt slip, and that belt wear over time results in the reduction of unidirectional idler braking ability. Based on the findings of these investigations, a conservative numerical method for specifying the required proportion of unidirectional idlers was derived.

Overall, the results indicate that the worst unidirectional idler performance case is that where a 900 mm wide empty conveyor belt is troughed to an angle of 35° and supported by idlers separated by a pitch of 1200 mm. Under these worst case conditions, it is recommended that HDPE and steel unidirectional idlers are *not specified* for inclination angles greater than 16° and 21° respectively.

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Nomenclature

Acronyms

CMA Conveyor Manufacturers Association of South Africa

COF Coefficient Of Friction

DOE Design Of Experiment

HDPE High Density Polyethylene

JDST Jenike Direct Shear Tester

PVC Polyvinyl Chloride

UDR Unidirectional Roller

Greek Symbols

α Angle of inclination [°]

β Troughing angle [°]

δ Angle of surcharge [°]

ϵ Unidirectional centre roller proportion factor [-]

η Fraction of rollers required to be unidirectional [-]

η_s Fraction of rollers required to be unidirectional (including a safety factor) [-]

$\eta_{c,s}$ Fraction of centre rollers required to be unidirectional (including a safety factor) [-]

$\eta_{w,s}$ Fraction of wing rollers required to be unidirectional (including a safety factor) [-]

κ Belt sag ratio [-]

λ Non-friction coefficient [-]

μ	Kinetic Friction coefficient [-]
ω	Normal coefficient [-]
ψ	Resistance coefficient [-]
ψ_c	Centre roller resistance coefficient [-]
ψ_w	Wing roller resistance coefficient [-]
ρ	Bulk density of the load [kg/m ³]
τ_{res}	Rotation resistance torque [N.mm]
θ	Angle of wrap [°]

Symbols

A	Cross-sectional area of the load on the belt [m ²]
a	Idler pitch [m]
A_c	Cross-sectional area of the load directly above the centre roller [m ²]
A_w	Cross-sectional area of the load directly above the wing rollers [m ²]
B	Belt width [m]
b	Usable width of the belt [m]
d	Roller diameter [mm]
e	Eccentric distance between roller axis and measured force F [mm]
F	Measured resistance to roller rotation force [N]
F_f	Friction force [N]
F_{res}	Resistance force [N]
g	Acceleration due to gravity, assumed to be 9.79 m/s ² in this work
K_S	Belt sag [%]
L	Length of the section of the belt that is inclined [m]
l_1	Roller face length [m]
M_{XY}	Interaction effect of factors XY

M_X	Main effect of factor X
N	Normal force [N]
n	Factor of safety [-]
N_c	Normal force acting at the centre roller [N]
N_w	Normal force acting at the wing rollers [N]
p	Roller-to-belt contact pressure [kPa]
q_b	Mass per metre of the conveyor belt [kg/m]
Q_m	Conveyor carrying capacity [kg/s]
q_m	Mass per metre of the material load [kg/m]
r	Roller radius [mm]
T	Belt tension [N]
t	Belt thickness [mm]
v	Belt velocity [m/s]
W	Belt and load combined weight per length of belting [N/m]
W_m	Cumulative weight of roller material sample block and loaded mass hanger [N]
$y_{\max}$	Maximum belt deflection [m]

Specialised Belt Conveying Terms

Head Pulley	The pulley at the head of the conveyor; where the conveyed material is usually discharged. This is often also a Drive Pulley.
Drive Pulley	A conveyor pulley powered by a motor which transmits tension to the belt.
Tail Pulley	The pulley at the tail of the conveyor; where the conveyed material is usually loaded.
Head-Driven	When the conveyor belt is powered by a motor connected to the head pulley.
Idler/Roller	Freely rotating cylinder which supports and guides the moving conveyor belt.
Idler Set	Consists of one or more freely rotating rollers.
Troughed Idler Set	Idler set with two or more inclined rollers used to increase the carrying capacity and prevent material spillage by bending the conveyor belt into a trough.
Three-Roll Idler Set	Idler Set containing three rollers.
Centre Roller	The middle roller in a 3 or 5 roll idler set.
Wing Rollers	The outer most rollers in an idler set which are angled to create the troughed profile.
Troughing Angle	The angle that the wing rollers make relative to the centre roller.
In-Line Idler Set	The axes of all of the rollers in an idler set are in the same transverse plane.
Offset Idler Set	The axis planes of the wing rollers are offset longitudinally from that of the centre roller.
Idler Pitch	The longitudinal distance between idler sets.
Belt Sag	The ratio of maximum vertical belt deflection (under expected load conditions) to idler pitch; expressed as a percentage.
Belt Splice	The location at which the two ends of the conveyor belt are joined together to form a continuous loop.
Rollback	An uncontrolled reversing motion of an inclined conveyor belt due to a loss of power to the drive pulley or a belt snap.
Runaway	A loss of control and subsequent acceleration of a conveyor belt down an incline.
Declined Conveyor	Conveyors that transport a load down an incline.
Backstop Mechanism	A drive pulley mounted device that inhibits reverse belt motion.
Belt Arrestor	A device that inhibits reverse belt motion in the event of a belt snap.
Belt Catcher	A type of belt arrestor that uses a pinching jaw mechanism to catch the belt in the event of a belt snap.
Unidirectional Idler	An idler that permits rotation in one direction only.

Specialised Design of Experiment Terms

Replicates	The number of times an experimental test is repeated to assess independent repeatability of the results.
Generator	A factor interaction whose high or low level treatment combinations are used to select which tests to run.
Interaction effect	Interactions between factors occur when the measured response corresponding to a particular factor differs according to the levels of other factors due to inherent dependencies between such variables.
Paired effect	A two-factor interaction effect.
Confounded	When two or more factor effects are estimated by the same combinations of observations. For example, factor A and interaction AB are confounded if their levels (+ or $-$) match at every test.
Resolution	An indication of the level of confounding present in a fractional factorial design.

Chapter 1 Introduction

1.1 Purpose of the Study

The purpose of this study is to investigate the individual and combined effects of various conveyor belt factors that influence the braking performance of unidirectional idlers; and to subsequently determine, through empirical evidence and numerical simulation, the suitability and quantity of unidirectional idlers required to hold a conveyor belt and material in place in the catastrophic event of a belt snap on an inclined conveyor.

1.2 Background and Motivation

The handling of bulk material is inherently dangerous and as such safety measures should be employed. One specific area of concern is that of inclined belt conveying systems which are particularly vulnerable to rollback and runaway, as well as uncontrolled motion of the conveyor belt and load in the event of a belt failure. Figure 1.1 illustrates a runaway condition caused by the conveyor belt snapping at the head pulley.

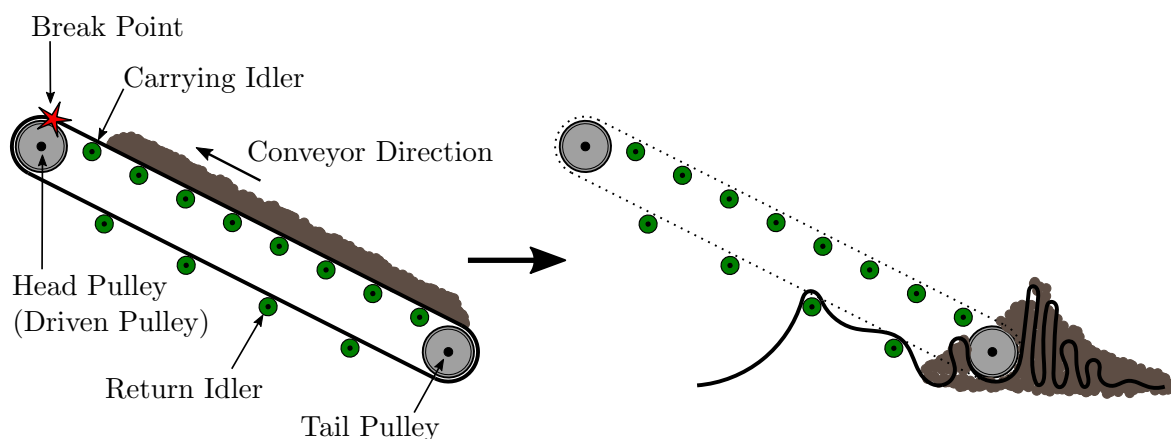


Figure 1.1: Schematic of a conveyor belt snapping at the head pulley; adapted from [1]

Without adequate safety precautions in place, belt runaways can lead to fatalities or injuries of personnel, damage to conveying equipment and loss of productivity [1]. In belt conveying, runaway refers to the loss of control and subsequent acceleration of the belt and load down an incline. To avoid this, the 2013 revision of South Africa’s *Mine Health and Safety Act 29 of 1996* requires that:

“The employer must take reasonably practicable measures to prevent persons from being injured by material or mineral falling from a conveyor belt installation, which measures must include the fitting and use of one or more devices to prevent run-back or run-on; when such conveyor belt installation is stopped” [2]

To prevent belt run-back, also referred to as rollback, inclined systems can be installed with a backstop mechanism which permits belt motion in the forward direction and prohibits it in the reverse direction. A backstop is generally mounted to the drive shaft of the motorised pulley and prevents runaway when the motor is not energised or when failures occur [3]. However, these devices do not guarantee the prevention of runaways. In addition, critically, in the catastrophic event of a belt snap, the backstop mechanism is unable to hold the entire belt in place and the portion of the loaded belt downhill of the break point will overcome the idler rotational resistances and accelerate down the incline [1]. In the case of a head-driven conveyor, the belt snap will usually occur at the head pulley where belt tension is at its highest (see Figure 1.1). Although failures are most common in this location, the belt can snap anywhere along its length and is particularly vulnerable to failure at the splice. The splice is the joint between two ends of a conveyor belt [4].

To prevent runaways from happening as a result of this kind of belt failure, other safety devices may be installed. A useful class of safety device in these scenarios is the conveyor belt arrestor – a mechanical device that limits or inhibits reverse belt motion in the event of a belt snap [5]. Belt arrestors are installed in conveyor systems near and around the area of maximum belt tension so as to lock a snapped belt in place before it can accelerate down the incline [1]. Commonly used belt arrestors include unidirectional idlers and belt catchers. Unidirectional idlers use single-direction bearings in conveyor rollers and belt catchers use a pinching jaw mechanism to catch and hold a runaway belt in place. Unidirectional idlers (also referred to as anti-rollback idlers) have become quite popular in the South African hard and soft rock mining industries in recent years and are the focus of this research. Figure 1.2 shows a unidirectional mechanism incorporated into a traditional roller.

Although popular, very little unidirectional idler research exists. This despite the higher cost of unidirectional idlers as compared to a conventional idlers. The relative sale prices provided by a local idler manufacturer showed that unidirectional rollers may cost between 2.1 and 2.9 times that of equivalent conventional rollers [7]. For specific examples of this difference in

cost, refer to Table 1.1. When the high costs of these devices, and the added rotational resistance that they impart to the system are considered, the importance of correctly specifying the required quantity of unidirectional idlers may be appreciated.

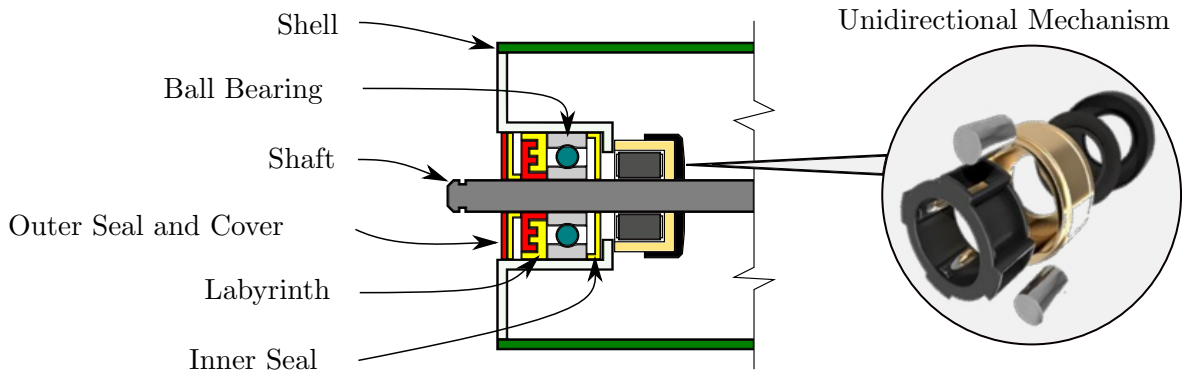


Figure 1.2: Cross-section of a unidirectional idler; adapted from [6]

Table 1.1: Relative cost of unidirectional idlers compared to standard idlers [7]

Diameter (mm)	Face Length (mm)	Material	Cost	
			Standard Idler	Unidirectional Idler
127	340	Steel	x	$2.9 x$
127	340	HDPE	x	$2.5 x$
127	450	Steel	x	$2.7 x$
127	450	HDPE	x	$2.1 x$
152	340	Steel	x	$2.7 x$
152	340	HDPE	x	$2.1 x$
152	450	Steel	x	$2.6 x$
152	450	HDPE	x	$2.3 x$

As such, the industry intended to perform some unidirectional idler focused research. The Conveyor Manufacturers Association of South Africa (CMA) commissioned a test rig to be built that could measure the resistance to motion between an inclined conveyor belt and a stationary idler. The desired outcome was anticipated to be a matrix of resistance factors corresponding to various operating conditions of inclined, three-roll idler set conveyor systems (see Section 2.2). The CMA proposed that experiments be run to investigate the effects of the following parameters on the resistance to motion between a conveyor belt and a stationary idler:

- | | |
|----------------------|-----------------------|
| i. inclination angle | vi. belt type |
| ii. troughing angle | vii. roller type |
| iii. belt sag | viii. roller diameter |
| iv. idler pitch | ix. belt tension |
| v. belt width | x. load |

Before the commencement of this research venture, there was no clear understanding of exactly how this list of parameters affected the performance of unidirectional idlers, nor was there any consensus regarding which of these had the most significant influence. Therefore research was to be conducted to allow for a design matrix to be generated that could be used to accurately determine the required quantity of unidirectional idlers; thus preventing the over/under-specification of these safety devices. Although varying all the proposed factors at multiple levels would create a large set of physical data to draw from, the greater the number of factors and levels, the greater the number of experiments required to be run. Therefore it was important that appropriate experimental design techniques were used to limit the number of test runs without losing the ability to accurately estimate the main and paired effects of the parameters being varied.

Not only does a well designed experimental procedure allow for conclusions to be drawn regarding which parameters have the greatest influence on unidirectional idler braking mechanisms, but the empirical data collected could also be useful in supplementing numerical simulations that assess the effects of belt failure under varying operating conditions. The motivation behind deriving a basic mechanical model to numerically simulate different belt failure scenarios was to extrapolate the established data trends into a broader investigation; thus generating greater insight and understanding. A simulation, validated and supplemented by the empirical work, allows for a much larger quantity of situations to be explored at a fraction of the time taken to conduct heavy duty physical experiments.

1.3 Problem Statement

A focussed test programme needs to be undertaken to assess the effects of a wide range of parameters associated with conveyor belts, in relation to the prevention of belt runaway by means of unidirectional idlers in the event of a belt failure. This would enable a practical, cost-effective approach to the quantity specification of unidirectional idlers for particular applications, and would allow for a greater understanding of the mechanics of the system by means of numerical simulation of various belt failure scenarios.

1.4 Research Questions and Objectives

Following on from the delineation of the problem statement, two research questions were formulated. In order to answer these important questions, a number of investigation objectives were also conceived. The following sub-sections list these research questions and objectives.

1.4.1 Research Questions

1. Which inclined belt conveying parameters significantly influence the resistive friction force developed between unidirectional idlers and a conveyor belt in the event of a belt snap or loss of drive pulley power, and how do these significant factors affect this friction force?
2. Is it possible through numerical modelling of various failure scenarios to predict the effect of different quantities of unidirectional idlers, in different configurations for various conveyor system geometries?

1.4.2 Objectives

1. Conduct experiments to ascertain which belt conveying factors (and interactions thereof) have the most significant effect on the resistance to conveyor belt slip over locked unidirectional idlers.
2. Determine the relationship between the significant belt conveying factors and the unidirectional idler resistive forces.
3. Derive a numerical model and associated factor matrix which may be used to predict the belt slip behaviour under different operating conditions, and which may be used to specify the quantity of unidirectional idlers required to prevent a belt runaway in a particular inclined belt application.

Chapter 2 Literature Review

2.1 Belt Conveying Overview

Belt conveyors play a pivotal role in the handling of bulk materials and are favoured due to their reliability, versatility, cost-effectiveness and simplicity of operation. Conveyors can transport large quantities of a wide variety of materials over long distances, up steep inclines, around corners, over undulating terrain, across bodies of water, and underground [8]. In addition, long distance conveyor belt systems typically run with the lowest transport, maintenance, energy and labour costs per tonne of material when compared with other materials-handling systems [9]. Some of the standard components of a gravity take-up belt conveying system are depicted in Figure 2.1.

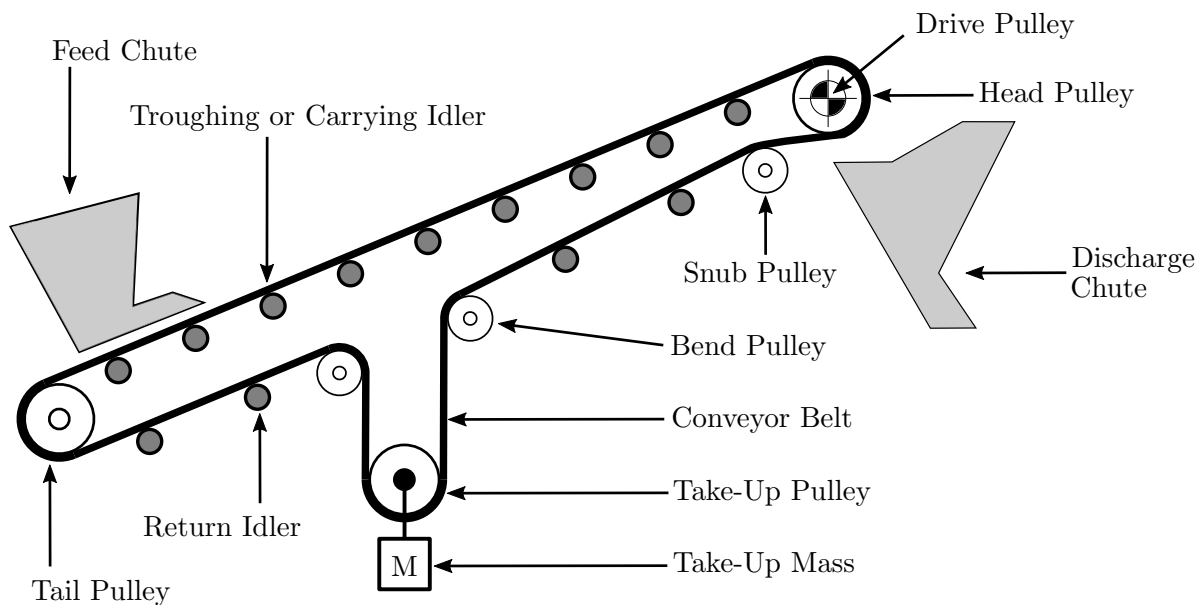


Figure 2.1: Annotated diagram of an inclined conveyor belt; adapted from [9]

A conveyor system is made up of a rubber belt wrapped around two terminal pulleys where the two ends are joined together (spliced) so that a continuous loop is formed. The tail pulley is located at the tail end of the conveyor where material is generally loaded onto the belt via a feed chute. The head pulley, located at the head of the system, is where material is usually offloaded into a discharge chute. In most conveyors, the head pulley is powered by a motor (i.e. head-driven). However, drive motors can be located at any point along the conveyor and multiple motors may be installed in long or heavily-loaded systems.

Between the terminal pulleys, the belt is supported and guided by flat or troughing idler sets on the carrying side and return idler sets on the bottom/return side. Idler sets consist of one or more freely rotating pulleys called idlers/rollers that are mounted in frames and supported by structures called stringers. Other than the terminal pulleys, conveyors can also make use of bend, take-up and snub pulleys. Bend pulleys direct the belt into a take-up pulley which is usually attached to a counterweight or winch that ensures adequate tensioning of the belt. A snub pulley is located just after the drive pulley on the return side of the conveyor and increases the angle of belt wrap around the pulley, thus decreasing the required torque to drive the belt and helping to prevent slip at the head pulley [9].

In terms of safety precautions, belt conveying systems may utilise such anti-runback devices as backstops, brakes, belt catchers and unidirectional idlers (as introduced in Section 1.2). Backstops make use of a unidirectional clutch that prevents a reversing motion by jamming sprags between the inner and outer races [1]. Backstops are favourable in part because their actuation is instant and automatic – there is no need for a control system. However, these devices are not suitable to be installed in reversing or declined conveyors. In such applications, back-stops may be replaced by brakes. Not only can drive pulley brakes prevent run-back by locking descending conveyor belts in place, but they can also be used together with backstops on inclined conveying systems to provide a back-up safety precaution [10]. Brakes are also desirable because they are tested during starting and stopping procedures.

The major drawback of drive pulley mounted backstops and brakes is that they cannot prevent runaway in the event of a belt snap. In order to combat this, belt arrestors (such as belt catchers and unidirectional idlers) may be installed. Belt catchers are able to stop both the carrying and return sides of the conveyor belt in the event of a belt snap. Belt catchers are fitted with counterweights that are released by the loss of tension experienced in a failed belt, thus closing the jaws of the device to trap both sides of the belt [1]. The major disadvantages of belt catchers are their high cost and the fact that they cannot arrest belt motion should failure occur away from the driven pulley around which these devices are normally installed.

Unidirectional idlers, on the other hand, are able to automatically prevent belt runaway regardless of where failure occurs on the carrying side of the belt. Figure 2.2a demonstrates that when the drive pulley induced belt tension pulls the belt up an incline, the unidirectional idlers rotate freely. In the event of a belt snap, the unidirectional mechanism prevents roller rotation in the reverse direction and the loaded belt is kept from accelerating down the incline, provided the belt remains in contact with the roller.

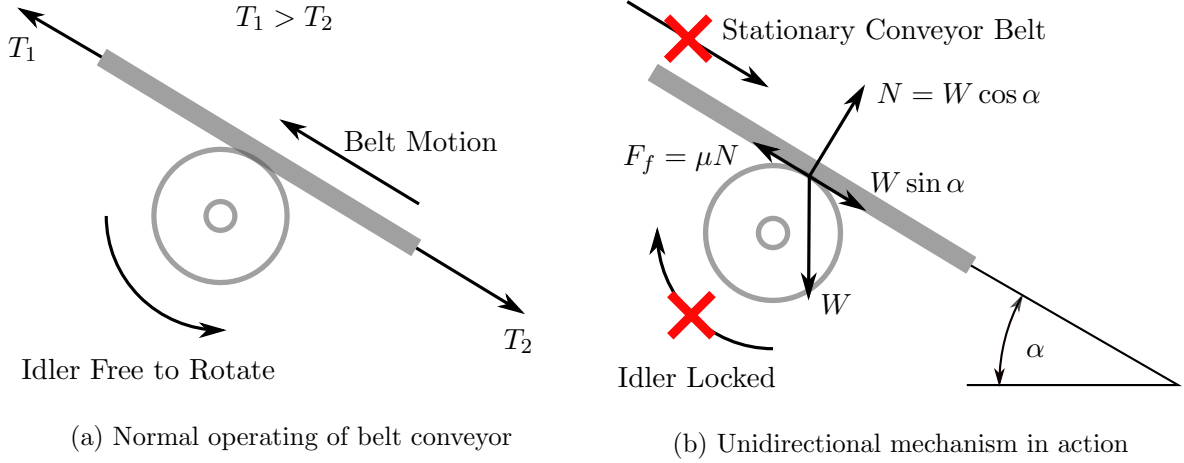


Figure 2.2: Schematic diagrams illustrating the functioning of unidirectional idlers

Figure 2.2b shows a simplified free-body diagram of a unidirectional idler in action. From this figure it may be seen that downhill belt motion will only occur if the component of the belt and material weight acting parallel to the inclined belt is greater than the friction acting in the opposite direction^a. In other words, downward belt motion may only occur by way of belt slip and the likelihood of this happening may be eliminated by specifying a sufficient quantity of unidirectional devices. Although effective, these devices generally add rotational resistance to the conveyor idlers and are unable to prevent runaway of the return side of the belt or of declined conveyor systems [10].

Overall, the different belt stopping devices have varying strengths and weaknesses and, as such, there is no single device that is better than, or that should be used in place of, another. Instead, combinations of these devices should be employed to ensure that the conveyor system is protected against runaway.

^aIn reality, however, the runaway resisting force is not likely as a result of friction alone. Other possible sources of resistance include belt sag, indentation and tension effects (see Section 2.2).

2.2 Belt Conveying Parameters

As mentioned in Section 1.2, inclined conveyors require special attention when it comes to safe operating procedures and precautions. This is of particular concern when considering that some South African mines still use man-riding belts to transport workers underground, and carry workers and ore to the surface [11]. These types of conveyors need stringent safety mechanisms in place to ensure that belt failures don't result in the uncontrolled falling of workers down the incline, causing injuries and/or deaths. Considering unidirectional idlers as the solution to these safety concerns, the parameters that possibly influence these safety devices will be explored in the following subsections.

2.2.1 Inclination Angle

Of paramount importance is the angle of inclination, α , as illustrated in Figure 2.3. The steeper this angle is, the greater the quantity of unidirectional idlers needed to hold the belt in place will be. This is because the component of combined load and belting weight acting down the incline increases with the angle of inclination, thus requiring a greater braking force to hold the loaded belt in place. Furthermore, the angle of inclination also affects the carrying capacity, Q_m [kg/s], of the conveyor according to [8] :

$$Q_m = \rho A v \cos \alpha \quad (2.1)$$

where ρ is the bulk density of the load [kg/m³], A is the cross-sectional area of the load on the belt [m²] and v is the velocity of the belt [m/s].

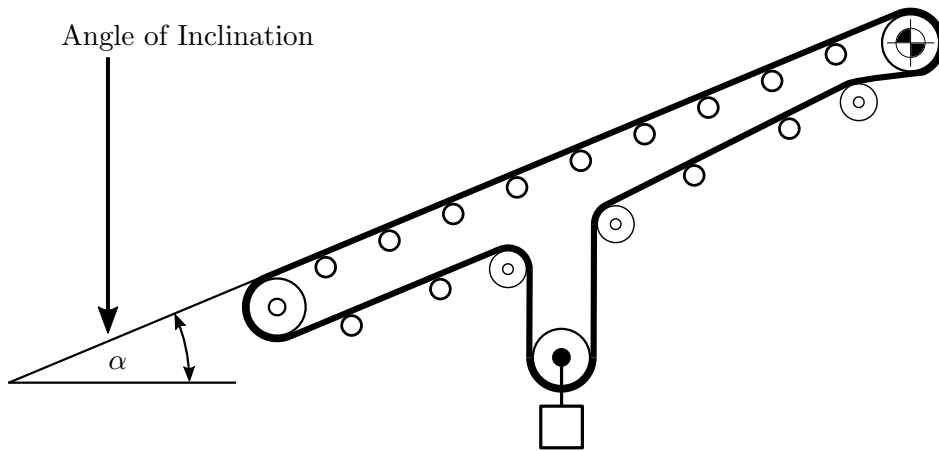


Figure 2.3: Inclined belt conveyor

Another geometric factor that affects carrying capacity is the cross-sectional area that the loaded material makes with the belt. Beyond simply increasing the belt width, B , this area can be enlarged by increasing the troughing angle.

2.2.2 Troughing Angle

In a three-roll idler set, the troughing angle, β , is the angle between the axes of one of the inclined wing rollers and the horizontal centre roller (as shown in Figure 2.4) . Although troughed idler sets may use up to five rollers, most systems use three. Therefore, only three-roll idler sets will be considered in this research. The rollers of a three-roll system are usually all the same length and may either be in-line (the axes of all rollers are in the same transverse plane) or offset (the centre roller axis plane lags or leads that of the wing rollers) [12]. The most common configuration in the South African mining industry is the belt-friendly offset idler and shall therefore be the focus of this research.

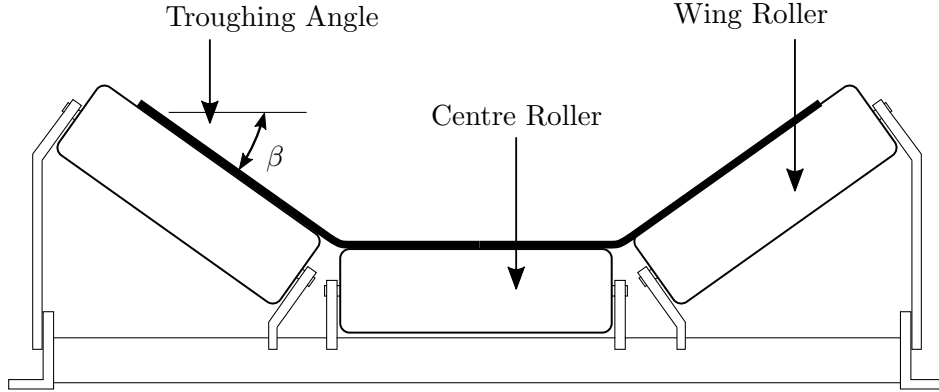


Figure 2.4: Three-roll troughed idler system

In terms of its effects on the braking capability of unidirectional idlers, different troughing angles result in different wing roller contact pressures. Previous research has shown that the reaction forces experienced by the wing rollers troughed at steeper angles are greater than those with shallower troughing angles. Previous research has shown that the wing roller normal force is 1.2 and 1.9 times greater than the combined weight of the material and belt directly above the wing roller for 30° and 45° troughed idlers respectively [13]. The contact pressure of a centre roller may be approximated by the weight of the volume of material directly above it.

Despite the contact pressure multipliers observed in wing roller sections, the majority of the conveyed material exists in the region above the centre rollers and as such, normal forces on these rollers tend to be significantly larger. A recent study by Liu et al. [14] recorded the

total percentage of normal force on the centre belt section of a 35° troughed three-roll idler system to vary from 66.2% to 74.2%. Altogether, the relationship between troughing angle and wing roller normal force extends to the braking capability of unidirectional rollers.

Although the centre rollers are prioritised in the installation of unidirectional devices, in instances where installing these devices in all of the centre rollers is insufficient to prevent runaway, unidirectional wing rollers are also installed. Since the friction force (F_f) between two objects in contact is given by the product of the normal force (N) and the coefficient of friction (μ) [15], a greater normal force experienced by a wing roller will improve the friction braking capability of the unidirectional device and therefore fewer of these devices will need to be used.

2.2.3 Belt Sag, Idler Pitch and Belt Width

As mentioned in the footnote (a) in Section 2.1, the friction between the belt and idler contact faces is not the only factor affecting the resistance to motion of a conveyor belt moving over static idlers. Instead, a portion of the force required to pull a conveyor belt down a slope, where the belt is supported on unidirectional idlers, comes from the inherent sag of the loaded belt that must be lifted up and over each idler. Belt sag, K_S , is a ratio of maximum belt deflection, $y_{\max}$ [m], to idler pitch, a [m], and is typically expressed as a percentage [8]. The maximum belt deflection is the perpendicular distance from the top of the centre roller to the lowest point of the belt before the next centre roller and the idler pitch is the distance between successive idler groups (see Figure 2.5).

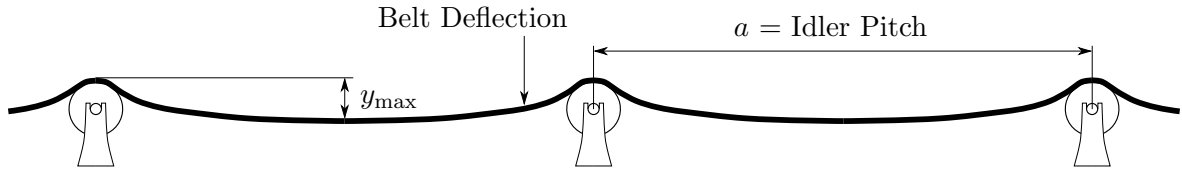


Figure 2.5: Belt deflection and idler pitch

In order to limit belt sag, industry design standards recommend specific idler pitches according to the aggregate density of the load and the width of the belt [16]. According to such standards, carrying idler pitches vary from 1.7 m for load densities between 480 and 800 kg/m³ with belt widths under 600 mm, to 0.6 m for densities above 2400 kg/m³ and belt widths above 1600 mm [8]. These standard spacings are recommended according to belt width and material density because belt sag is closely related to the weight of the belt and its

load; the wider the belt is, the greater the carrying capacity (see Equation 2.1) and therefore the greater the weight of the loaded belt [17].

Other factors that affect the belt sag include idler pitch and belt tension. Together, all these factors affect belt sag according to:

$$K_S = \frac{y_{\max}}{a} = \frac{W a}{8T} \quad (2.2)$$

where W is the combined weight of the belt and conveyed material per length of belting [N/m] and T is the tension of the belt [N]. This equation shows that for the same belt tension, long idler spacings and heavy average loaded belt weights (affected by belt width) result in high belt sag.

2.2.4 Belt Type

Conveyor belts are designed to transmit the necessary tension to move a load, absorb impact energy from falling material, provide a friction surface for driving and tracking, and allow for abrasion and wear [18]. Belts essentially consist of two parts; the structural carcass and the protective covers.

Belt carcasses provide the necessary strength and stability needed to withstand the various loading conditions experienced by the belt as a result of belt troughing, belt tension, cargo weight and material impact [19]. Most belt carcasses consist of one or more plies of woven fabric (such as Nylon, polyester or Aramid) with lengthwise warp fibres (providing tensile strength) bound to transverse weft fibres (providing impact resistance, load support and general stability) [18].

Steel cord carcasses, consisting of a single layer of parallel steel cables embedded in rubber, are often used in high tension applications or when minimal belt stretch is desired [20]. Carcasses may also be solid woven; a construction that became prominent in response to the desire for fire-resistant belting [19]. This type of carcass is made up of a single solid multiple woven fabric with at least two warp fibre systems and is impregnated with fire retardant Polyvinyl Chloride (PVC).

To protect the carcass from load abrasion, this section of the belt is sandwiched between two elastomer covers. These covers provide very little, if any, structural strength and are generally made from natural/synthetic rubber or PVC. The top cover provides impact and wear protection while the bottom cover provides a friction surface for driving and tracking the belt [18]. To account for abrasion, impact damage and elastomer wear, the top cover is generally thicker than the bottom cover. Various belt constructions are depicted in Figure 2.6.

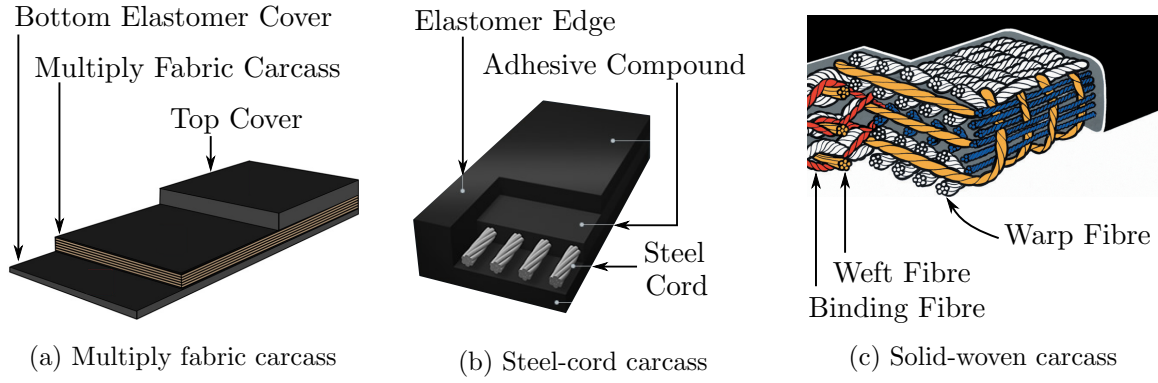


Figure 2.6: Annotated diagrams of various conveyor belting constructions [21, 22, 23]

2.2.5 Roller Type

Conveyor rollers are essentially made up of an outer cylindrical shell that rotates on a shaft via bearings which are housed in the end plates and protected from contamination by seals [16]. Figure 2.7 illustrates this general construction.

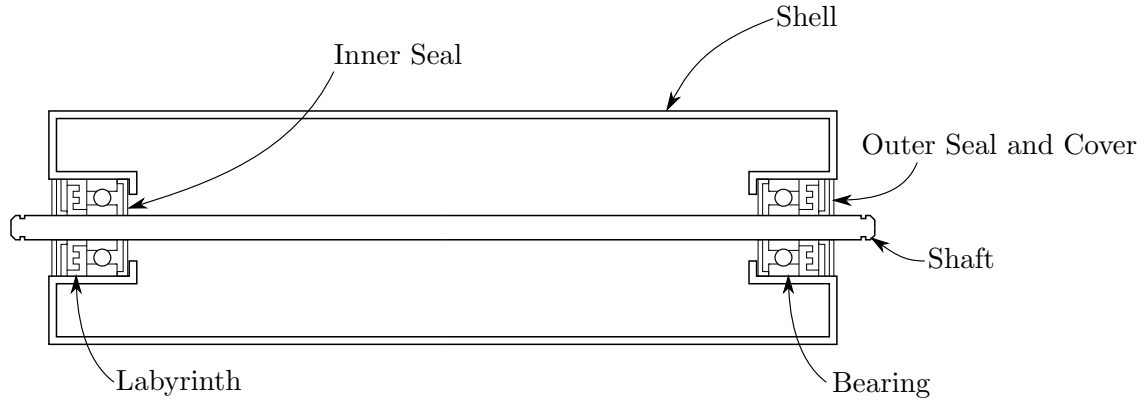


Figure 2.7: Longitudinal cross-section through a conveyor roller

In the past, the typical material choice for roller shells was steel, due to its availability and durability. These days, however, due to corrosion, noise and weight considerations, polymer shells are sometimes preferred. The most frequently chosen polymer shells are made from Nylon and High Density Polyethylene (HDPE). Nylon shells are stronger than similarly priced HDPE rollers but are sensitive to moisture absorption and do not have as good wear and impact resistant properties as HDPE [24]. Various advantages and disadvantages of steel and polymer shells are presented in Table 2.1.

Table 2.1: Advantages and disadvantages of steel compared to polymer idlers [16, 24, 25]

	Advantages	Disadvantages
Steel	• Relatively cheap • Dimensionally stable • Withstand higher loads • Non-flammable	• Noisy • Sensitive to corrosion • Higher drive power required
Polymer	• Quiet • Lightweight • Corrosion resistant	• Expensive • Less rigid • Sensitive to high temperatures

Knowing that conveyor belting and rollers may be constructed from a variety of different materials, unidirectional idler braking capability implications may now be considered. Since unidirectional devices rely on the friction generated between the conveyor belt and stationary idlers to provide the required braking force, it is intuitive that the belt and roller material combination will directly affect the performance of this safety precaution. This would mean that surface contact materials with low coefficients of static friction (such as a PVC conveyor belt in contact with HDPE idler rollers) would have inferior braking capabilities as compared with high friction pairs (such as a rubber belt on steel rollers) and therefore more unidirectional idlers would need to be installed into the system.

2.2.6 Roller Diameter

Belt conveying systems are designed with rollers of various standard outer diameters, d , typically ranging from 102 to 178 mm (although larger rollers are possible and are occasionally used)[26]. Roller diameter affects the resistance to motion of the conveyor belt due to the development of indentation rolling resistance and rotating resistance, which in turn influences belt tension and drive pulley power requirements [27]. Indentation rolling resistance develops in conveyor systems as a result of the asymmetric compression of the belt bottom cover as it traverses over successive idlers. This arises due to the viscoelastic characteristic of the elastomer bottom cover which cannot recover at the same rate as it is compressed [28]. The asymmetric pressure distribution that results, creates an offset force and resistance torque that opposes belt movement over the idler. Figure 2.8 illustrates this belt resistance.

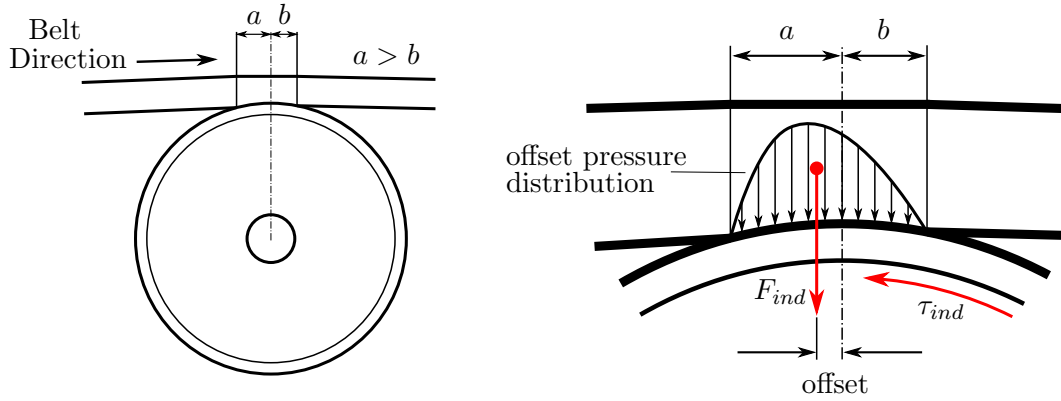


Figure 2.8: Indentation rolling resistance; adapted from [28]

Idler rotating resistance is related to the power requirements of idlers and develops as a result of the torsional resistance to rotation experienced in the roller bearing sub-assembly. This torsional resistance is the combined result of friction due to the contact between moving parts of the bearing itself, shear of the lubricant and the seal drag [29].

Both of these resistances may be reduced by changing the diameter of rollers. In the case of indentation rolling resistance, increasing the roller diameter will reduce the contact pressure and stresses at the belt-to-roller interface and subsequently reduce the opposing offset force and resistance torque [30]. Rotational resistance may also be reduced by increasing roller diameter. This is mainly because a larger distance between the roller shell and axis results in a larger moment arm (i.e radius) and subsequent torque to overcome bearing resistances. Increasing the diameter will also decrease the angular velocity of the roller at a given belt speed which in turn lowers the viscous resistance in bearing lubricants [30]. Increasing roller diameter is also beneficial in its reduction of vibration amplitudes that result from a lower angular velocity. The disadvantages of increasing the roller diameter include the increase in roller mass (making them more difficult to handle) and the higher material cost. This added mass is also detrimental during starting and stopping procedures which require more time to be executed due to the increased inertial mass.

2.2.7 Belt Tension

The amount of tension needed in a conveyor system depends on a number of motion resisting factors that must be overcome. These factors primarily include the belt length, quantity of idlers, inclination angle, belt width and load weight. Most of the tension is generated at the drive pulley and the take-up mechanism [1]. Belt tension is not constant throughout the conveyor system but instead increases as the belt moves closer to the drive pulley and then reduces drastically immediately afterwards. This tension gradient is what pulls the loaded

belt along the idlers. Minimum belt tensions are normally prescribed according to pulley drive and belt sag requirements [8].

With regard to the effects of belt tension on unidirectional idler performance, there are two failure scenarios worth considering. Firstly, the event in which the drive motor trips and there is no backstop mechanism in place. In such an occurrence, the high tension area around the drive pulley will relax and the belt tension, now determined only by the take-up mass, will become uniform throughout the system. Without a tension gradient present, there will be no direct tension induced force to add to the downward pull on the loaded conveyor belt. However, since belt tension affects the sag ratio (see Equation 2.2), this parameter is still likely to have an indirect effect on the functionality of unidirectional devices.

The second scenario worth considering is that of a belt snap which completely releases the belt tension from the conveyor system. In this case, the low tension state of the snapped belt is likely to induce considerable belt sagging between idlers which increases the force required to cause belt slip; thus helping to prevent a runaway. While the potential "surfing" of the belt over some idlers may occur as the tension is relaxed, this is not expected to significantly reduce the effectiveness of the unidirectional idlers due to the increased belt sag.

2.2.8 Load

The final parameter of concern is the material load on the conveyor. Many of the effects of this factor have already been discussed to some extent in the previously presented belt conveying parameters. All of these parameters depend on the material load in some way. The aggregate material density of the load affects the following factors ([5],[8],[13] and [17]):

- drive power requirements,
- induced belt tension,
- required belt type,
- allowable angle of inclination,
- idler loading condition,
- required idler roller dimensions and type,
- troughing angle and belt width needed to achieve the desired carrying capacity,
- idler pitch required to minimise belt sag.

Belt load also plays a major role in unidirectional idler functionality. A larger load will increase the normal forces acting at the idlers which in turn will increase the resistive friction forces between the conveyor belt and the unidirectional idlers. However, a larger load also means a larger downhill belt pulling force acting parallel to the inclined system (see Figure 2.2b). Furthermore, in the event of a belt snap, the combined effect of the loss of tension in the belt and the downward pull of the material weight will increase the belt sag between idlers (see Equation 2.2), thus assisting the unidirectional mechanism.

2.3 Design of Experiment Techniques

When undertaking experimental research in which many parameters are varied, it is useful to employ statistical Design of Experiment (DOE) techniques in order to identify the variables that have significant influence on a measured output, and to make better informed conclusions about the relationships between parameters and the measured outcome. Of particular interest to this research problem are the DOE techniques associated with factorial designs – studies in which each complete trial or replication of all the possible level combinations of multiple factors of an experiment are investigated [31].

From the data of a full factorial design, statistical methods may be used to determine the main effects of individual factors as well as the interactions between different combinations of the factors. The main effect of a factor is determined by the change in response experienced when there is a change in the level of the factor. Interactions between factors occur when the difference in response of a factor is not the same at all the levels of other factors. In other words, there is said to be an interaction between factors A and B if the effect of A depends on the particular level of factor B .

Factorial designs help to avoid misleading conclusions when interactions may be present. The downside, however, is the large number of tests required to vary many factors at numerous levels. In the general case of a full factorial design where there are a levels of factor A , b levels of factor B , c levels of factor C and so on, and if there are n replicates of each test, then there will need to be $a \times b \times c \times \dots \times n$ total observations [31]. Considering this particular research venture and assuming that the ten aforementioned belt conveying parameters are varied at three levels each and three replicates are employed, then the required number of tests would be equal to $3^{10} \times 3 = 177\,147$.

An experimental procedure of this size is obviously over-scoped. One of the methods that may be used to downsize the proposed testing procedure is to remove the multiple levels of the factors and instead focus on the high (+) and low (−) levels of each factor. This is known

as the 2^k factorial design and provides the fewest experimental runs that may be studied in a complete factorial design with k factors. This DOE is particularly useful during early experimental work in which there are a large number of factors of interest, and is therefore widely used in factor screening experiments [32].

In order to explain the statistics behind this type of experimental design, which is usually done with a computer, consider an arbitrary 2^3 factorial design. Such a design would consist of $\binom{k}{1} = k = 3$ main effects, $\binom{k}{2} = \binom{3}{2} = 3$ two-factor interactions and $\binom{k}{3} = \binom{3}{3} = 1$ three-factor interaction. The full factorial experiment in this case would consist of eight tests as shown in Table 2.2. With only eight different variable configurations (plus replicates) it is possible to determine which of the parameters have the most influence on the response, and whether any factor interactions are relevant. This is achieved by varying the three factors of interest between two levels each – a high (+) and low (−) value. The response variable y is a measured output of a particular experiment.

Table 2.2: Hypothetical 2^3 full factorial design of experiment

Experiment	Factor			Response variable	Interactions			
	A	B	C		$A \cdot B$	$A \cdot C$	$B \cdot C$	$A \cdot B \cdot C$
1	−	−	−	$y_{l,l,l}$	+	+	+	−
2	−	−	+	$y_{l,l,h}$	+	−	−	+
3	−	+	−	$y_{l,h,l}$	−	+	−	+
4	−	+	+	$y_{l,h,h}$	−	−	+	−
5	+	−	−	$y_{h,l,l}$	−	−	+	+
6	+	−	+	$y_{h,l,h}$	−	+	−	−
7	+	+	−	$y_{h,h,l}$	+	−	−	−
8	+	+	+	$y_{h,h,h}$	+	+	+	+

With this experiment in mind the main effect, M , of factor A may be defined as the difference between the average response variable at high levels of factor A (experiments 5 to 8) and that at low levels (experiments 1 to 4):

$$M_A = \frac{y_{h,l,l} + y_{h,l,h} + y_{h,h,l} + y_{h,h,h}}{4} - \frac{y_{l,l,l} + y_{l,l,h} + y_{l,h,l} + y_{l,h,h}}{4} \quad (2.3)$$

This equation may easily be expanded to determine the main effects of B and C . Furthermore, it may be extended to the general case of the main effect of a factor, say x , in a 2^k design:

$$M_x = \frac{\sum y_{x=h}}{2^{k-1}} - \frac{\sum y_{x=l}}{2^{k-1}} = 2^{1-k} \left(\sum y_{x=h} - \sum y_{x=l} \right) \quad (2.4)$$

where $y_{x=h}$ is any response variable corresponding to an experiment in which factor x is set at high level and $y_{x=l}$ is the opposite.

Referring back to the example laid out in Table 2.2, the interactions of various factors may be determined in a similar manner. Going straight to the general case, any effect of an n -factor interaction may be determined according to:

$$M_{x_1 \cdot x_2 \cdots x_n} = 2^{1-k} \left(\sum y_{x_1 \cdot x_2 \cdots x_n=h} - \sum y_{x_1 \cdot x_2 \cdots x_n=l} \right) \quad (2.5)$$

The high (+) and low (−) levels of the n -factor interactions are determined by the product of the signs corresponding to the main effects. For example, the sign of interaction $A \cdot B \cdot C$ in experiment 1 is equal to the product of the signs of factors A , B and C during the same run (i.e. $(-) \times (-) \times (-)$) which corresponds to a low (−) level.

Once the main and interaction effects have been determined, an analysis of variance (ANOVA) may be performed to test for the significance of these effects. This form of statistical analysis compares the amount of variation between different experimental factors and multi-factor interactions with the variation within those factors and interactions themselves. In the context of this research application, the most important metric derived from an analysis of variance is the P-value.

In statistical design of experiments, the P-value is the probability that the experimentally observed result could occur when a factor, or interaction, has no effect on the measured outcome. If the P-value is very low (< 0.05), it means that the observed result is very unlikely to occur when the factor/interaction is insignificant – therefore the factor/interaction must be significant [33]. To put this simply, if the P-value of a specific factor/interaction is lower than 0.05 then the effect of that factor/interaction may be considered significant.

Although a list of P-values summarises the factor/interaction significance, the results of an analysis of variance are more appreciably summarised in the form of a half-normal plot (such as that presented in Figure 2.9). A half-normal plot presents the absolute values of the different main and interaction effects (see Equations 2.4 and 2.5) plotted against the their standardised half-normal scores as points on the graph. Half normal scores relate to the theoretical order of statistical medians that would result from randomly drawing samples from a half normal distribution (i.e. the absolute values of a normal distribution) centred about a zero mean. In addition to these points, a half-normal plot also contains a distribution fit

line for the case when factors/interactions have absolutely no effect on the measured outcome (i.e. when P-value = 0) [32].

One of the most advantageous benefits of factorial DOEs is the identification of interactions. In complex study areas, such as this, the independent factors being varied may interact with each other, and therefore simply assessing the main effects can be misleading. This is found to be the case in the hypothetical 2^3 factorial experiment which shows the interaction between factors B and C to be significant (see Figure 2.9).

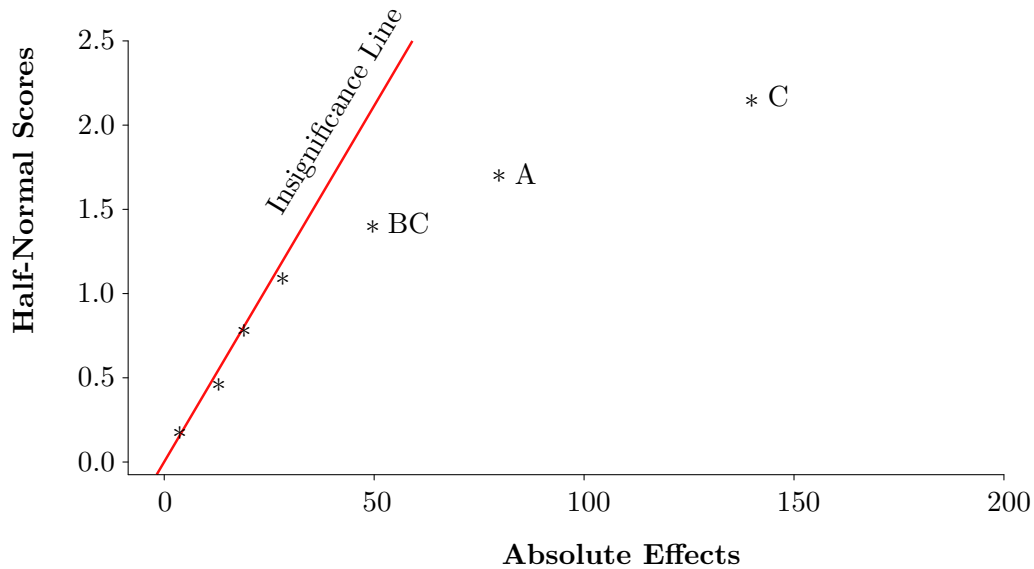


Figure 2.9: Half-normal plot for hypothetical 2^3 factorial DOE

To better understand how interactions affect the measured outcome, interaction effect plots may be generated following an analysis of variance. Continuing with the hypothetical experiment, Figure 2.10 presents the interaction effect plot for the significant interaction between factors B and C.

An interaction effect plot presents the average measured outcome on the y-axis, the varied levels of one of the factors making up the interaction on the x-axis, and individual lines for each level of the other factor. When interpreting one of these plots, it is important to consider the difference in gradient of the interaction lines. Parallel lines indicate that there is no interaction between the factors being compared; while non-parallel lines (i.e. lines with differing gradient) indicate that there is an interaction.

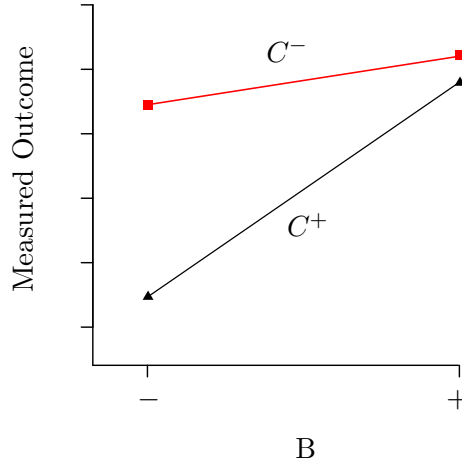


Figure 2.10: Interaction effect plot for hypothetical 2^3 factorial DOE

The greater the observed difference in line gradient, the greater the strength of the interaction. With this in mind, Figure 2.10 reveals a considerable interaction to exist between factors B and C. In this case, it may be seen that the effects of factor B are more significant at higher levels of factor C (indicated by the steeper gradient of the C^+ line).

Relating the 2^k factorial design back to this particular investigation in the field of belt conveying, it is observed that although this kind of design reduces the number of experimental tests, varying ten parameters at two levels and three replicates would result in $2^{10} \times 3 = 3072$ tests needing to be run. One of the ways that this number may be further reduced is by transforming the 2^k factorial design into a 2^{k-p} fractional design.

Fractional factorial designs allow researchers to run a subset of a full factorial design and still provide sufficient information about the main effects and some of the interaction effects [34]. If it may be assumed that certain high-order interactions are negligible, then fractional factorial designs may be used to investigate the main effects and low order interactions by running only a fraction of the full factorial design. The downside of running smaller and smaller fractions of a full factorial design is the confounding of higher order interactions that results, making it impossible to test the sensitivity of these interactions.

Two or more effects are said to be confounded (or aliased) if the linear combinations of observations used to estimate the effects are the same [35]. For example, interactions XY and XZ are confounded if their levels (+ or -) match at every test. It is impossible to estimate the significance of confounded factors. However, if the research is focused on the low order interactions and main effects, then confounding is of no real consequence.

Fractional factorial designs are often categorized and grouped according to the degree of confounding that they exhibit. These categories are called design resolutions and are often indicated (using Roman numerals) in the subscripts of 2^{k-p} design names. The higher the design resolution, the less confounding and the better the results are expected to be [35]. Definitions of some of the most common design resolutions may be summarised as:

- *Resolution III designs* – main effects are not confounded with any other main effects but are confounded with two-factor interactions and two-factor interactions may be confounded with other two-factor interactions [35].
- *Resolution IV designs* – main effects are not confounded with any other main effects or with any two-factor interactions, but two-factor interactions are confounded with other two-factor interactions [35].
- *Resolution V designs* – main effects and two-factor interactions are not confounded with any other main effect or two-factor interaction, but two-factor interactions are confounded with three-factor interactions [35].

These points reveal that in order to accurately determine main effects and two-factor interaction pairs, fractional designs need to have a resolution of at least V (i.e. 2_V^{k-p}). Table 2.3 presents the resolutions and number of experimental runs (without replicates) corresponding to common fractional factorial designs. From this table the best suited DOE for the ten factor belt resistance investigation is the 2_V^{10-3} fractional factorial design. This design has the required resolution and if replicated 3 times will require $128 \times 3 = 384$ experimental runs.

Table 2.3: Common 2^{k-p} fractional factorial designs, adapted from [35]

Number of Factors, k	Design	Resolution	Number of Runs
6	2^{6-1}	VI	32
	2^{6-2}	IV	16
	2^{6-3}	III	8
7	2^{7-1}	VII	64
	2^{7-2}	IV	32
	2^{7-3}	IV	16
	2^{7-4}	III	8
8	2^{8-2}	V	64
	2^{8-3}	IV	32
	2^{8-4}	IV	16
9	2^{9-2}	VI	128
	2^{9-3}	IV	64
	2^{9-4}	IV	32
	2^{9-5}	III	16
10	2^{10-3}	V	128
	2^{10-4}	IV	64
	2^{10-5}	IV	32
	2^{10-6}	III	16

Although 384 experimental runs is a lot lower than the initially estimated 177 147 runs, this is still a large testing procedure and there is room for improvement. One way that the number of tests could potentially be reduced further is by combining certain parameters into single two-level factors.

By combining the parameters of belt type and roller type into a coefficient of friction factor, and combining idler pitch, belt sag and belt tension into a sag factor, the total number of factors reduces to 7. Furthermore, if it is assumed that adding load to the system improves the friction braking of unidirectional idlers due to increased belt sag and normal forces at the idlers, then the load factor may be disregarded for initial testing and instead, a handful of additional tests may be run with a loaded belt so as to confirm this assumption. This would then reduce the total number of factors to 6. With this many factors, Table 2.3 suggests a suitable DOE to be the 2_{VI}^{6-1} fractional factorial design. This design has a higher resolution than the minimum requirement and if replicated 3 times will require only $32 \times 3 = 96$ experimental runs plus a few extra loaded belt runs.

Chapter 3 Experimentation

3.1 Introduction

In order to realise the objectives presented in Section 1.4.2, a range of experiments were planned using DOE techniques. The core experimental work was conducted using a unidirectional roller (UDR) Test Rig, which made use of cantilevered load cells to measure the resistive force at belt slip (see Section 3.3). In addition to this, supplementary experiments were conducted with a Jenike Direct Shear Tester (JDST) to determine the friction coefficients of various conveyor belt and roller material pairs (see Section 3.4). For a complete list of the experiments conducted, refer to Appendix A.

Before these experimental programmes are introduced, however, it is worth developing an understanding of the mechanics of unidirectional idlers. This is especially important when specifying the required quantities of these devices. Without knowing the resistive forces at play, the specified quantities of unidirectional rollers could be excessive or insufficient. The consequences of over/under-specifying these products could be quite serious, especially when considering the financial and safety losses. In order to develop a better understanding of the mechanics of the system, a basic force analysis was conducted and a simplified unidirectional idler quantity formula derived. The contents of the following section explain the derivation.

3.2 Preliminary Analysis of the Mechanics of Unidirectional Idlers

A free body diagram of a single unidirectional idler preventing a loaded section of belt from accelerating down an incline is presented in Figure 3.1. In this diagram, the conveyor belt is assumed to be a rigid and straight body, the material is assumed to not slip on the conveyor belt, and the resistive force is assumed to be as a result of belt-idler contact friction alone (see footnote on page 8).

Figure 3.1 shows a section of the belt and load with a weight (W) supported on a unidirectional idler where a normal reaction force (N) and friction force (F_f) act at the contact point of the belt and idler.

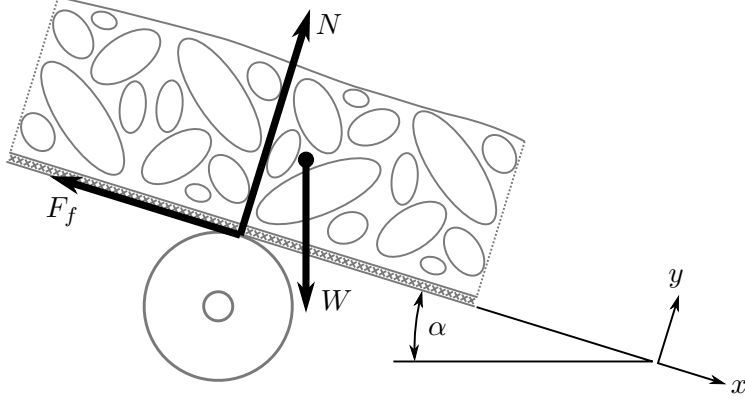


Figure 3.1: Free body diagram of a unidirectional idler

In addition to the assumptions detailed above, it is also assumed that the static friction limit equals that of the kinetic limit^a (i.e. $\mu_s = \mu_k = \mu$) and that belt slip occurs at a constant velocity. With this in mind, consider the limiting case where the belt is about to slip over the locked roller. Knowing that the system is in equilibrium, the following force balances may be stated:

$$\sum F_x = 0 \quad \implies \quad F_f = \mu N = W \sin \alpha \quad (3.1)$$

$$\sum F_y = 0 \quad \implies \quad N = W \cos \alpha \quad (3.2)$$

From Equation 3.1 and 3.2, it may be seen that the conveyor belt will slip when:

$$\tan \alpha = \mu \quad (3.3)$$

To determine the quantity of idlers required to be locked, the carrying side of the belt must be considered in its entirety. Figure 3.2 illustrates the simple mechanics of a length of belt (L), at a specific angle of inclination (α), supported by idlers separated from each other by a constant pitch (a).

^aThis conservative assumption adds an implicit safety factor to unidirectional idler quantity specifying calculations. Specifying these quantities based on kinetic friction (the friction type measured in this research) will be more than sufficient to prevent the loaded belt from overcoming the characteristically higher static friction.

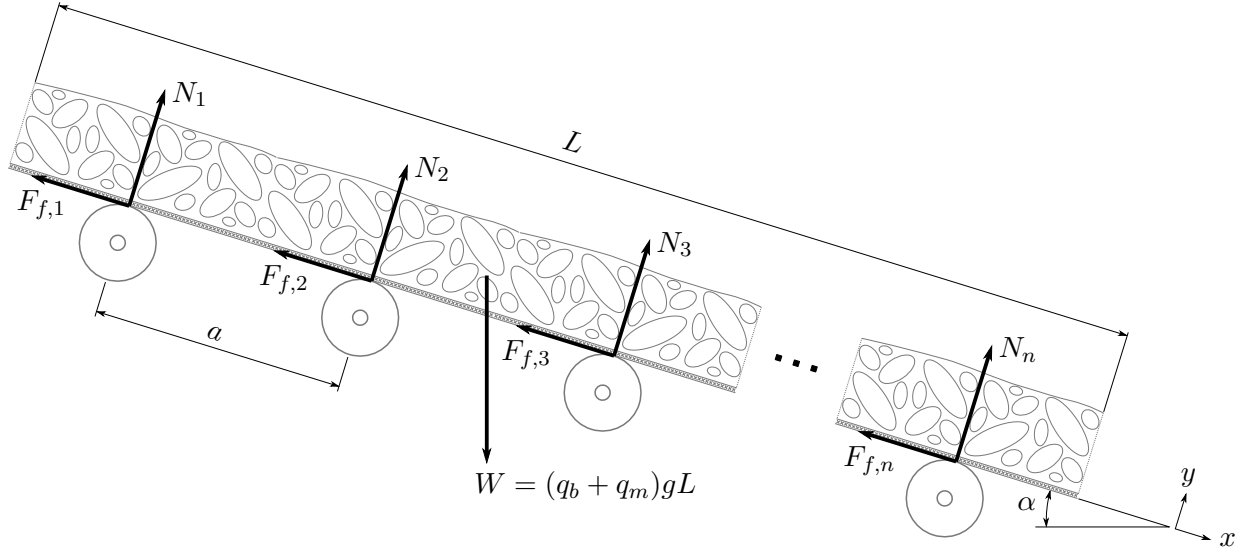


Figure 3.2: Free body diagram of an inclined conveyor belt

Since every inclined belt installation contains a unique quantity of rollers, it is more meaningful to calculate the braking needs of a system according to the proportion of all the rollers that are required to be unidirectional (defined in this research as η) rather than the explicit number of these rollers. Taking this into account, and with reference to Figure 3.2, the force balances may be stated as:

$$\sum F_x = 0 \quad \Rightarrow \quad \eta\mu \sum_{i=1}^n N_i = (q_b + q_m)gL \sin \alpha \quad (3.4)$$

$$\sum F_y = 0 \quad \Rightarrow \quad \sum_{i=1}^n N_i = (q_b + q_m)gL \cos \alpha \quad (3.5)$$

where η is the required proportion of idlers that need to be unidirectional so as to prevent slip, q_b is the mass per metre of the conveyor belt and q_m is the mass per metre of the conveyed bulk material.

Assuming that the material load is distributed evenly along the entire length of the conveyor belt (i.e. $N_1 = N_2 = N_3 = \dots = N_n = N$), and where the number of idler sets is equal to L/a , Equations 3.4 and 3.5 may be rewritten as:

$$\sum F_x = 0 \quad \Rightarrow \quad \frac{\eta\mu LN}{a} = (q_b + q_m)gL \sin \alpha \quad (3.6)$$

$$\sum F_y = 0 \quad \Rightarrow \quad \frac{LN}{a} = (q_b + q_m)gL \cos \alpha \quad (3.7)$$

Substituting the LN/a term from Equation 3.7 into 3.6 gives:

$$\begin{aligned}\eta\mu(q_b + q_m)gL \cos \alpha &= (q_b + q_m)gL \sin \alpha \\ \therefore \eta &= \frac{(q_b + q_m)gL \sin \alpha}{\mu(q_b + q_m)gL \cos \alpha} \\ \therefore \eta &= \frac{1}{\mu} \tan \alpha\end{aligned}\tag{3.8}$$

Finally, by introducing a safety factor (n), the specified proportion of unidirectional idlers (η_s) may be calculated according to:

$$\eta_s = n\eta = \frac{n}{\mu} \tan \alpha\tag{3.9}$$

This represents a first approximation of the proportion of rollers that are required to resist belt slip. This calculation method depends on the frictional characteristics of the belt-to-roller interface and the angle of inclination.

3.3 Unidirectional Roller Experiments

3.3.1 Research Methods

The core set of experiments were executed using the UDR Test Rig (see Figures 3.4 and 3.5) that was designed and constructed by various member companies of the Conveyor Manufacturers Association of South Africa (CMA). A basic description of the functionality of this test rig is presented in Section 3.3.2.1. The experiments conducted with the UDR Test Rig were designed to determine the resistive forces that develop between fixed rollers and a conveyor belt being pulled over these fixed rollers, to represent a belt slipping down an incline in the event of a belt snap.

The method used to simulate a unidirectional roller and infer the resistive force between a single locked roller and the conveyor belt is illustrated in Figure 3.3. Due to the rigid connection between the roller and the fixed cantilever load cell (mounted to the frame of the UDR Test Rig) the roller is always locked in place. As the rig pulls the belt down the incline, the roller cannot rotate – thus imitating the functionality of a unidirectional idler during reverse motion of the conveyor belt.

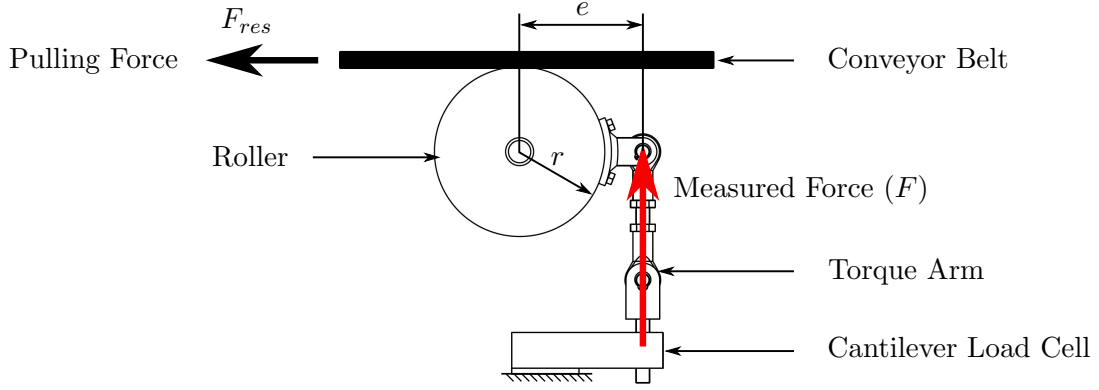


Figure 3.3: Diagram of the resistive force measurement technique

The force (F) measured by the load cell at an eccentric distance (e) from the roller axis is used to determine the rotation resisting torque (τ_{res}) according to:

$$\tau_{res} = Fe$$

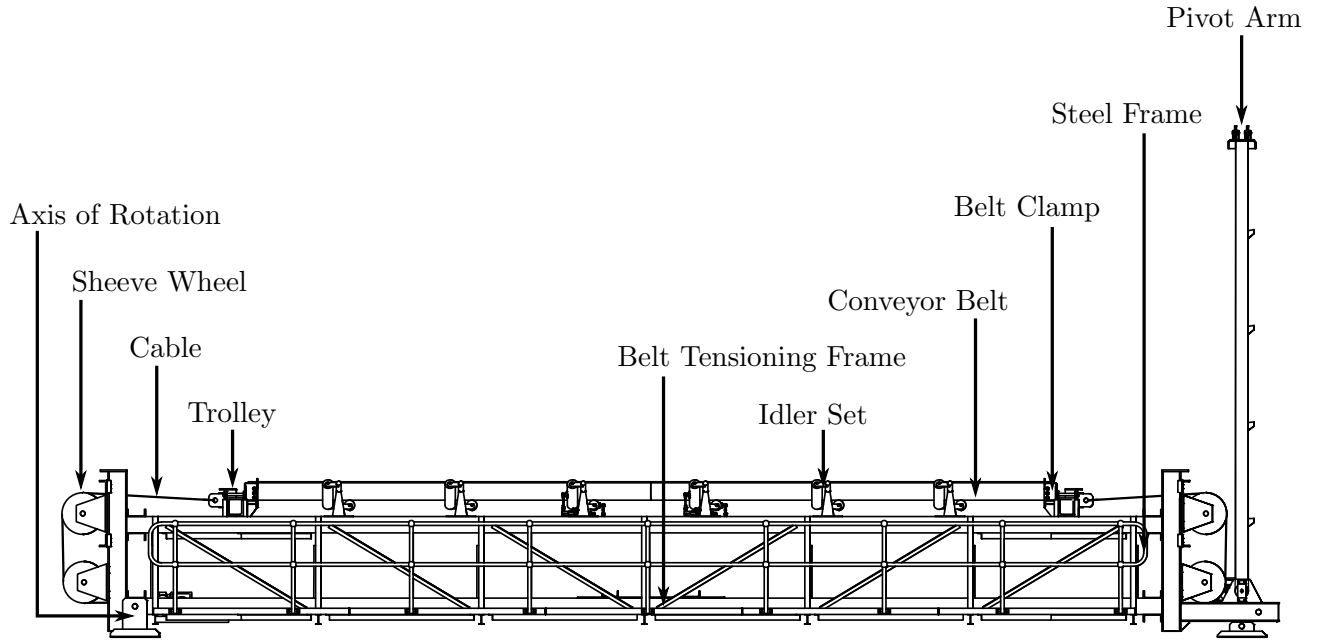
Since the forces are in equilibrium just before slippage, the opposite oriented resistive torque acting at the belt-roller interface ($\tau_{res} = F_{res}r$) has the same magnitude as the measured torque. Therefore the load cell measurements taken during experimentation may be used to calculate the resistive slip force at the belt-roller interface according to:

$$F_{res} = F\left(\frac{e}{r}\right) \quad (3.10)$$

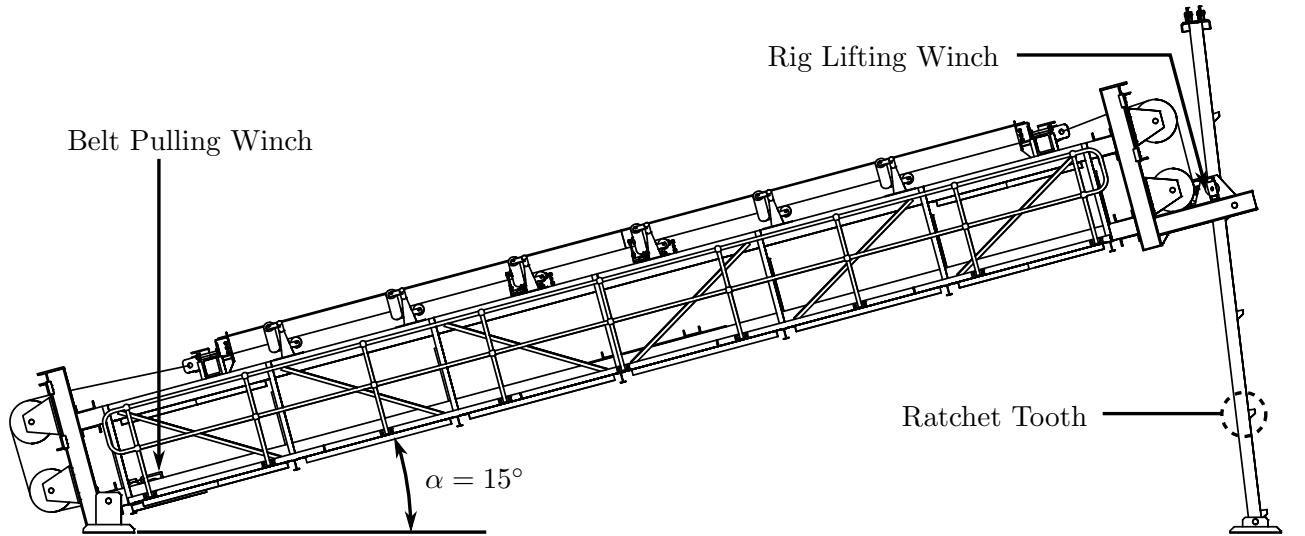
3.3.2 Apparatus

3.3.2.1 Unidirectional Roller Test Rig

The UDR Test Rig (shown in Figures 3.4 and 3.5) was installed in a section of donated factory space at Dunlop Belting Products in Benoni. Once the rig was commissioned, it was used to measure the belt slip forces corresponding to a wide variety of conveyor belt operating conditions. The test rig consisted of a steel frame onto which idler sets, sheave wheels, winches and a belt tensioning frame were fitted (see Figure 3.4a). A length of conveyor belting was tensioned over the idlers with cables that were connected to both ends of the belt via clamps attached to trolleys. These cables were guided by four sheave wheels to the bottom of the rig where they were connected to the tensioning frame. This frame made use of a six tonne lever hoist to provide the necessary belt tension, in the range of 500 to 2 500 kg force, which was measured with a load cell connected between one of the trolleys and cables (see Figure 3.5). The belt pulling force was provided by a winch fixed to the rig which pulled the tensioning frame along steel rails.



(a) Test rig in the horizontal position



(b) Test rig inclined at 15°

Figure 3.4: Annotated diagrams of the unidirectional roller test rig

The test rig could also be pivoted vertically at one of its ends, thus allowing tests to be run at different inclinations. This was achieved with a winch, mounted to the end of the frame, whose cable was guided by small sheave wheels on the top of the pivot arm and fixed to the frame via an eye-bolt. When the winch reeled in the cable, the frame was pulled up along the pivot arm to the desired angle (Figure 3.4b). During testing the frame was held in place by the pivot arm ratchet teeth. The rig could be rotated to angles of 5°, 10° and 15°.

Besides varying the inclination angle, other important belt conveying parameter levels could also be changed with this rig. Troughing angle, roller type, roller diameter and idler pitch were varied by attaching idlers to the steel frame at different spacings, with different troughing angled frames, installed with rollers of varying size and type. The possible idler pitches that could be preset were 1200, 1500 and 1800 mm. Different levels of belt sag could also be achieved by varying idler pitch, installing belts with different widths, varying the material load on the belt and tensioning the belt at different levels. The rig was designed to allow for belt widths of 900, 1200 and 1500 mm, and belt load was simulated through the use of sand bags (as shown in Figure 3.5). Finally, conveyor belt type could also be varied on this rig.^b

3.3.2.2 Miscellaneous Test Rig Apparatus

The design and construction of the UDR Test Rig was commissioned and completed by the CMA before the commencement of this research venture. This includes the design and manufacture of the steel frame, pivot arm, tensioning frame, trolleys, sheeve wheels, rotational supports, winches as shown in Figure 3.4. Although the UDR Test Rig had been almost entirely constructed when starting with this research, there were still two key components that needed to be designed and manufactured. These outstanding components were the belt clamps used to fix both ends of the conveyor belt to the trolleys and the mounting brackets for the cantilever load cells. Images of these components are shown in Figure 3.5 and the corresponding manufacturing drawings are presented in Appendix C.

The belt clamps were designed to be incorporated into the existing test rig structure. Different belt clamps were designed to accommodate the three belt widths (900, 1200 and 1500 mm) and the three different troughing angles of interest (0° , 35° and 45°). All the belt clamps were designed with the same mounting base plate which was fixed to the belt trolleys via four M30 threaded rods (see Figure 3.5). Onto these mounting base plates were welded a number of different steel plate extensions, belt supporting angle irons and stiffener plates of various constructions according to the belt width and troughing angle desired. The conveyor belt was prepared for clamping by punching 22 mm diameter holes corresponding to those drilled into the support angle iron of the specific belt clamp assembly. Finally, the belt could be clamped to the trolleys by lining up the punched holes in the belt with those of the support angle iron and the clamping angle iron, and bolting them all together with M20 bolts. The clamps were designed to withstand belt tensions of up to six tonnes.

^bAlthough the UDR Test Rig allowed for the variation of belt type and width, all experiments conducted in this research venture were done so with a single belt sample – a 900 mm wide Nitrile rubber covered three-ply fabric belt. This was mainly due to the impracticality of varying belt type and width; both in terms of the increase in the required number of experimental iterations that would result and also due to the difficulty of changing belts and dealing with wider belts without the necessary equipment and manpower.

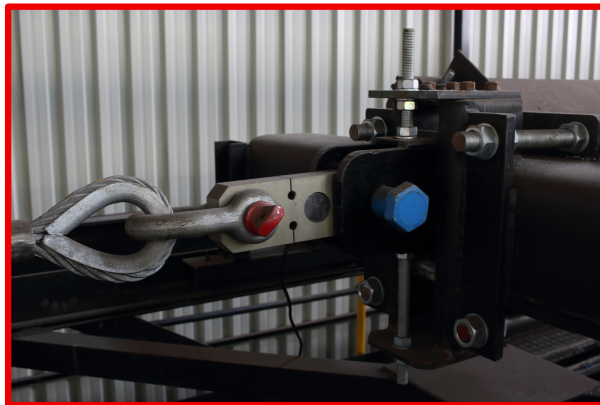
35° Trough Belt Clamp Assembly



135 kg/m of Material Load in Sand Bags



20t Tensioning Load Cell



2t Cantilever Load Cells and Brackets



Control Panel



6t Lever Hoist

Figure 3.5: Images of the UDR test rig and its key components

Load cell mounting brackets were also required to be designed and manufactured for the UDR Test Rig. The purpose of these brackets was to fix one end of a cantilever load cell to the test rig and the other to the roller that was desired to be locked in place. This was done to measure the resistive force at the belt-roller interface and to simulate a unidirectional idler locked in place during reverse motion of the belt. An example of one of the load cell bracket and torque arm designs used to achieve these objectives is shown in Figure 3.6.

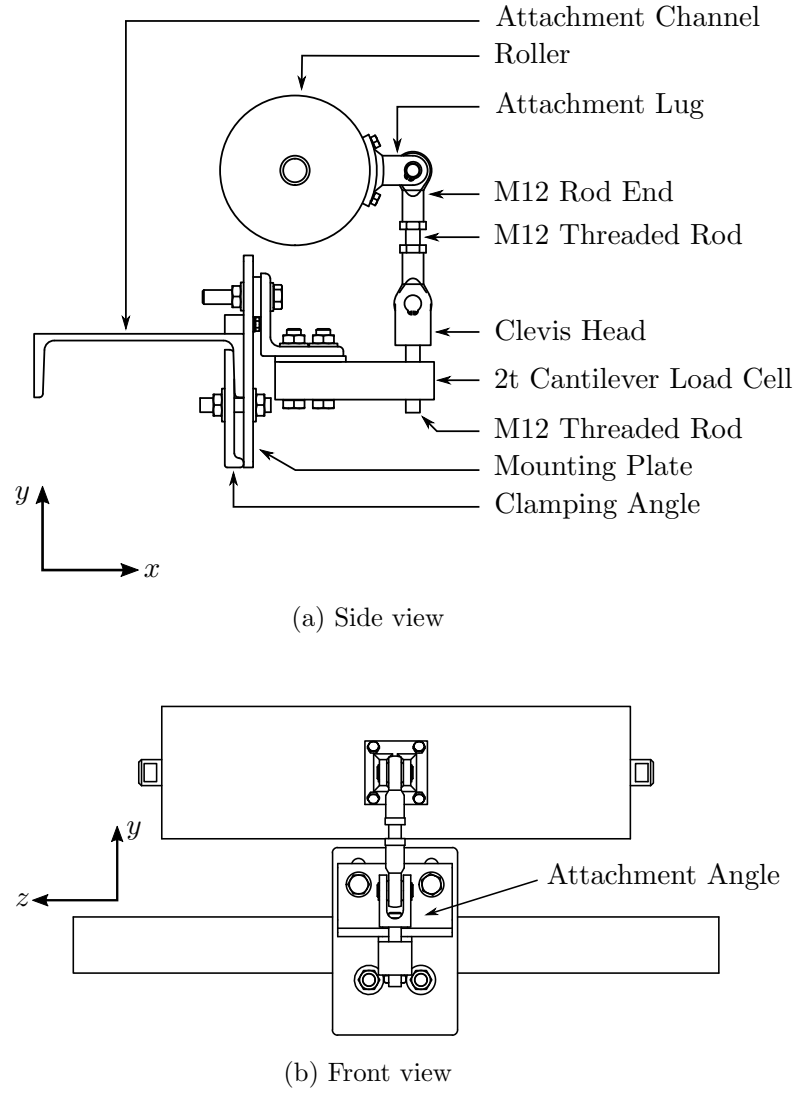


Figure 3.6: Load cell mounting bracket and torque arm design

The mounting bracket assemblies consist of a mounting plate that is clamped to the test rig structure via two bolts and a clamping angle. An attachment angle iron is bolted to the mounting plate at a distance corresponding to the number of 5 mm spacers fitted between the two. The load cell is bolted to the attachment angle at one end and connected to a clevis head at the other end. The rigid connection to the roller is completed via a series of clevis

pins, rod ends and a threaded rod that fix the load cell to attachment lugs that are mounted to the roller shell. In the case of steel rollers, these lugs were welded directly to the outer shell. In the case of a polymer rollers, these lugs were bolted in place.

Figure 3.6 illustrates how the position of the load cells could be fine-tuned by moving the mounting plate along the attachment channel (adjustment in the z -direction), by increasing or decreasing the length of threaded rod used for the torque arm (adjustment in the y -direction) and by adding or removing spacer plates between the mounting plate and the attachment angle iron (adjustment in the x -direction when testing rollers of different diameters). In the case of an inclined wing roller, the mounting bracket assembly used to correctly position the load cell differed slightly from the design shown in Figure 3.6. In that case, a mounting plate with attachment holes drilled at the specific wing roller inclination angle was used. An example of this case is shown by the load cell mounting photo in Figure 3.5.

It should be noted that although the test rig was fitted with six three-roll idler frames, only the rollers of the two centre most idler sets (i.e. six rollers) were connected to load cells. The remaining frames were installed with conventional rollers. The data collected from the two measured idler frames was used to compute the average slip forces per centre roller, wing roller and idler set.

3.3.2.3 Data Collection Apparatus

A variety of National Instruments' hardware and software was used to collect and pre-process the experimental data. The seven load cells used on the rig (six cantilevered load cells and a tension link load cell) were connected to NI 9237 Analogue Input Modules via ethernet cables. These modules were plugged into a cDAQ-9178 CompactDAQ Chassis which sent the measurement signals to a laptop via a Hi-Speed USB 2.0 interface. NI LabVIEW 2014 software was used to process and save this data. Using this software, an interactive data capturing panel was coded to allow a user to specify the test rig set-up before commencing with an experiment and then start the data capturing function when ready. Once data capturing had commenced, the panel would plot live resistive force data (pre-processed from the actual load cell measurements using Equation 3.10) and then write all the data to a file location corresponding to the experimental test and iteration numbers. LabVIEW was also used to correctly scale the measured force data according to the details contained within the load cell calibration certificates (see Appendix D).

An example of the user specified inputs and live force plots of the data acquisition panel created with LabVIEW is shown in Figure 3.7. This figure also shows the graphical code

that was constructed for the panel and which runs in the background during experimental data collection.

Block Diagram Code

Data Acquisition Panel

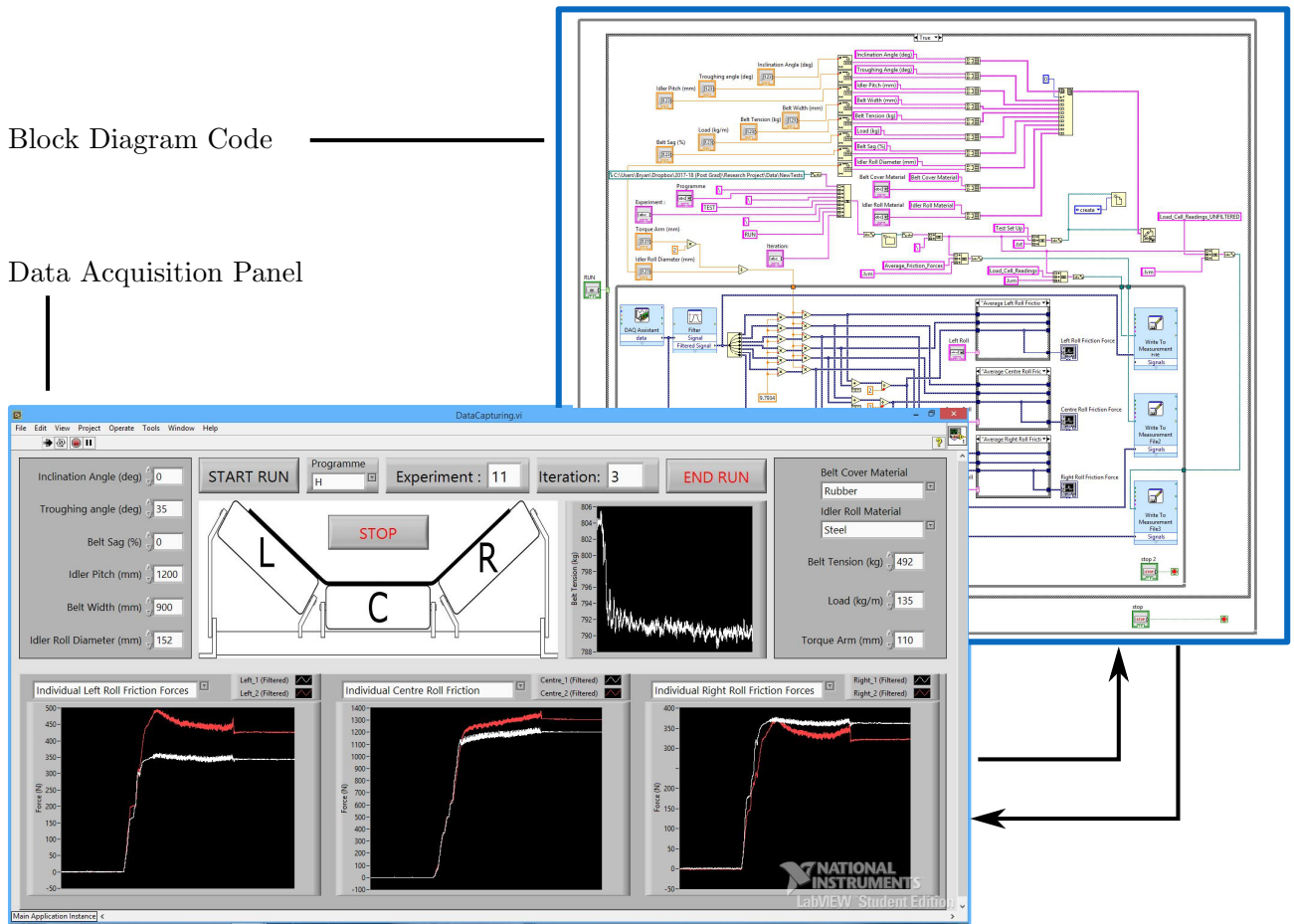


Figure 3.7: Data acquisition panel and background code

3.3.3 Experimental Procedures and Precautions

Before unidirectional roller experiments can be conducted, the test rig needs to be set up according to the operating conditions desired to be investigated. These include the troughing angle, idler pitch, roller diameter, roller type, belt load, belt tension and inclination angle factors. The steps followed to successfully set up the rig for testing may be broken up into three procedures in the following order:

1. General set-up procedure
2. Belt tensioning procedure
3. Test rig lifting procedure

The following content will expand on these set-up procedures, explain the unidirectional roller experiment methodology, describe the various shut down processes and list the precautions followed when working on the UDR Test Rig.

3.3.3.1 General Set-up Procedure

The general set-up of the UDR Test Rig differs with every experiment. This is because of the large number of factors that may be varied on the rig. The general set-up includes all activities required to install the desired idler frames and rollers at a specific idler pitch, supporting a specific belt sample clamped to the belt trolleys and loaded with sand bags. Although the steps followed to prepare the test rig for experimentation were never the same, the test procedures that followed were standardised and are presented in the sections to follow.

3.3.3.2 Belt Tensioning Procedure

Once the general test rig set-up has been completed, the belt may be tensioned as follows:

1. Inspect the belt, clamps, trolleys, cables and tensioning frame; ensuring that everything is in order.
2. Check that the mechanical switch on the lever hoist is on the lifting setting and ensure that there is nothing obstructing the movement of its chains.
3. Open the belt tension measuring panel in LabVIEW, ensure that all wiring is properly connected, and start recording the belt tension.
4. Using the lever hoist, slowly increase the belt tension until the load cell reading is approximately 10% greater than the desired tension.
5. Using the test rig control panel, pull belt about 500 mm forwards and backwards, and allow the load cell reading to settle.
6. If the converged tension reading is not consistent with the desired belt tension, use the lever hoist to increase or decrease the tension and repeat step 5.
7. Repeat step 6 until the desired belt tension is achieved.

3.3.3.3 Test Rig Lifting Procedure

The final set up procedure before commencing with an experiment is the lifting of the UDR Test Rig. This is achieved as follows:

1. Inspect the condition of the test rig, making sure that there are no obstructions to, or loose objects on, the rig, that the lifting components are in working order and that the test area is clear of all personnel.
2. Turn the control panel hoist switch clockwise to the lifting setting to engage the lifting winch.
3. Keep the switch actuated until the rig has been lifted just past the desired inclination angle (5° , 10° or 15°). This is evident by the loud noise that the ratchet pawl makes once it has passed one of the pivot arm teeth.
4. Slowly lower the test rig by repeatedly flicking the hoist switch anticlockwise in short intervals to unwind the cables on the lifting winch. Do this until the rig is seen to stop moving – indicating that the pawl has made contact with the ratchet tooth. Be careful not to unwind the cables of the lifting winch any more as this will release the tension in the lifting cables, and potentially cause slipping and damage to the cables.

3.3.3.4 Unidirectional Roller Test Procedure

1. Use the rig control panel to move the belt into the starting position (the limit of the belt stroke, furthest away from the axis of rotation).
2. Open the data acquisition and belt tension measuring panels in LabVIEW.
3. Enter all the test rig set up details and the experiment number into the data acquisition panel.
4. Using LabVIEW's offset nulling function (under the DAQ Assistant calibration tab), zero the force readings of the cantilever load cells.
5. Measure the belt tension (using the dedicated LabVIEW panel) and enter the value into the data acquisition panel.
6. Begin the data capturing process by clicking the "START RUN" button on the data acquisition panel.
7. Turn the travel switch on the control panel anticlockwise to the forward setting (pulling the belt down the incline, towards the axis of rotation) and keep the switch engaged

until about five seconds after initial belt slip. This is evident from the live force graphs on the data acquisition panel.

8. Terminate the data capturing process by clicking the “STOP” button on the data acquisition panel.
9. Reset the belt into the starting position by turning the travel switch clockwise, pulling the belt back up the incline (away from the axis of rotation).
10. Repeat steps 4 to 9 until eight test iterations have been run, remembering to update the iteration number on the panel each time.

3.3.3.5 Test Rig Lowering Procedure

Once an inclined experiment has been successfully conducted, it is important to lower the test rig back down to the horizontal position before any work is done on, or changes made to, the rig. The rig may be lowered as follows:

1. Inspect the area around the test rig, making sure there are no obstructions to the rig and that it is clear of all personnel.
2. Turn the control panel hoist switch clockwise to the lifting setting to engage the lifting winch and keep it engaged for about three seconds, thus separating the pawl from contact with the ratchet tooth.
3. Disengage the ratchet mechanism by pulling on the steel wire rope connected to the pawl, thus rotating it out of the way (passive position).
4. Turn the control panel switch anticlockwise to the lowering setting.
5. Keep the switch engaged until the rig has been lowered to the point where the ratchet pawl is just below the 5° inclination tooth.
6. Engage the ratchet mechanism by pulling the steel wire rope connected to the pawl, thus rotating the pawl into the active position.
7. Lower the rig all the way down, until the pawl makes contact with the 0° inclination ratchet tooth and the hoist limit switch is engaged, thus stopping the hoist motor and preventing any additional tension release in the lifting cables.

3.3.3.6 Shut Down Procedure

Once the test rig has been lowered to the horizontal position, the shut down procedure may commence as follows:

1. Open the tension measuring panel in LabVIEW and start recording the belt tension.
2. Switch the lever hoist to the lowering setting and use the lever to slowly release the tension. Keep doing this until the load cell tension reading has dropped below 300 kgf.
3. If more experiments are to be run, repeat the procedures presented in Sections [3.3.3.1](#) to [3.3.3.5](#). Otherwise, follow the shutdown steps as follows:
 - (i) Power off the rig at the welding plug switch.
 - (ii) Unplug the power cable from the socket, roll the cable up around the control panel and secure the end of the plug to the panel frame using a chain and padlock.

3.3.3.7 Precautions

When executing the aforementioned experimental procedures, the following precautions must be observed:

- Only perform experimental set-up work on the test rig when it is in the horizontal position and the belt tension is below 300 kgf.
- Wear hearing protection (due to the noisy environment of the factory) and steel toe boots when working on the rig.
- Conduct rig inspections before tensioning the belt and lifting/lowering the rig.
- Ensure that the rig is clear of personnel when tensioning the belt, lifting/lowering the rig and when running experiments.
- Make sure that all test rig components are secured in place and that there are no loose objects on the rig before executing a lifting procedure.
- Do not walk underneath the rig when it is inclined.
- Do not stand under any part of the rig during lowering of the frame, particularly while dealing with the ratchet pawl.
- Keep the area around the rig cordoned off at all times so that nobody enters the testing facility during operation.

- Do not leave the testing facility until a shut down procedure has been completed.
- Keep clear of all cables during belt tensioning/pulling and rig lifting/lowering activities.

3.4 Additional Friction Study

During the course of the unidirectional roller investigation, the need to isolate the friction force from the overall resistance measured at the belt-roller interface became apparent. Decoupling the contribution of friction from the other sources of resistance (such as belt sag, indentation resistance and tension effects) helps to explain the differences in unidirectional roller braking performance that results from the variation of friction independent factors (such as idler pitch and troughing angle). Separating friction from the overall resistive force also allows empirically derived trends to be extrapolated to the generic case by adjusting the data according to the friction properties of specific conveyor belt and roller material pairs. This enables the collected resistive force data to be used in the design of unidirectional systems with different conveyor belt or roller types to those tested on the UDR Test Rig.

In order to develop a better understanding of the friction developed between a rubber conveyor belt and a stationary idler, and to determine its contribution to the overall resistive force, an additional friction study was conducted as outlined in the sections to follow.

3.4.1 Research Methods

The additional friction study was conducted in the TUNRA Bulk Solids Africa (TBSA) Laboratory located at the University of the Witwatersrand. The experiments that were run in this lab were adapted from TUNRA's standardised wall friction testing method; which uses a Jenike Direct Shear Tester (JDST) to measure the friction between wear liners and a bulk material sample. This method was adapted by clamping conveyor belt samples to the JDST (in the place of wall liners) and pushing common roller material sample blocks along the belt surface (instead of consolidated rings of bulk material). The measurement technique used to determine the kinetic friction coefficient^c (μ_k) for a specific belt and roller material pair is illustrated in Figure 3.8.

^cRecall the conservative assumption made in Section 3.2 that the static friction limit equals that of the kinetic limit (i.e. $\mu_s = \mu_k = \mu$).

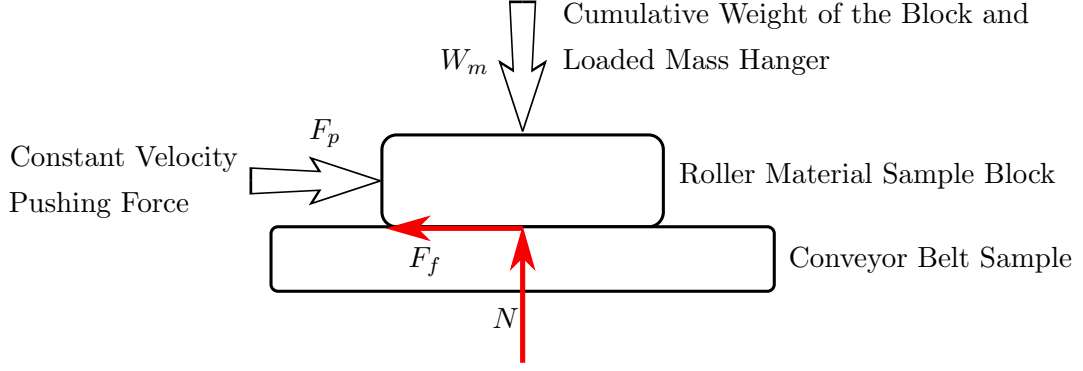


Figure 3.8: Diagram of the friction coefficient measurement technique

When the sample block is pushed at a constant velocity (as shown above), the forces acting on the block are in equilibrium. Therefore, the magnitude of the friction force will be the same as the pushing force ($F_f = F_p$) and the normal force acting at the belt-block interface will be equal to the cumulative weight of the block and loaded mass hanger ($N = W_m$). The apparatus used to conduct the tests (see Section 3.4.2) is capable of measuring the pushing force (F_p) and the loaded block weight (W_m). Since there are no forces which may be considered adhesive in nature, the measurements taken during testing can be used to calculate the coefficient of friction according to:

$$F_f = \mu_k N$$

$$\therefore F_p = \mu_k W_m$$

Finally, recalling that $\mu_s = \mu_k = \mu$:

$$\therefore \mu = \frac{F_p}{W_m} \quad (3.11)$$

3.4.2 Apparatus

Each friction measuring experiment was conducted using a Jenike Direct Shear Tester (JDST) and a mass hanger. The JDST (shown in Figures 3.9 and 3.10) was equipped with an extendible loading stem connected to a load cell, thus allowing for shear force measurements to be recorded. The mass hanger was also equipped with a load cell which, given the known masses of the roller material sample (see Table 3.1) and the hanger (451.8 g), allowed for the measurement of applied normal loads. With the sample dimensions known, contact pressures relating to the different loading cases could also be inferred. The applied loads were varied between 0 and 20 kg and the frictional resistive force recorded for each normal load.

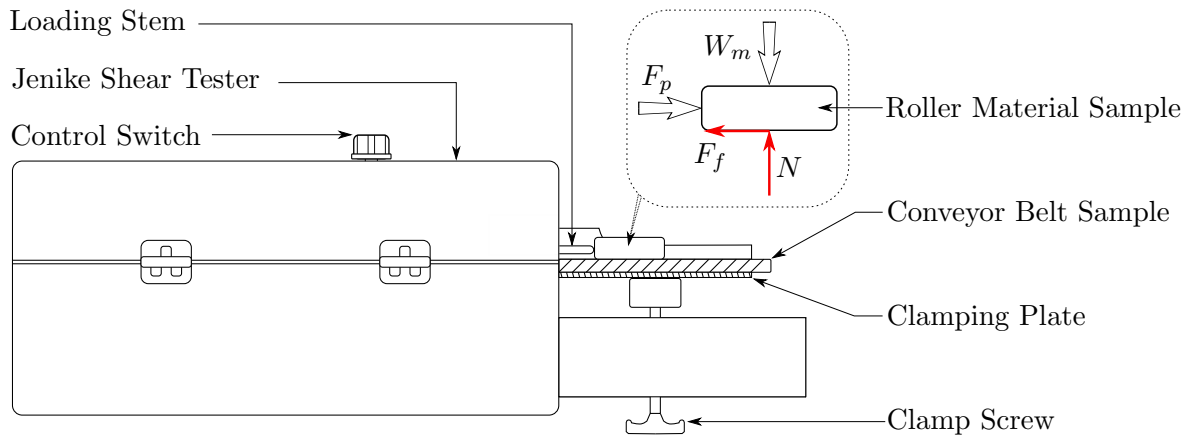


Figure 3.9: Annotated diagram of the test apparatus

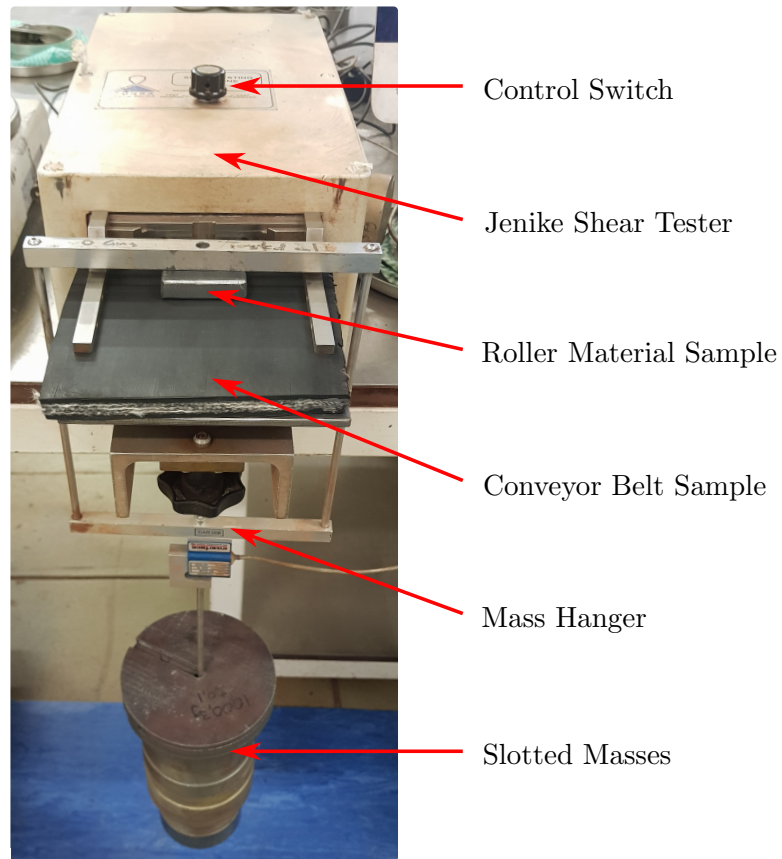


Figure 3.10: Labelled photograph of the friction study apparatus

Friction measurements were recorded using four common roller materials (Table 3.1) paired with four different conveyor belt samples (Table 3.2). In the case of the steel roller material sample, two surface preparations were considered: a rough surface (mimicking that of a

painted steel roller) and a smooth surface (for the case of a steel roller whose paint layer has worn away^d). The mass values in Table 3.1 represent the mass of the roller material sample.

Table 3.1: Roller material sample details

Sample Material	Mass (g)	Contact Area (mm ²)	Contact Pressure Range (kPa)
Steel (Smooth)	223.5	2500	10.7 – 82.6
Steel (Rough)	223.5	2500	10.7 – 82.6
Nylon	75.7	3000	8.5 – 68.8
HDPE	28.4	2900	8.1 – 66.8

Table 3.2: Conveyor belt sample details

Sample No.	Carcass	Cover	New/Used
1	3-ply Fabric	Nitrile Rubber	New
2	Solid Woven PVC	Nitrile Rubber	New
3	Solid Woven PVC	Nitrile Rubber	Used
4	Steel Cord	Type M Rubber	New

3.4.3 Experimental Procedure and Precautions

By varying the aforementioned belt and roller material pairs and changing the belt preparation (wet, dry, dirty and clean belt surfaces) an in-depth investigation was undertaken. A total of 25 coefficient of friction measuring experiments were conducted with two-to-three runs per experiment (see Appendix A). The procedure to follow when running these tests is presented in the following section.

^dThe case of a steel roller whose surface has been smoothed during operation, and the paint layer subsequently worn off, was not considered from the start of the project. Instead, this issue was discovered whilst conducting unidirectional roller experiments. As such, initial experimentation on the UDR test rig did not take the “paint-less” steel rollers case into account. However, after the realisation of this important scenario, subsequent tests included steel rollers without a paint layer.

3.4.3.1 Procedure

1. Clamp the conveyor belt sample to the JDST.
2. Extend or retract the loading stem to a suitable starting position.
3. Place the roller material sample onto the belt, just in front of the loading stem.
4. Place the hanger on top of the roller sample and load it first with eight 2 kg slotted masses and then with four 1 kg slotted masses.
5. Start the test by turning the control switch to the slow extension setting.
 - The JDST will begin to push the roller material sample at a speed of 2.5 mm/min across the belt.
6. When the measured shear force (F_p) output stabilises, start the data capture program.
7. Allow 30-40 seconds of data to be recorded and then remove a slotted mass. Continue to remove slotted masses every 30-40 seconds until there is a single 2kg mass left on the hanger^e.

3.4.3.2 Precautions

When executing the above procedure, the following precautions must be observed:

- Clamp the belt sample to the JDST with the correct orientation; ensuring that the sample block is pushed along the longitudinal warp direction of the belt.
- Position the sample block such that the loading stem acts perpendicularly to, and in the middle of, the rear face of the block. This is important to ensure that the sample block is pushed straight.
- Modify the leading edges of the sample blocks by chamfering them, thus preventing any sharp edges from digging into the belt as it is pushed along.
- Wear safety boots so as to avoid injury when working with the heavy slotted masses.
- Do not line up the slots of the masses when loading the mass hanger.
- Only begin data recording when the measured shear force is seen to stabilise.
- Take care not to bump the mass hanger when removing slotted masses. Use your hands to dampen any swaying motion of the hanger if necessary.

^eNote that the sequence of the normal loads applied to the roller material samples is 20, 19, 18, 17, 16, 14, 12, 10, 8, 6, 4, 2 kg.

Chapter 4 Results

4.1 Unidirectional Roller Results

Before the results of the various unidirectional roller experimental programmes are presented, a few typical examples of the data collected during a UDR Test Rig experiment will be considered. One such example of collected test data is presented in Figure 4.1.

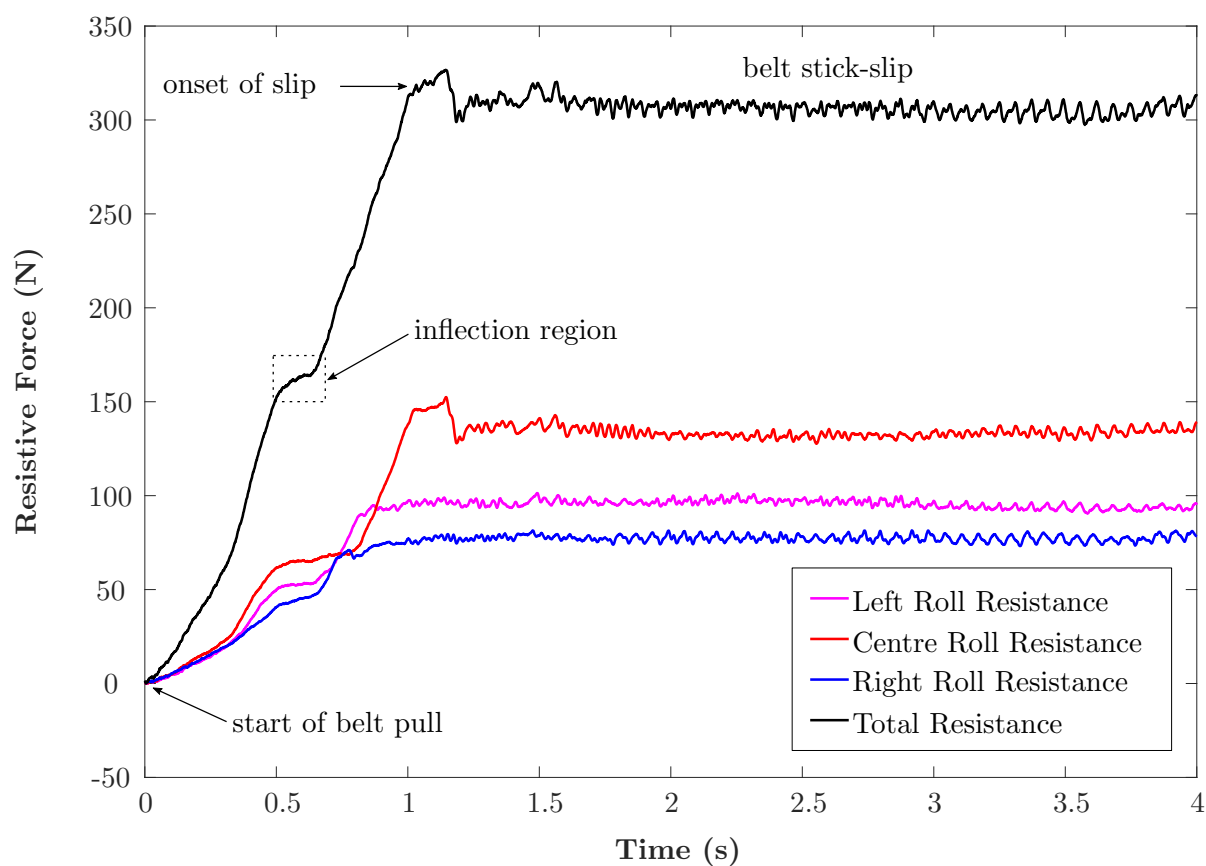


Figure 4.1: Typical unloaded belt test run resistive force graph (Experiment A26, Run 1)

This figure illustrates the change in resistive force over time, starting at the instant of initial belt pull. The individual resistances for the left wing, right wing and centre rollers (averaged over the two measured idler sets) are plotted together with the overall total idler resistance. A graph such as this was produced for each test run conducted and was examined after each experimental iteration to ensure that the recorded data adhered to the typically observed format and that there were no issues with reading in the load cell data.

Figure 4.1 highlights the difference in resistive force response at the centre rollers as compared with the wing rollers. Overall it can be seen that higher resistive forces develop at the centre roller contact area. However, the experiment summarised in this graph was set up with an empty conveyor belt (i.e. no material load). To see how loading affects the resistive response, with particular attention to the relationship between centre and wing rollers, refer to Figure 4.2. This graph summarises a typical test run of an experiment similar in set up to that presented in Figure 4.1, except for the inclusion of belt loading.

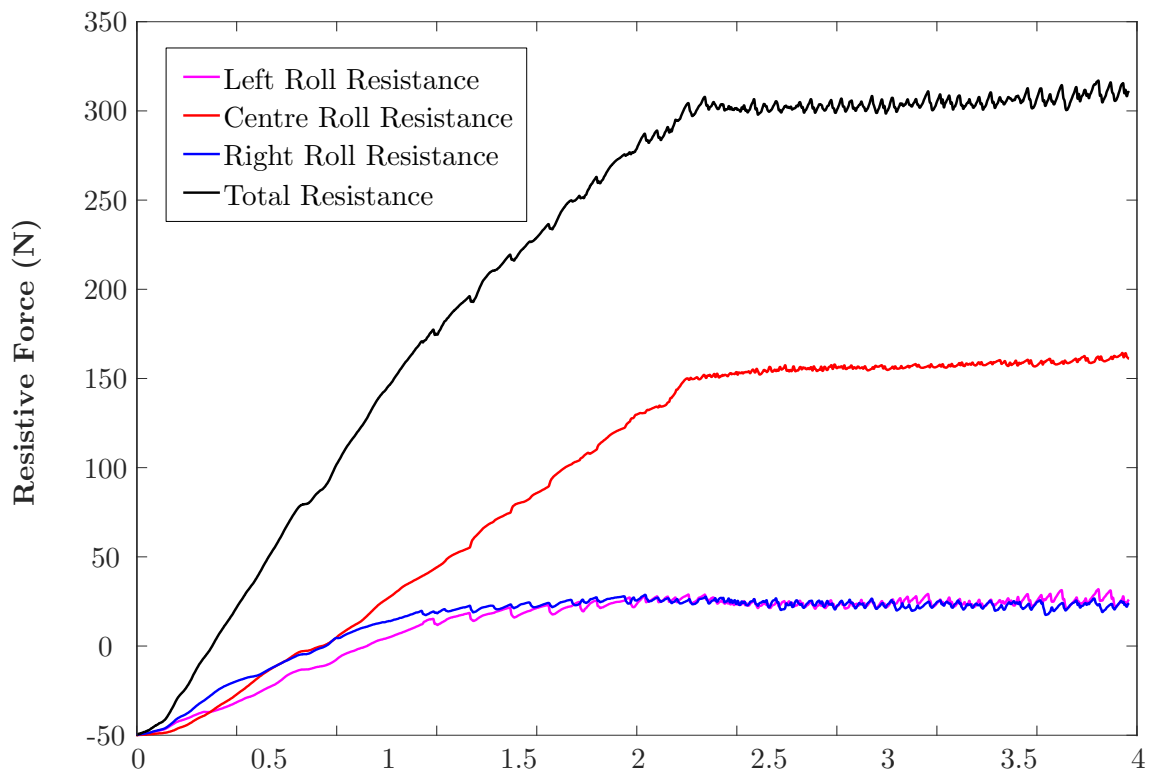


Figure 4.2: Typical loaded belt test run resistive force graph (Experiment D26, Run 1)

Once all the test runs of a particular experiment had been conducted, the total resistive force data for each run were plotted together for comparison. A typical example of this plot is shown in Figure 4.3.

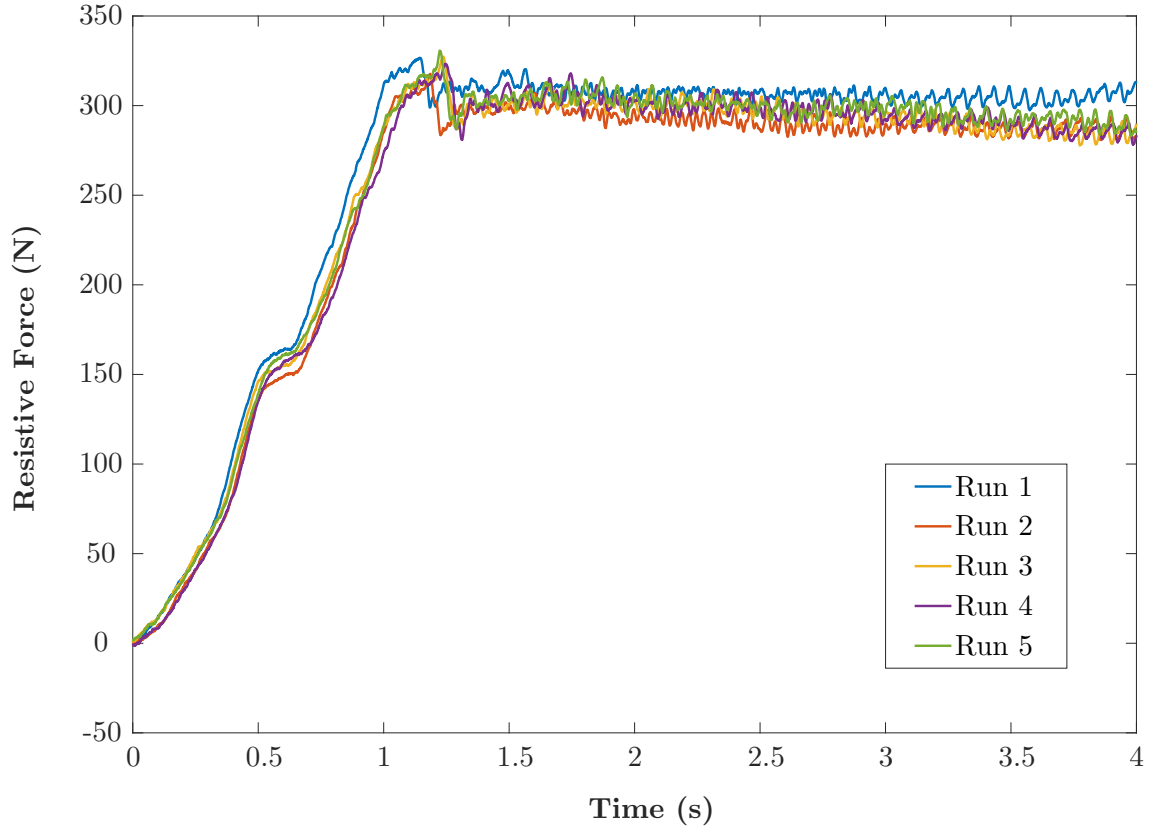


Figure 4.3: Typical five run resistive force graph (Experiment A26)

These graphs were generated for every experiment conducted with the UDR Test Rig in order to assess the repeatability of each experiment and to identify any experimental outliers. In the case of an outlier being identified, an additional test would be run and used in place of the outlier. Each unidirectional roller experiment consisted of five experimental iterations in order to infer the variability of the force measurements and to increase the accuracy of averaged slip force estimates^a.

From the five test run resistive force readings, a single resistive force relationship was derived for each experiment by averaging the data from all iterations. This was done after setting the time offsets of the five data sets so that the start of the increase in tension of each experiment coincided. An example of the overall average resistive force plot that results is shown in Figure 4.4.

^aIt may be noted in Figure 4.3 that the first experimental run deviated slightly from the subsequent runs. Although the source of this deviation is unknown, the effects of this were inconsequential because the difference was very small and because the friction response was averaged over all five experimental results and then used to calculate the mean slip force (the target measurement of this research) over an interval of three seconds.

This graph shows that the measured resistive force increases with increasing belt pulling force until a peak value is reached after which belt slip occurs. The oscillatory nature of the resistive force experienced after the peak is representative of a typical stick-slip motion of the belt as it moves over the locked idlers^b.

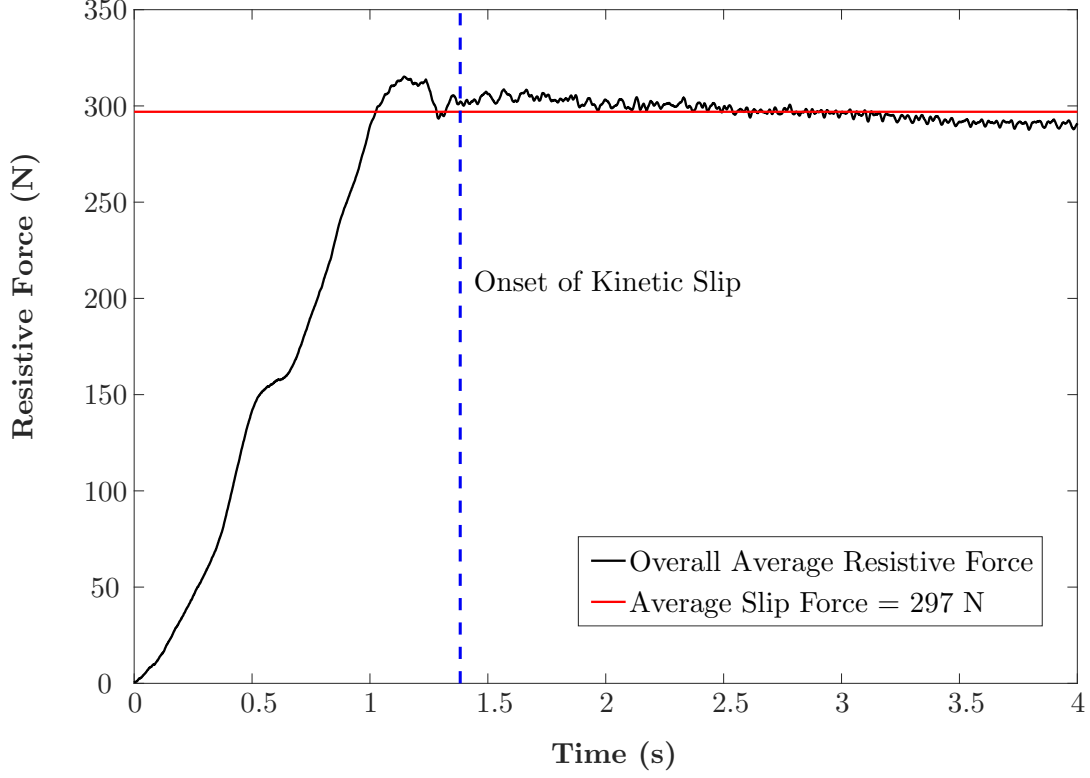


Figure 4.4: Typical averaged resistive force graph (Experiment A26)

It may be noted that in the study of friction, stick-slip is defined as a “relaxation-oscillation phenomenon”, characterised by friction force portraying a sawtooth curve over time [36] (see Figure 4.1). It is widely believed that this sawtooth pattern is linked to a decreasing coefficient of friction with increasing velocity [37]. This takes the form of the friction force rising during periods of deceleration, until the ceasing of relative motion, followed by the static friction force being exceeded and stored potential energy being released, thus resulting in a rapid acceleration and subsequent lower sliding friction coefficient. Immediately after this, the rapidly slipping surface catches up to the opposite surface and begins to decelerate, thus resulting in the rise of friction force once again [36]. This process then repeats itself, resulting in the sawtooth shaped friction profile.

^bThe source of the apparent drop-off in slip force over time as observed in Figure 4.4 is unknown. It is also not a general trend, as is observable in Appendix E.2 where the slip force measurements in some experiments are seen to drop-off, others rise, but the majority to remain constant over time. Additionally, even those experiments characterised by an increase or decrease in slip force over time, the magnitude of this deviation is consistently slight and therefore of no significant consequence.

From the averaged resistive force graphs (such as the one presented in Figure 4.4), the mean slip force for each experiment could be determined. This was achieved through a simple MATLAB program which calculated the average slip force over an interval of three seconds, beginning at the user defined onset of kinetic slip (indicated by the blue dashed line on the graph). The onset of kinetic slip refers to the point in time, just after the maximum static resistive force has been overcome, when the belt begins to slip over both the locked idler sets at a constant speed.

4.1.1 Factor Screening Experimental Results

In order to determine which factors and combinations of factors contribute most significantly to the resistive force developed at the contact between the conveyor belt and a locked roller, a factor screening experimental programme was conducted. This study (Experimental Programme A) measured the belt slip forces that resulted from varying the levels of inclination angle (α), idler pitch (a), roller type (RT), roller diameter (d), troughing angle (β) and belt tension (T) as shown in Table A.1, Appendix A. The results of this investigation were summarised according to the computed average belt slip forces. Figure 4.5 presents these slip forces, plotted according to their corresponding experiment number.

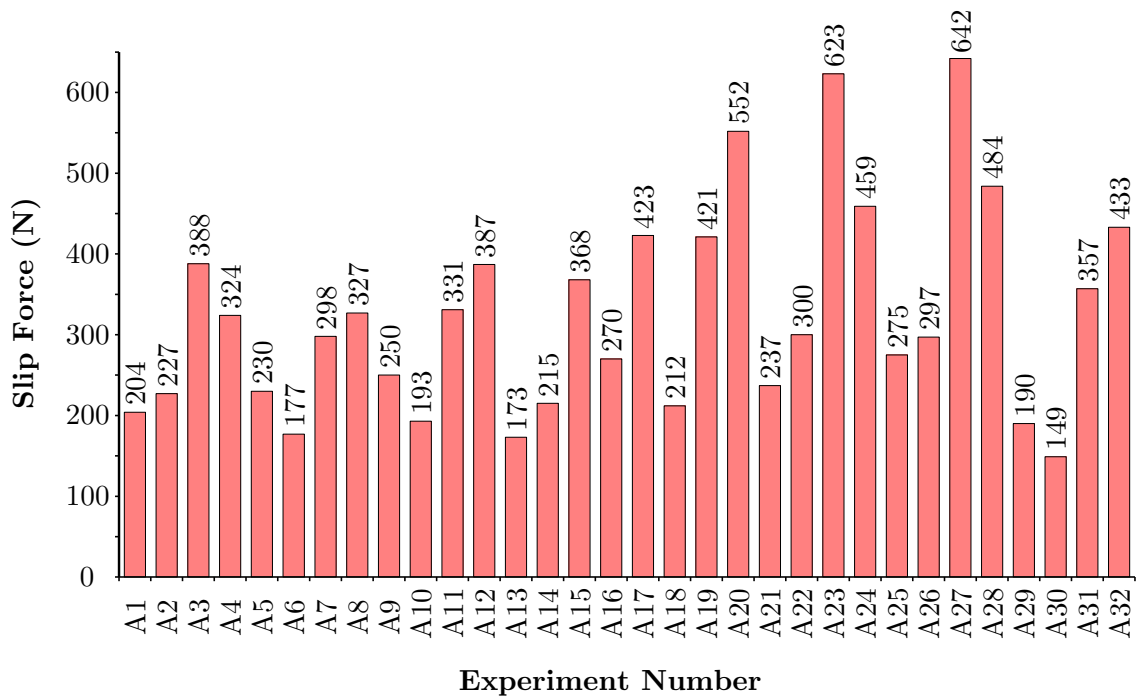


Figure 4.5: Overall average kinetic slip forces (Experimental Programme A)

Using the determined experimental slip forces, an Analysis of Variance (ANOVA) table was drawn up for the factor screening programme (see Table 4.1). This table presents the sum of squares, F-values and, ultimately, the P-values for each conveyor belt factor varied and for each two-factor interaction as well.

Table 4.1: Analysis of variance (ANOVA) table for Experimental Programme A

Source of Variation	Sum of Squares	Degrees of Freedom	F-value	P-value $P(F > F_{\text{crit}})$	
Inclination (α)	5542	1	2.682	0.133	
Idler Pitch (a)	268173	1	129.8	0.476×10^{-6}	***
Roller Type (RT)	20199	1	9.775	10.75×10^{-3}	*
Roller Diameter (d)	4761	1	2.304	0.160	
Trough (β)	90755	1	43.92	58.75×10^{-6}	***
Belt Tension (T)	51464	1	24.91	0.545×10^{-3}	***
$\alpha : a$	0	1	0.1×10^{-3}	0.991	
$\alpha : \text{RT}$	322	1	0.156	0.701	
$\alpha : d$	238	1	0.115	0.742	
$\alpha : \beta$	803	1	0.389	0.547	
$\alpha : T$	3560	1	1.723	0.219	
$a : \text{RT}$	36	1	0.017	0.898	
$a : d$	669	1	0.324	0.582	
$a : \beta$	23459	1	11.35	7.129×10^{-3}	**
$a : T$	1953	1	0.945	0.354	
$\text{RT} : d$	11412	1	5.523	40.64×10^{-3}	*
$\text{RT} : \beta$	3183	1	1.540	0.243	
$\text{RT} : T$	978	1	0.474	0.507	
$d : \beta$	4800	1	2.323	0.158	
$d : T$	1113	1	0.539	0.480	
$\beta : T$	6269	1	3.034	0.112	

This table also shows, by way of asterisks, which factors or interaction pairs are deemed significant according to the probability that the data could deviate from the null hypothesis. The more asterisks, the more significant the effect. In this case, it was assumed that a factor or interaction is significant if its corresponding P-value was less than 0.05 (i.e. $P(F > F_{\text{crit}}) < 5\%$). This results in a 95% confidence level for the identification of significant factors and interaction pairs.

Following on from the ANOVA table, the comparative significance of the different factors and interactions could be graphically presented by means of a half normal plot. This plot, as seen in Figure 4.6, summarises the absolute effects and corresponding half-normal scores for each source of variation (i.e. each factor and interaction combination). If the source of variation complies with the null hypothesis, then its associated point on the graph will fall close to the insignificance line (shown as a solid straight line in the figure). If, however, the source of variation deviates from the null hypothesis, then its point will fall noticeably off the straight line. The further the point is from this line, the greater the significance of the corresponding source of variation. With this in mind, it may be seen from Figure 4.6 that the three most significant factors as determined through the factor screening experiments are idler pitch, troughing angle and belt tension.

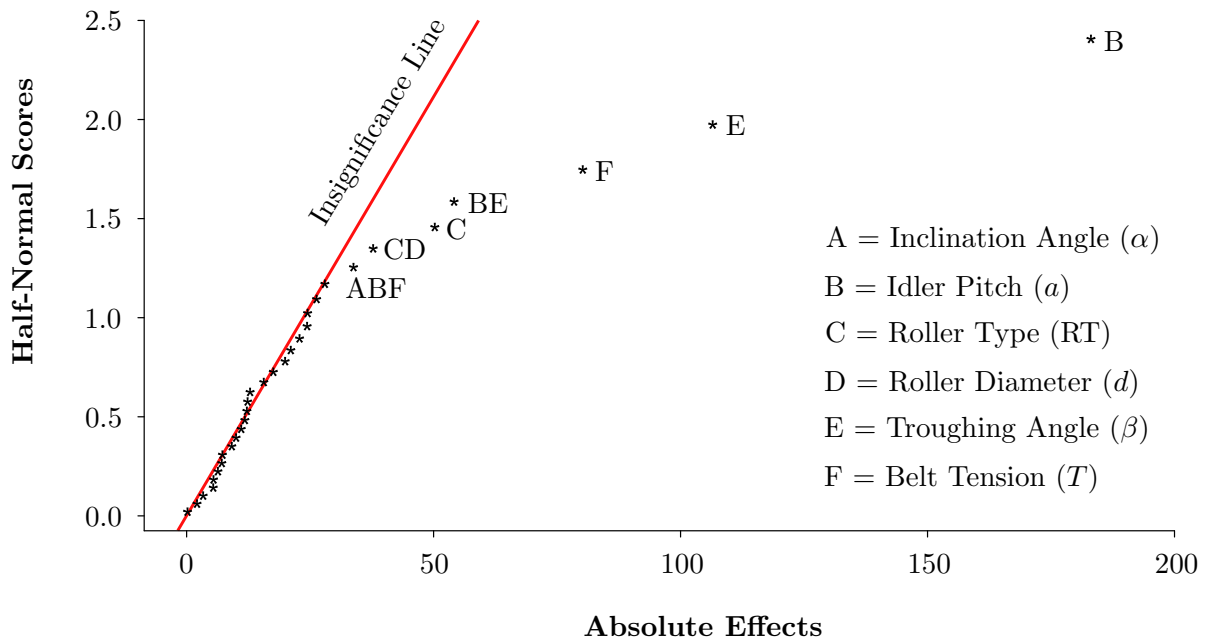


Figure 4.6: Half normal plot for Experimental Programme A

Although the ANOVA table and the half normal plot demonstrated the significance of the sources of variation, the actual nature of the main and interaction effects required further investigation. The comparative plot of the main effects observed in Experimental Programme

A is presented in Figure 4.7. This figure shows the magnitude and direction of each factor main effect in ascending order. The main effect of any factor is the average amount that the measured output will increase/decrease by when varying the factor from a low to a high level.

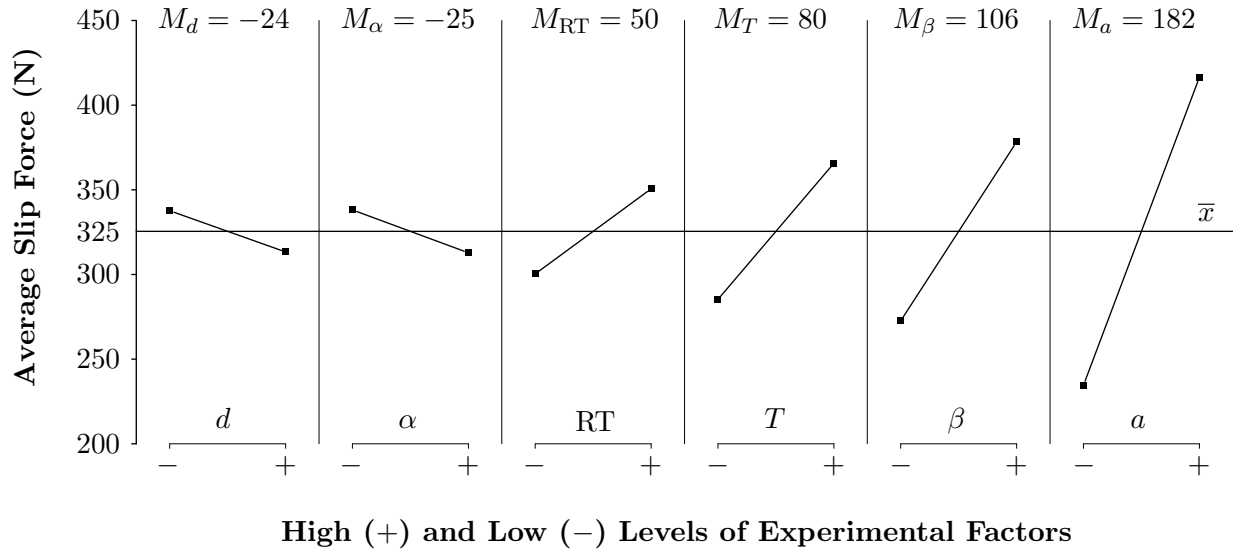


Figure 4.7: Main effects plot for Experimental Programme A

In terms of the interaction effects, the previously introduced half normal plot (Figure 4.6) shows that the only significant interaction pairs were the roller type-diameter pair and the pitch-trough pair. With this in mind, the interaction plots for these two interactions were plotted as shown in Figure 4.8. In this figure, the significance of the two interactions is evident when examining the difference in gradient between the interaction effect lines.

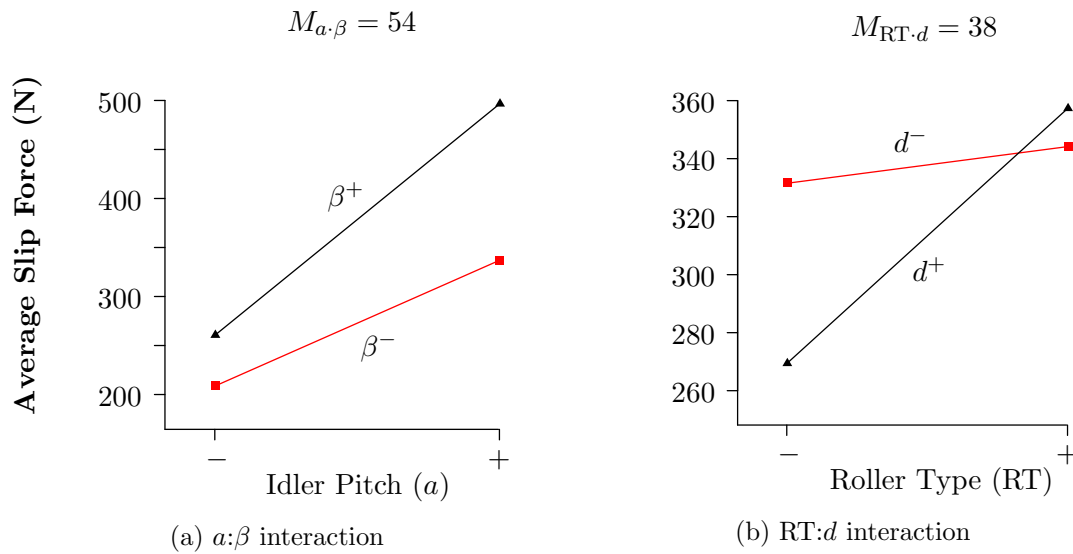


Figure 4.8: Significant interaction effect plots for Experimental Programme A

Considering the pitch-trough combination (Figure 4.8a), the interaction effect plot shows that the average increase in belt slip force experienced when increasing the idler pitch is greater at high levels of troughing angle (β^+) than at low levels (β^-). This is evident by the steeper gradient of the black β^+ line as compared to the red β^- line. Similarly, Figure 4.8b shows that the effect of roller type on belt slip is greater when rollers with larger diameters are installed.

After the collected data was processed and analysed using the various DOE techniques, and the significance of the different sources of variation determined, a bar chart of the slip forces observed in Experimental Programme A was plotted with the experiments grouped according to the three most significant factors: idler pitch, troughing angle and belt tension. This bar chart, presented in Figure 4.9, shows the approximate slip force trend regardless of the inclination angle, roller type or roller diameter. From this figure it can be seen that increasing the belt tension, idler pitch or troughing angle will result in a significant increase in the resistive force experienced during belt slip. Viewed another way, the resistive force of the idler on the belt motion is reduced for lower belt tensions, idler pitched and troughing angles.

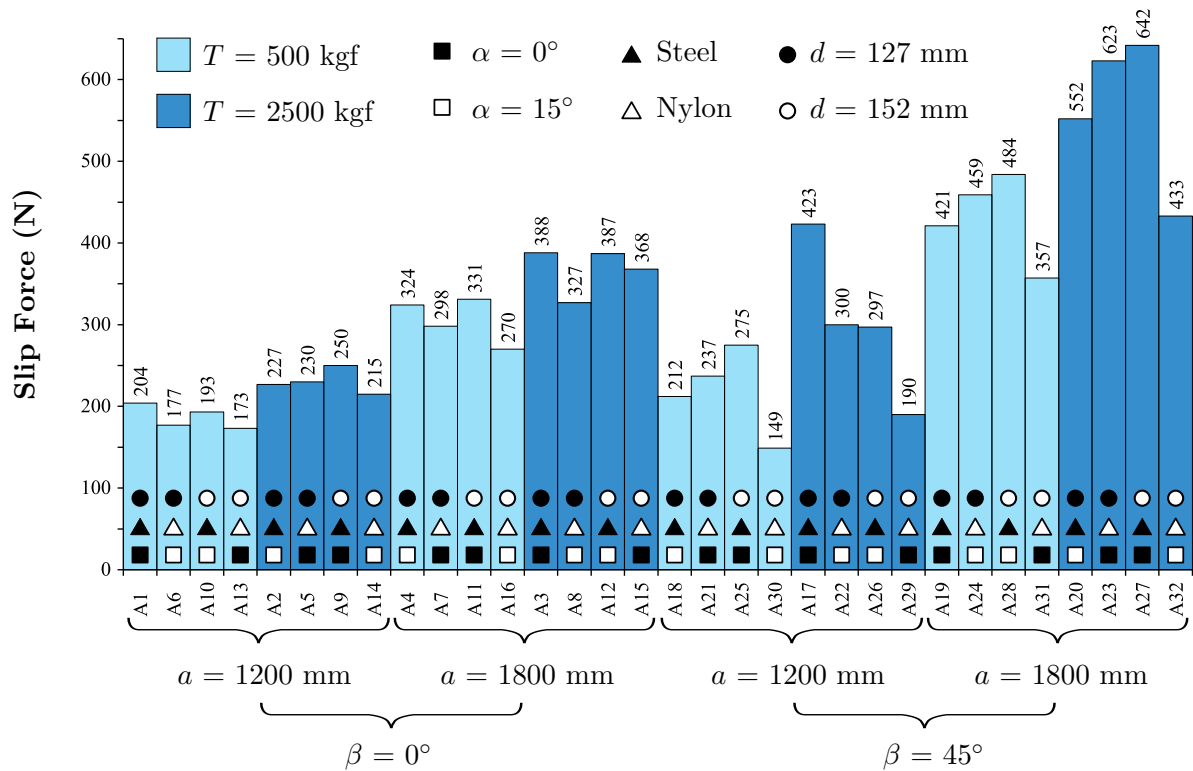


Figure 4.9: Slip forces grouped according to tension, pitch and trough (Programme A)

4.1.2 Focused DOE Test Results

Based on the results of the factor screening experimental programme, a series of additional experiments was conducted using the UDR Test Rig in order to develop a better understanding of how the various factors affect unidirectional idler performance. The implementation of supplementary experiments also allowed for a larger data set to be used to develop a unidirectional idler quantity specification model. For experimental set-up details of the additional research programmes conducted, refer to Tables [A.2](#) - [A.10](#).

Before the results of the various focused experimental programmes are presented, it is important to note that in the context of unidirectional idlers, performance is driven by the resistive force developed between a locked roller and the conveyor belt. When it comes to unidirectional idlers, low slip forces are undesirable and conditions that cause it should be avoided. Conversely, high slip forces are desirable and conditions that promote it should be implemented. The higher the slip force limit of a single unidirectional idler, the lower the total required quantity of these devices will be.

Experimental Programme B was designed to gain further insight into the effects of the type of roller and the level of belt tension. Whereas the factor screening investigation only compared steel and Nylon rollers cycled between two tension states, Programme B introduced HDPE rollers and a medium belt tension level. The measured slip forces from this mini-investigation are presented in Figure [4.10](#). This chart highlights the observation that a significantly lower force is required to pull the belt over locked HDPE rollers as compared with steel and Nylon. This finding represented a worst case scenario that needed to be explored further through numerical simulation.

In addition to this bar chart, typical resistive force graphs for an HDPE idler test are presented in Figure [4.11](#). This figure has been included in the results section in order to showcase the difference in the typical resistive force response of an HDPE roller as compared with a steel or Nylon roller during belt slip. The main difference observed in this HDPE response graph is the more defined and proportionally greater resistive force peak experienced at belt slip. This observation indicates the difference between static and kinetic slip force to be greater for HDPE rollers than for steel and Nylon.

The next study conducted (Programme C) was intended to test one of the main findings of the factor screening investigation: that the effect of inclination on the measured slip force is insignificant. In this experimental programme, the inclination angle of the test rig was varied, and all other belt conveying factors (including idler pitch, troughing angle and belt tension) were kept constant. The results of this short validation study are summarised by the transient resistive force plot and kinetic slip force bar chart presented in Figure [4.12](#).

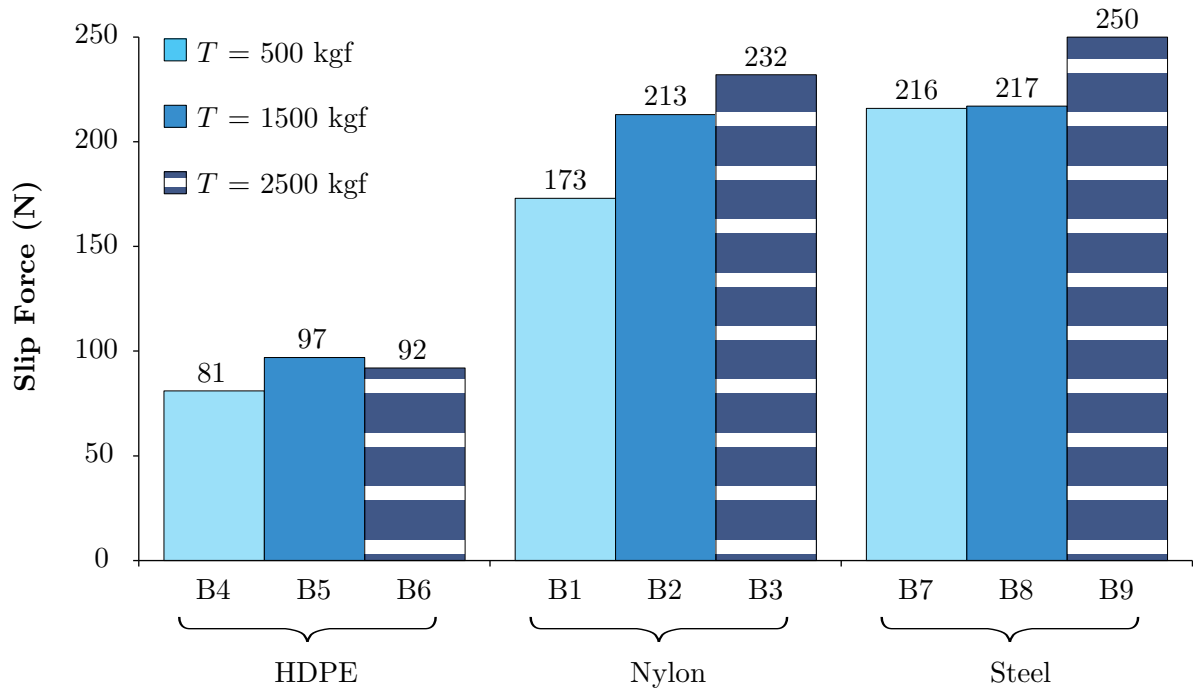


Figure 4.10: Kinetic slip forces (Experimental Programme B)

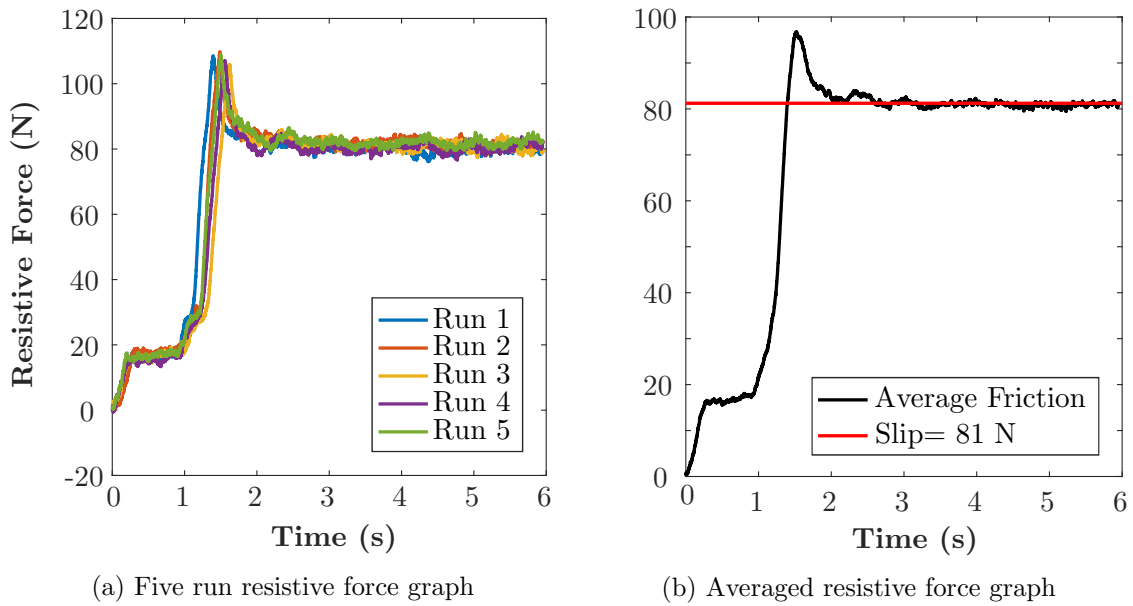


Figure 4.11: Typical resistive force graphs (Experiment B4)

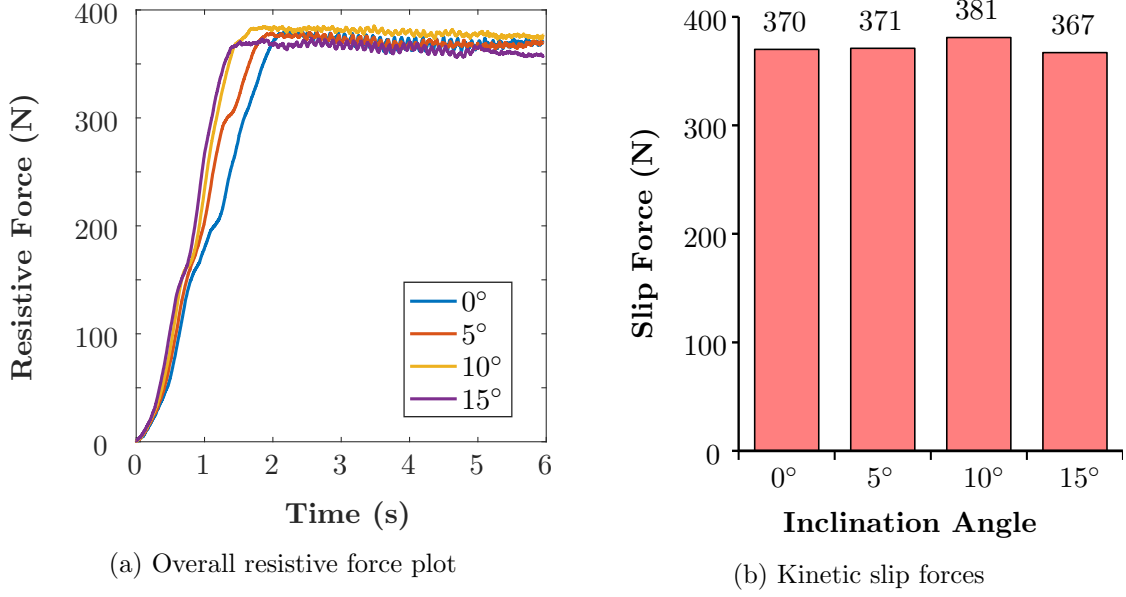


Figure 4.12: Resistance plots for Experimental Programme C

At this point in the investigation, it was discovered that the steel idlers used in the test rig were not totally representative of those in operating conveyor belt systems. This is because the paint applied to a roller in the manufacturing process is likely to wear away very quickly when in operation. The steel rollers that had been installed in the rig were new and still had a layer of paint covering them, therefore resulting in a totally different friction coefficient to what would be expected when this paint wears away. To address this issue, Experimental Programme D was designed to incorporate paintless rollers into the investigation, where the paint was removed from the rollers and the surfaces polished.

In addition to this, Experimental Programme D introduced the factor of belt loading for the first time, whilst also varying the three most significant factors as determined through the factor screening analysis: idler pitch, troughing angle and belt tension.

Using this data, a half normal plot was generated in order to reassess the significance of the belt conveying parameters. This was important due to the inclusion of belt loading as a source of variation in this particular investigation. The resulting half normal plot is presented in Figure 4.13. This plot confirmed that idler pitch and troughing angle were significant factors, and also showed the main effect of material load, and the interactions of pitch-load and load-trough, to be significant.

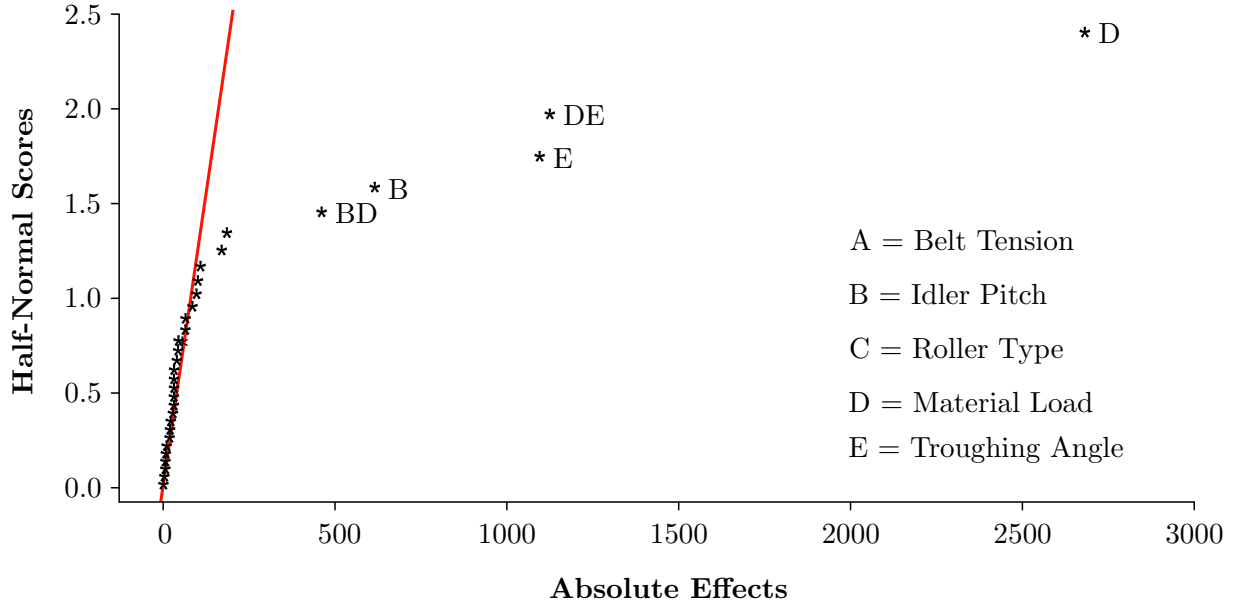


Figure 4.13: Half normal plot for Experimental Programme D

To supplement the half normal plot, a comparative main effects plot and two interaction effects plots were drawn up for Experimental Programme D. The main effects plot ranks the conveyor belt factors from least to most significant and is presented in Figure 4.14.

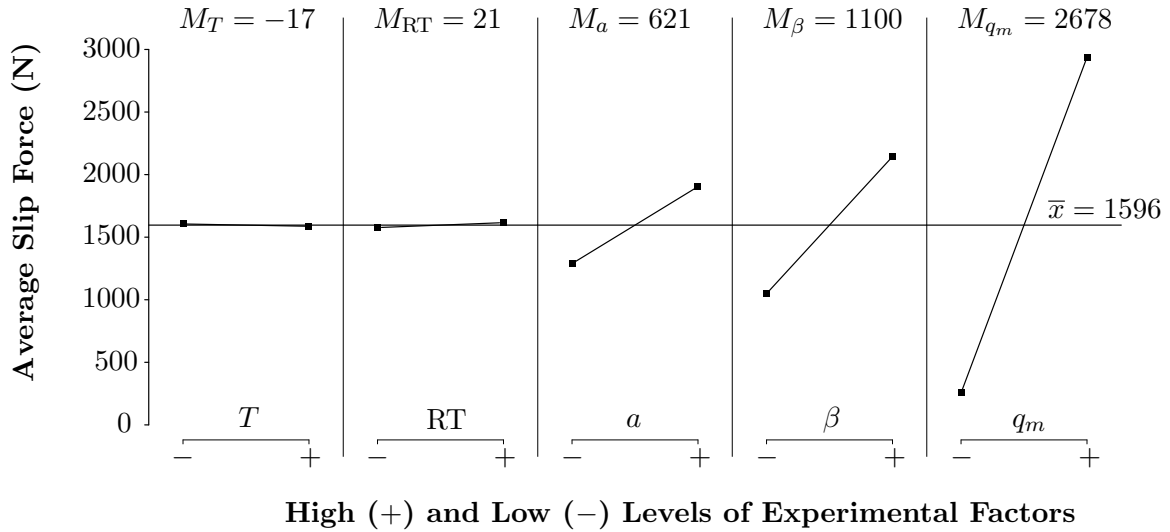


Figure 4.14: Main effects plot for Experimental Programme D

This figure reveals that the three most influential factors varied in this programme (idler pitch, troughing angle and material load) all have a positive effect on belt slip. In other words, an increase in any of these three factors results in an increase in resulting slip force.

The two interaction effects plots (Figure 4.15) demonstrate the close connection between the two factor combinations deemed to have the most significant influence on belt slip. From these plots it can be seen that material load has a greater impact on belt slip when the belt is troughed and the idler pitch is large.

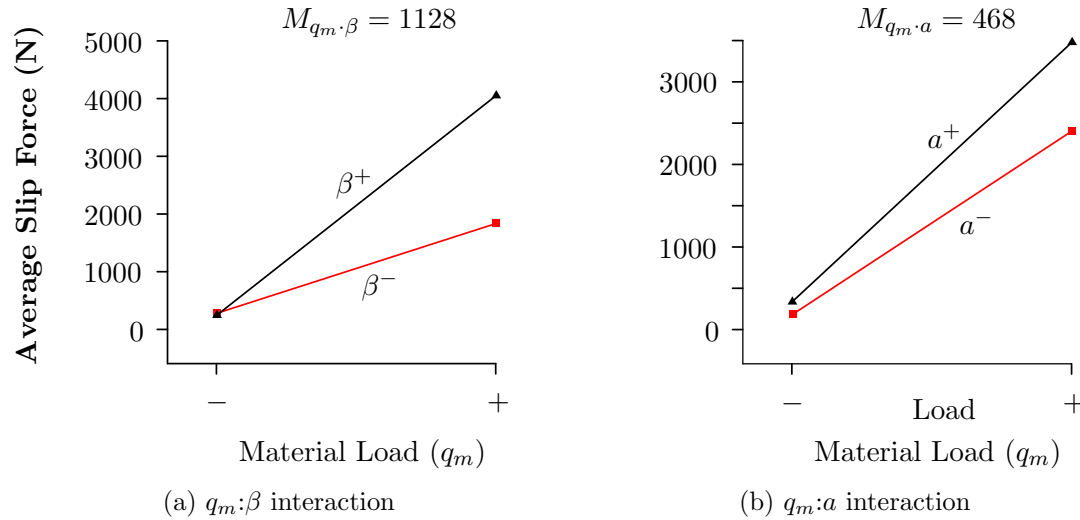


Figure 4.15: Significant interaction effect plots for Experimental Programme D

After all of this analysis, the slip force bar chart for Programme D could be revisited and replotted with the experiments grouped according to the most significant factors. In this case, the revised results were split into two separate charts for each loading condition. The slip forces recorded in the unloaded and loaded belt cases are presented in Figures 4.16 and 4.17 respectively. These figures illustrate the difference in slip force behaviour experienced when the belt is and is not loaded. In the case of an empty belt, increasing idler pitch has the greatest positive effect whilst the variation of troughing angle, steel roller surface preparation and belt tension have a much smaller effect. In the loaded case, however, increasing the troughing angle has the greatest impact, with idler pitch having the second greatest influence and the other sources of variation having almost no effect.

Following this, Experimental Programme E was designed to expand on the three most significant belt conveying factors as identified through factor screening experiments (i.e. idler pitch, troughing angle and belt tension). In the case of the first experimental programme, each factor was varied about two levels (low and high), whereas in Programme E, medium levels of idler pitch, troughing angle and belt tension were added. The factor levels varied included idler pitches of 1200, 1500 and 1800 mm, troughing angles of 0° , 35° and 45° , and belt tensions of 500, 1500 and 2500 kgf. This was done in order to focus the experimentation on the significant sources of variation and to develop a better understanding of the relationship between these factors and the performance of unidirectional idlers.

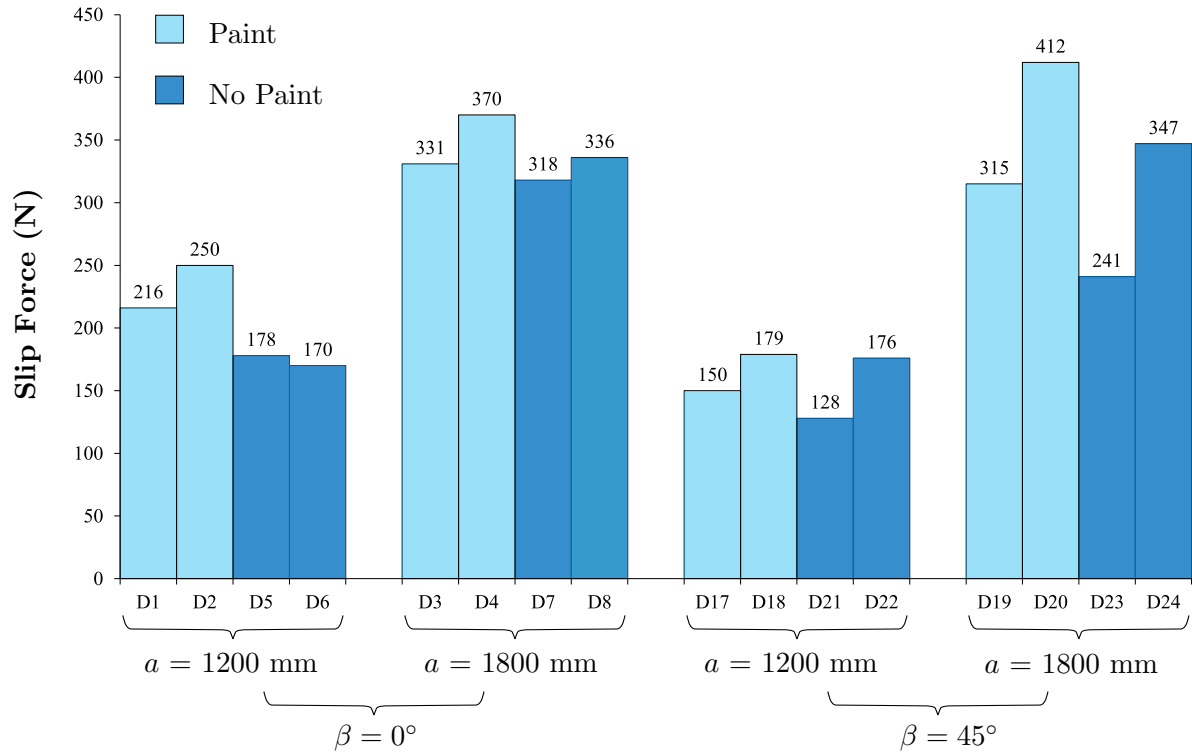


Figure 4.16: Unloaded slip forces grouped according to pitch and trough (Programme D)

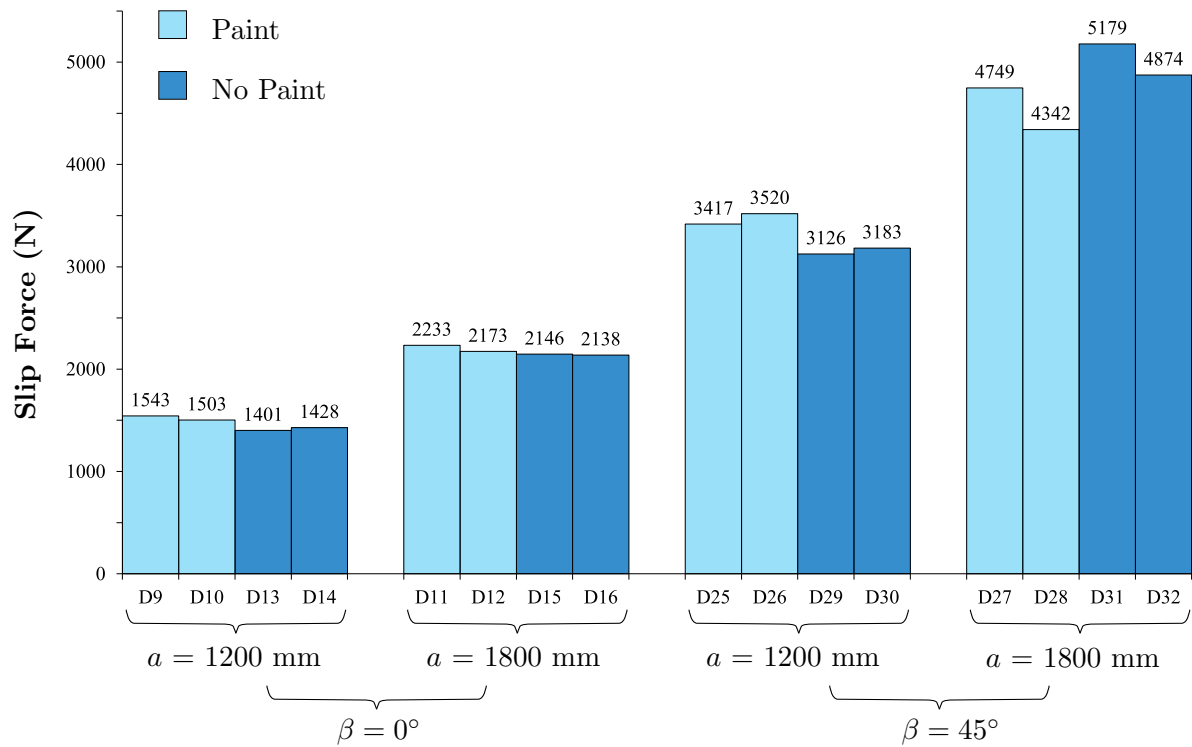


Figure 4.17: Loaded slip forces grouped according to pitch and trough (Programme D)

The belt slip trends observed throughout Experimental Programme E are summarised by the three-dimensional dot plot presented in Figure 4.18. In this figure, the blue triangles, red squares and black circles relate to the low, medium and high belt tension states respectively. From this figure it can be seen that belt tension is considerably less significant than idler pitch and troughing angle. As such, the effects of belt tension could be disregarded, and focus could be dedicated to idler pitch and troughing angle. With this in mind, average kinetic slip forces for each factor level combination of idler pitch and troughing angle were calculated (as summarised in Table 4.2) and a kinetic slip force surface was plotted (as seen in Figure 4.19) to better visualize the slip force trends.

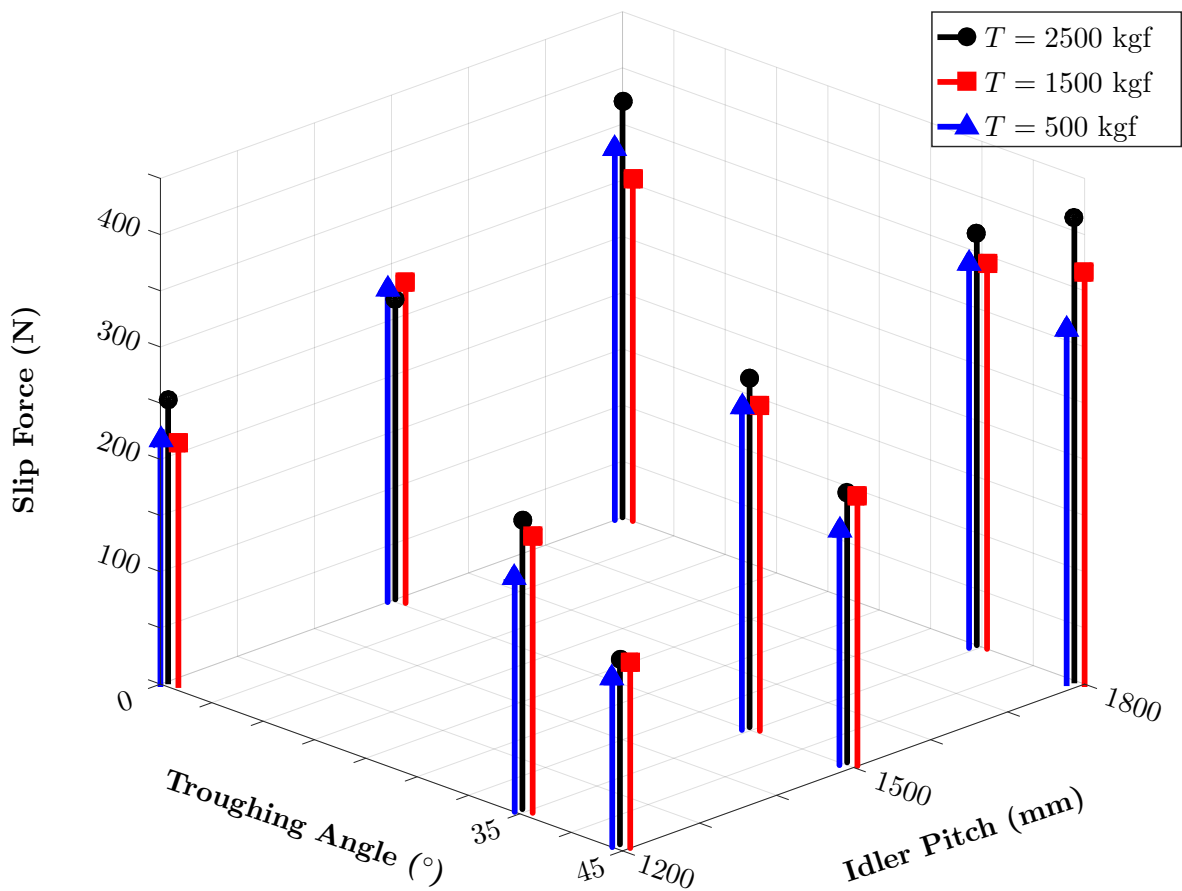


Figure 4.18: Three-dimensional dot plot of kinetic slip forces according to idler pitch, troughing angle and belt tension (Experimental Programme E)

Table 4.2: Kinetic slip forces averaged across belt tension states (Programme E)

		Idler Pitch (mm)		
		1200	1500	1800
β	0°	$F = 227 \text{ N}$	$F = 277 \text{ N}$	$F = 335 \text{ N}$
	35°	$F = 238 \text{ N}$	$F = 296 \text{ N}$	$F = 351 \text{ N}$
	45°	$F_{\min} = 160 \text{ N}$	$F = 230 \text{ N}$	$F_{\max} = 365 \text{ N}$

This three-dimensional graph clearly illustrates the best and worst case scenarios in terms of resulting belt slip. It is important to reiterate at this point that in the context of this project, a low slip force is not desirable (i.e. worst case) as it represents a situation of potential belt runaway in the event of a belt snap. It is therefore preferable to maximise slip forces. With this in mind, Figure 4.19 reveals the best case to be when idler pitch and troughing angle are at their highest (1800 mm and 45° respectively), and the worst case to be when the idler pitch is 1200 mm and the belt is troughing angle is 45°.

The next study that was conducted on the UDR Test Rig, Experimental Programme F, was intended to incorporate belt loading into a focused study of the three most significant factors (pitch, trough and tension). This investigation was very similar to Programme E, except for the inclusion of belt loading and the exclusion of medium factor levels. In this programme, the loaded experiments amounted to a uniform loading of 73 and 170 kg/m of belt length for the flat and 45° troughed belt cases respectively.

The slip forces collected from this study are shown in Figure 4.20. In this graph, the experimental results have been grouped according to relative significance of the various factors being varied. This figure reiterates the previously observed trend of slip force increasing with idler pitch, material load and, in the case of a loaded belt, troughing angle.

Following this, Experimental Programme G was conducted in order to gain further insight into the braking performance differences between the different material constructions of rollers. In this study, HDPE, painted steel and polished steel rollers were installed in idlers separated by pitches of 1200 and 1800 mm, and supporting loaded and unloaded belts. This study was also used to collect more data, and therefore develop a better understanding and ultimately allow for numerical modelling of the 35° belt trough case (which up to this point had only been tested in nine experiments). The slip forces from this programme are outlined in Figure 4.21. Similarly to all previous experiments, this figure shows slip force to increase with idler pitch and material loading, and also reiterates that HDPE rollers are the worst roller type in terms of unidirectional idler performance.

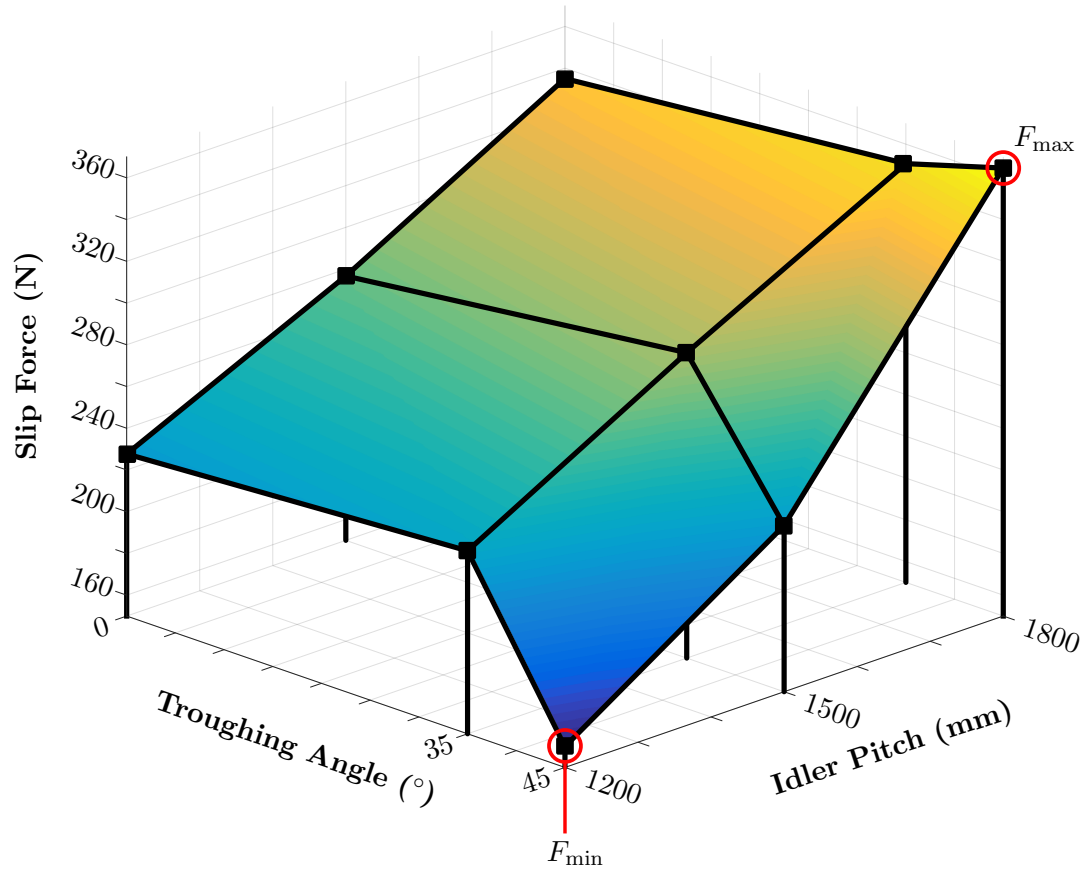


Figure 4.19: Three-dimensional surface of tension averaged slip forces (Programme E)

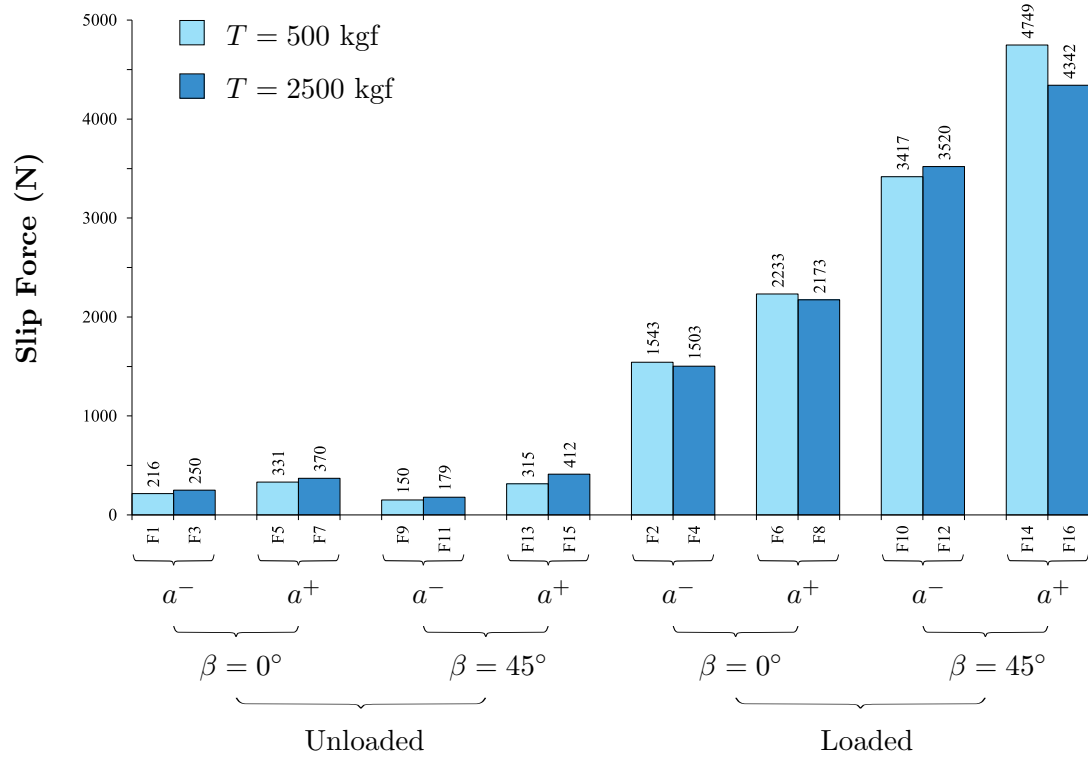


Figure 4.20: Slip forces grouped according to pitch, trough and load (Programme F)

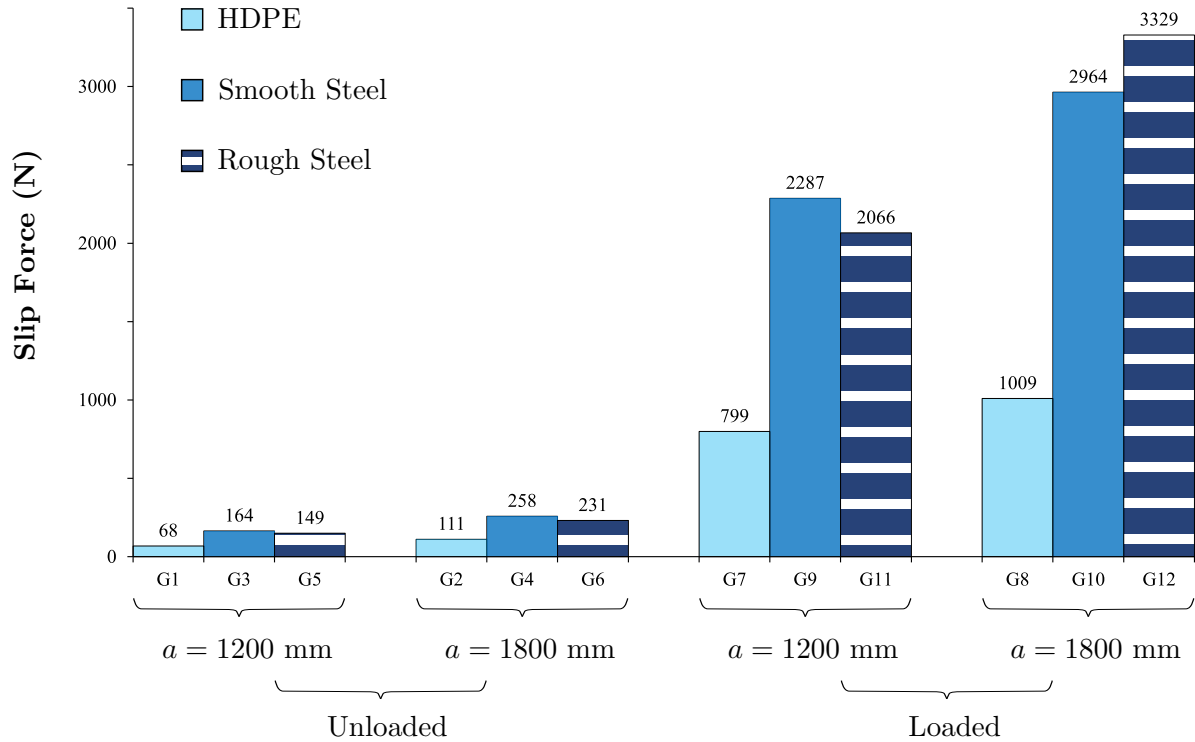


Figure 4.21: Slip forces grouped according to pitch and load (Programme G)

The next mini-investigation that was run, Experimental Programme H, was intended to test the effect of belt inclination on slip force for a loaded belt case. Although Programme C showed that inclination had no significant effect on belt slip, that study focused on empty belts and therefore didn't take any material load effects into account. As such, the experimental set-up for Programme H was exactly the same as that for Programme C^c (refer to Figure 4.12) except that in this case, the belt was loaded with sand bags. This was done in order to verify whether or not the material load introduced any additional downward pulling forces which could potentially reduce the critical slip force and thus deviate from the original results collected in Programme C.

The results of this study are presented by the transient resistive force plot and kinetic slip force bar chart in Figure 4.22. These figures show that, unlike the unloaded belt case previously presented, the critical slip force of a loaded conveyor belt is influenced by inclination angle. Both figures show that the resistive force at kinetic belt slip decreases steadily with increasing inclination angle, with approximately 10% reduction in slip force over the range of inclinations.

^cThe inclination angle of the test rig was varied, and all other belt conveying factors (including idler pitch, troughing angle and belt tension) were kept constant.

The last formal study that was run (Programme I) was designed to test the influence of inclination angle on the slip force of HDPE rollers. This investigation was similar to an amalgamation of Programmes C and H, except that in this case HDPE idlers were installed on the rig and a smaller idler pitch of 1200 mm was used. These factors were chosen as a worst case scenario. The results from this investigation show slip force to decrease with increasing inclination angle when the belt is empty (Figure 4.23a). However, when the belt is loaded, Figure 4.23b reveals inclination angle to have no meaningful effect on slip force.

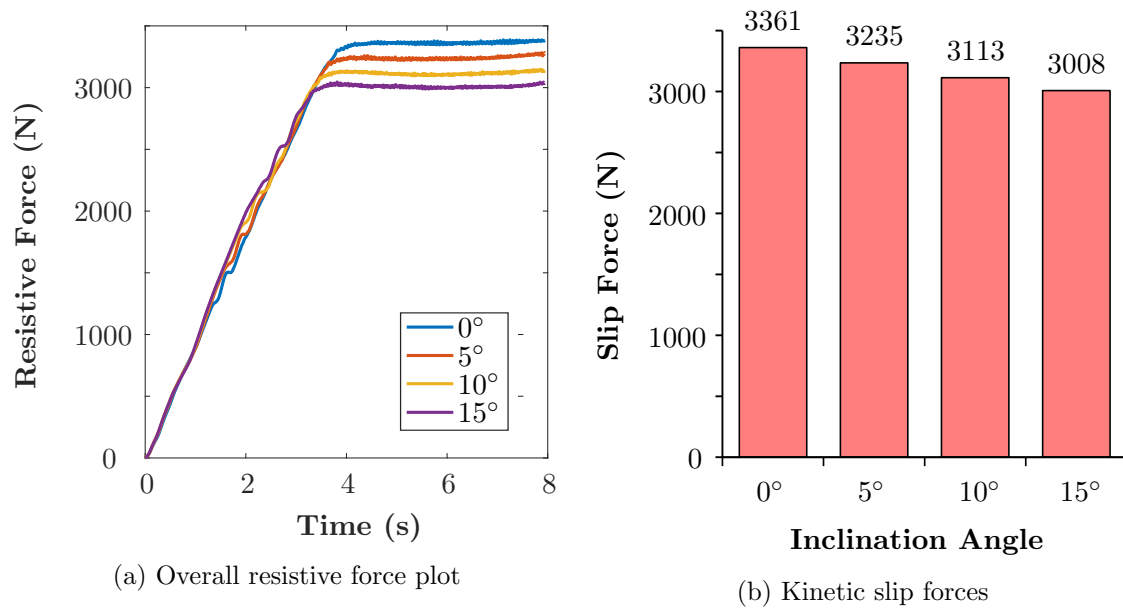


Figure 4.22: Resistance plots for a loaded belt inclined at varying inclination angles (Experimental Programme H)

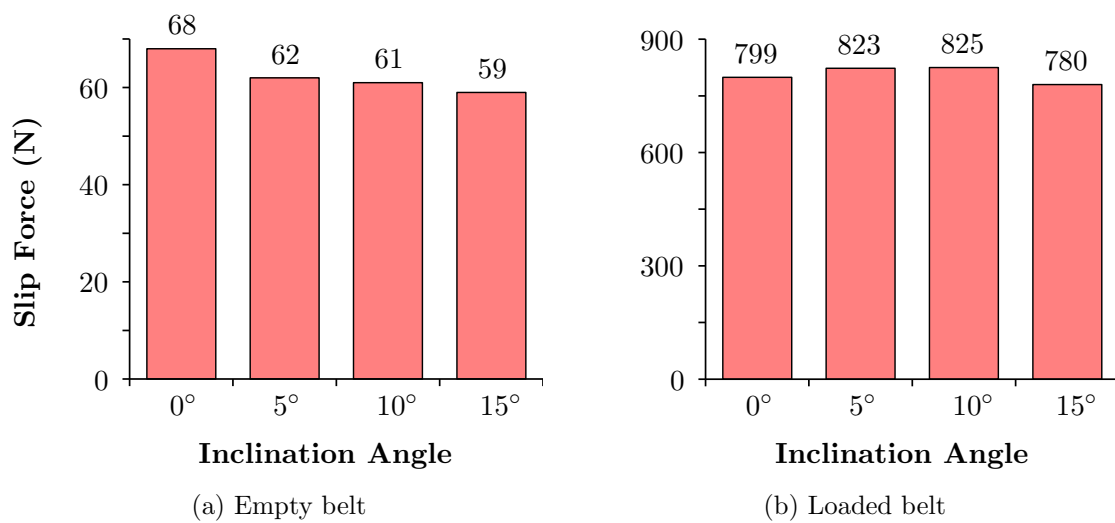


Figure 4.23: Kinetic slip forces for HDPE rollers at varying inclination angles (Experimental Programme I)

During the course of unidirectional roller experimentation, a number of additional experiments were conducted. These experiments did not form part of any formal testing programme but were run to supplement the experimental data when training the unidirectional idler quantity specifying numerical model. This mix of experiments consisted of tests focussed on the less comprehensively investigated HDPE and paintless steel rollers. The slip forces collected from these experiments are listed in Table 4.3. In this table, the experiments conducted with paintless steel rollers are denoted by the acronym NP, which stands for No Paint.

Table 4.3: Kinetic slip forces (Experimental Programme J)

TEST	Idler Pitch (mm)	Roller Type	Troughing Angle ($^{\circ}$)	Belt Tension (kgf)	Slip Force (N)
J1	1200	HDPE	0	500	82
J2	1200	Steel (NP)	0	500	177
J3	1200	HDPE	35	500	68
J4	1200	HDPE	35	1500	68
J5	1200	HDPE	35	2500	78
J6	1200	HDPE	45	500	106
J7	1200	Steel (NP)	45	500	128
J8	1800	HDPE	0	500	157
J9	1800	Steel (NP)	0	500	318
J10	1800	HDPE	35	500	111
J11	1800	HDPE	35	1500	123
J12	1800	HDPE	35	2500	126
J13	1800	HDPE	45	500	180
J14	1800	Steel (NP)	45	500	241

After all the experimental programmes had been conducted, further analysis could be realised with reference to the overall results. One such analysis that was considered was the effect that the cumulative belt and roller wear had on the measured slip forces. This was inspected due to the experimental observation that the test rig belt and roller surfaces became smoother after repeated use. Two graphs were generated to illustrate the effect of surface wear on the experiments conducted (Figures 4.24 and 4.25).

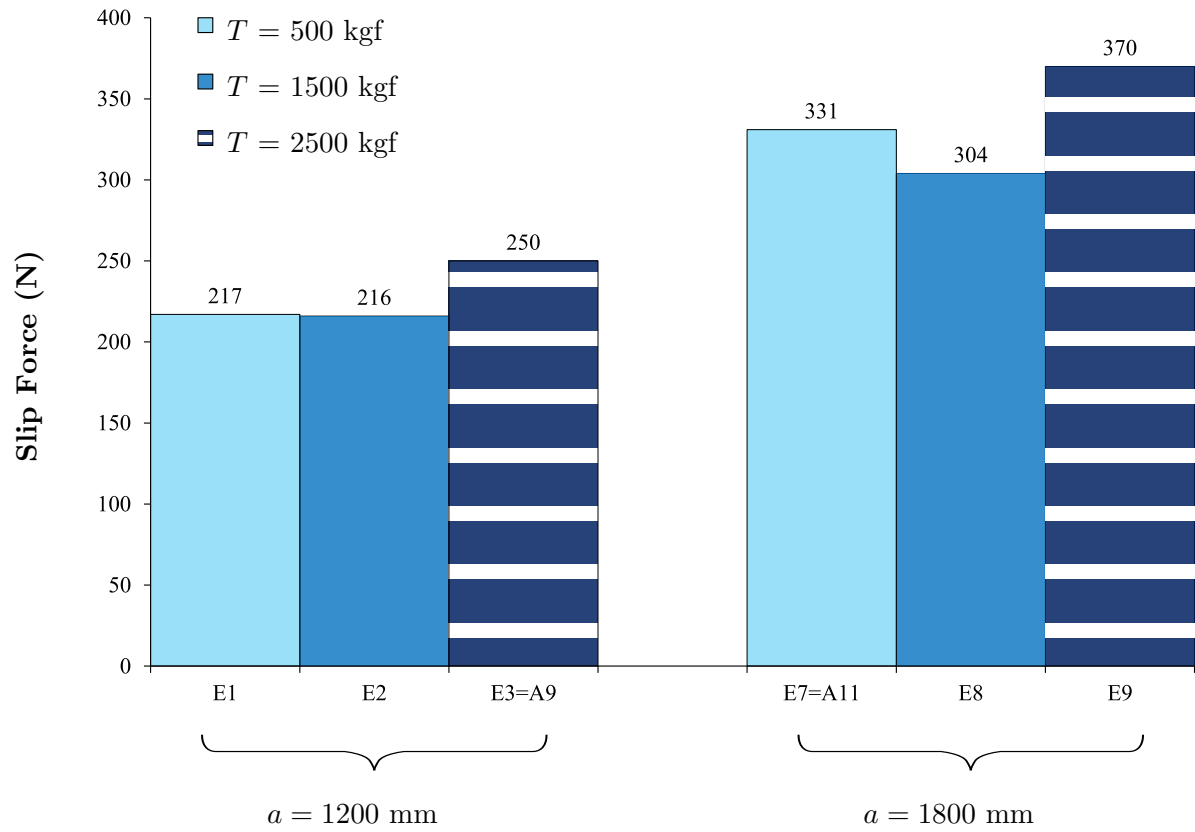


Figure 4.24: Flat belt slipping forces grouped according to idler pitch and belt tension

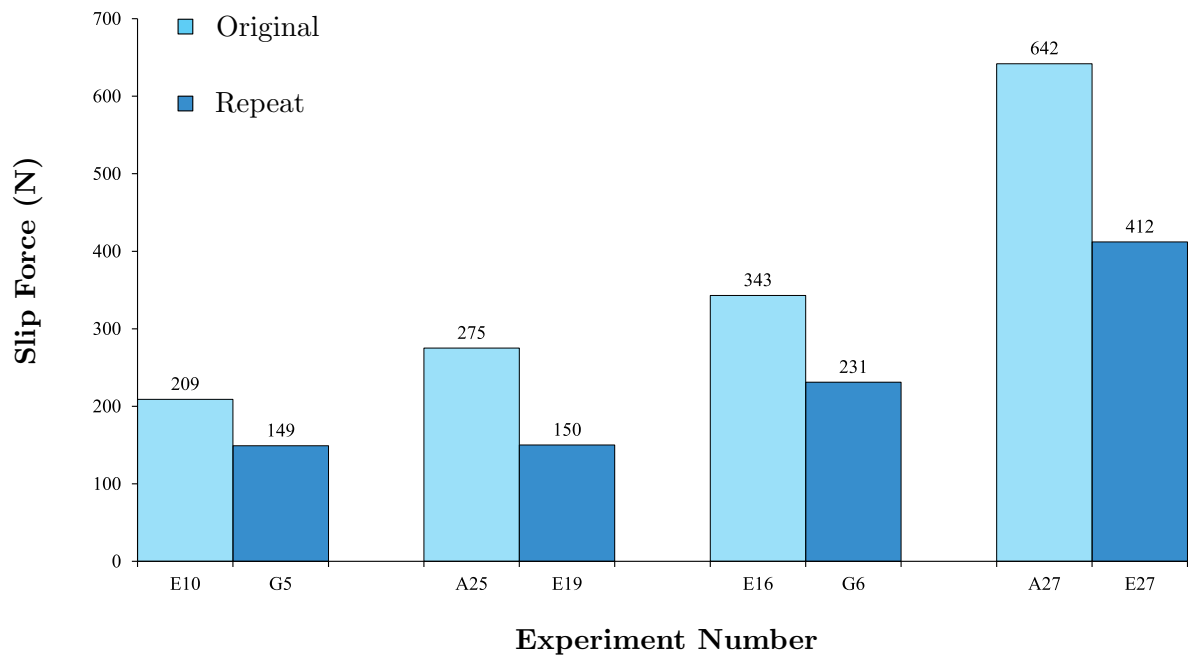


Figure 4.25: Kinetic slip force comparison of repeated experiments

The first of these graphs (Figure 4.24) shows the slip forces of experiments E3 and E7 (which had already been examined in Programme A and were then substituted into Programme E) to be uncharacteristically higher than expected. This indicates a change in the surface properties of the belt as it wears down from continuous use. This apparent change in belt slip force during the life span of a conveyor belt is even more evident in the results presented in Figure 4.25. This graph compares the slip forces of repeated experiments which may be seen to be constantly (and considerably) lower than their original duplicates (characterised by a 36% reduction in slip force on average). Together these figures demonstrate the deterioration of unidirectional idler performance over time that results from the continuous wear of a conveyor belt in operation.

Before the next section of results is considered (those pertaining to the additional friction study) it is worth summarising the results of the focused unidirectional roller experiments.

The corresponding results of this investigation were obtained through the execution of eight experimental programmes (B to I) using the UDR test rig. These experiments were designed based on the results of the initial factor screening experimental programme, and were conducted in order to develop a better understanding of how the various factors affect unidirectional idler performance. A summary of the supplementary experiments conducted may be seen in the flow chart presented in Figure 4.26.

In addition to the development of a better understanding of unidirectional idlers, the experiments presented in Figure 4.26 also allowed for a larger data set to be compiled and used to develop a unidirectional idler quantity specification model (refer to Chapter 5). The first of the supplementary experiments conducted, Programme B, was designed to gain further insights into the effect of roller material. The results from this programme showed that HDPE rollers exhibit a significantly lower resistance to belt slip when compared to the previously studied Nylon and steel rollers. This indicated a new worst case to consider in terms of friction contributions to overall unidirectional idler resistive force.

The next study conducted, Programme C, confirmed that the effect of inclination angle on measured slip force was insignificant in the case of an empty belt. However, programme H illustrated that this was not the case for a loaded belt. When there is material load on the belt, the added downward pulling force due to gravity works in favour of the test rig belt pulling action and the resistive force is therefore seen to decrease with increasing inclination angle. These changes in measured slip forces, however, aren't substantial, and actually weren't observed in the loaded cases of experimental programme I (which was mostly conducted in order to collect worst case inclined belt data to be used in the numeric simulation).

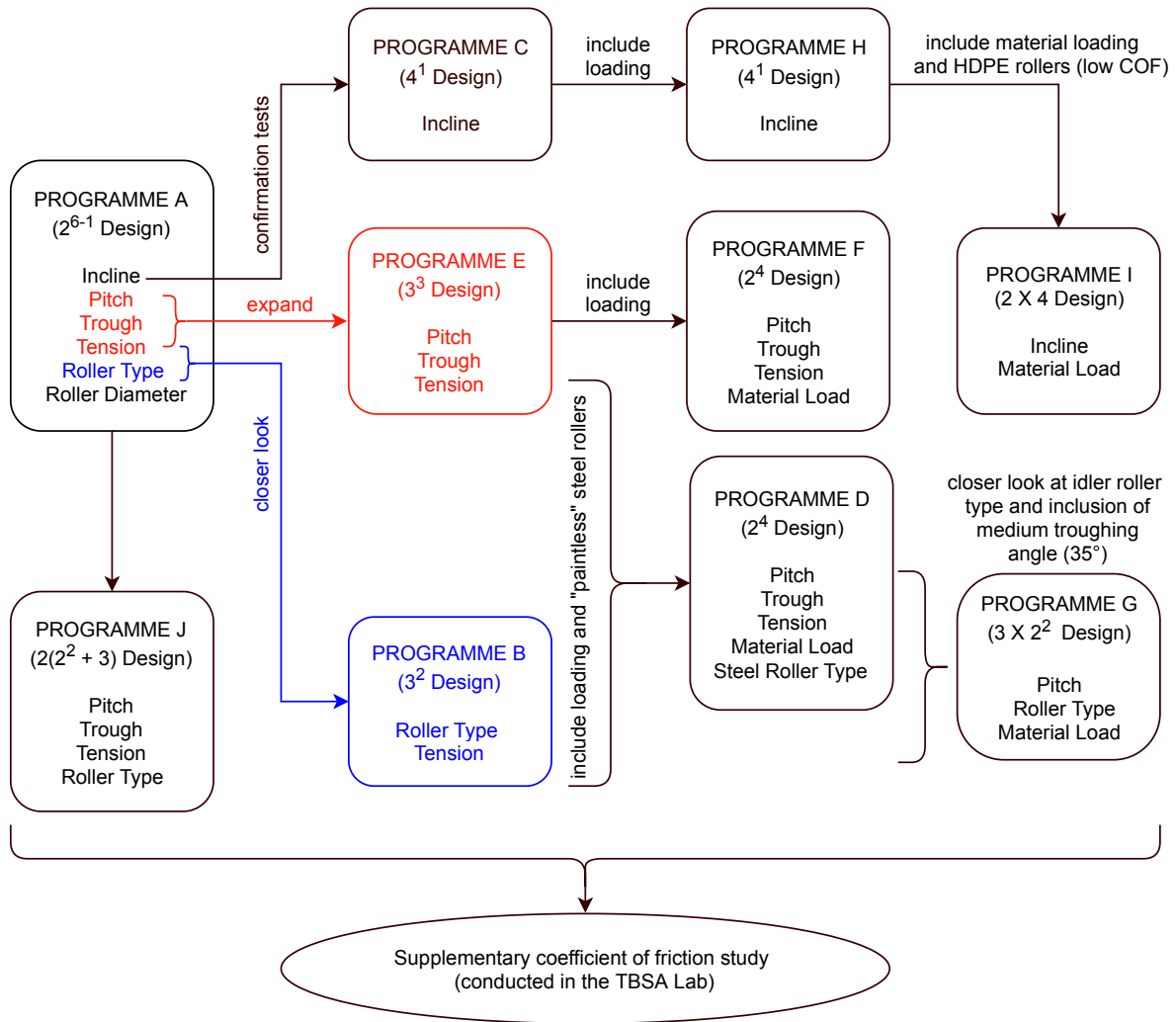


Figure 4.26: Schematic diagram of the experimental programmes conducted

Experimental Programme D was conducted in order to include loaded belt cases and polished steel rollers into the investigation. This study showed that although polished steel rollers exhibited lower slip forces on average, the difference between painted and polished steel was not significant - particularly when loaded belts were considered. Other key results of this study were determined from a half-normal significance analysis which revealed material load to have the most significant influence on belt slip, followed by the factors of troughing angle and idler pitch. All three of these factors have a positive effect on belt slip (i.e. an increase in the factor results in an increase in belt slip). The most significant interaction-pairs identified in the results were the load-trough and load-pitch interactions. In these interactions, material load was seen to have a greater impact on unidirectional idler performance when the belt troughing angle and idler pitch were larger.

A deeper look into the key findings of the factor screening investigation was fulfilled through Experimental Programme E. This study was used to develop an average empty belt kinetic slip force profile based on the most significant belt conveying factors (as identified by Programme A). This investigation indicated a worst case scenario (i.e. lowest slip force) to occur when the idler pitch was 1200 mm and the troughing angle was 45° .

The results of Programmes F and G reiterated the findings of previous experimentation – that slip force increased with increasing idler pitch, material load and troughing angle, and that HDPE rollers resulted in noticeably lower slip forces when compared to other roller materials.

Additionally, the analysis of substituted and repeated experiments showed that belt wear affected the critical slip force of a conveyor belt sliding over locked unidirectional idlers. This analysis showed that an increase in belt wear resulted in a lower observed resistive force at belt slip (i.e. belt wear is detrimental to unidirectional idler performance).

All in all, these results, together with those obtained from the additional friction study (Section [4.2](#)), informed the numerical simulation to be developed in Section [5.3](#).

4.2 Additional Friction Study Results

In order to develop a better understanding of the friction developed between a conveyor belt and an idler, a supplementary friction study was conducted. Refer to Section 3.4 and Table A.11 for the description of the test apparatus and procedure, and for the variables in this study^d.

Following the test procedure in Section 3.4.1, an example of the data collected in a typical friction experiment is presented in Figure 4.27. The steps in the red contact pressure line correspond to different normal loads applied to test samples (in the sequence 20, 19, 18, 17, 16, 14, 12, 10, 8, 6, 4, 2 kg applied load), while the frictional resistance shear stress is shown in the blue line, where small variations are seen due to stick-slip behaviour of the roller material block moving along the belt sample surface.

It may also be noted from Figure 4.27 that the first shear stress measurements recorded when the mass hanger was fully loaded (from 0 to 500 data point numbers) was not constant, but rather displayed a negative exponential trend. The reason for this is likely due to the fact that while loading the mass hanger, before the experiment starts, the roller material sample is being indented into the surface of the rubber belt. Upon start up, the sample has to "climb out" of this induced indentation, which requires considerable force at first, but less and less as it is pushed out of this indentation area.

From this data, shear loads were averaged across each contact pressure state (see red markers) and coefficients of friction (COF) were calculated using Equation 3.11. These coefficients of friction were then averaged across all the corresponding test run repeats and a linear regression model fitted.

The first conveyor belt sample to be investigated with the JDST was cut from the same conveyor belt installed on the UDR Test Rig: a Nitrile rubber covered three-ply fabric belt. The first two roller materials tested with this belt sample were rough steel and Nylon – the same two roller materials tested during the factor screening experimental programme. The results of this initial friction comparison are presented in Figure 4.28.

^dIt should be noted that the effects of idler roller diameter on COF were not considered because the effects of roller diameter on UDR performance was already discounted by the factor screening experiments, and because COF is velocity dependant – not area dependant

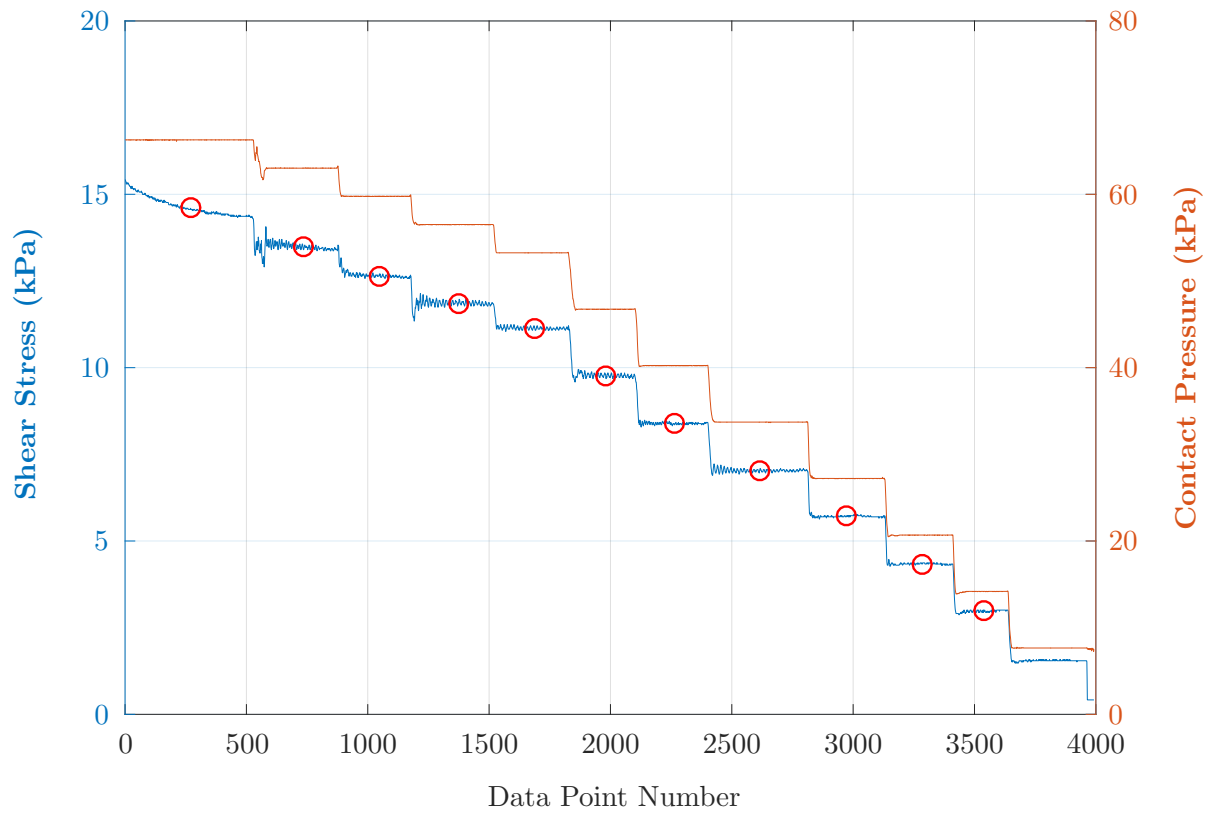


Figure 4.27: Typical test run kinetic friction vs contact pressure graph

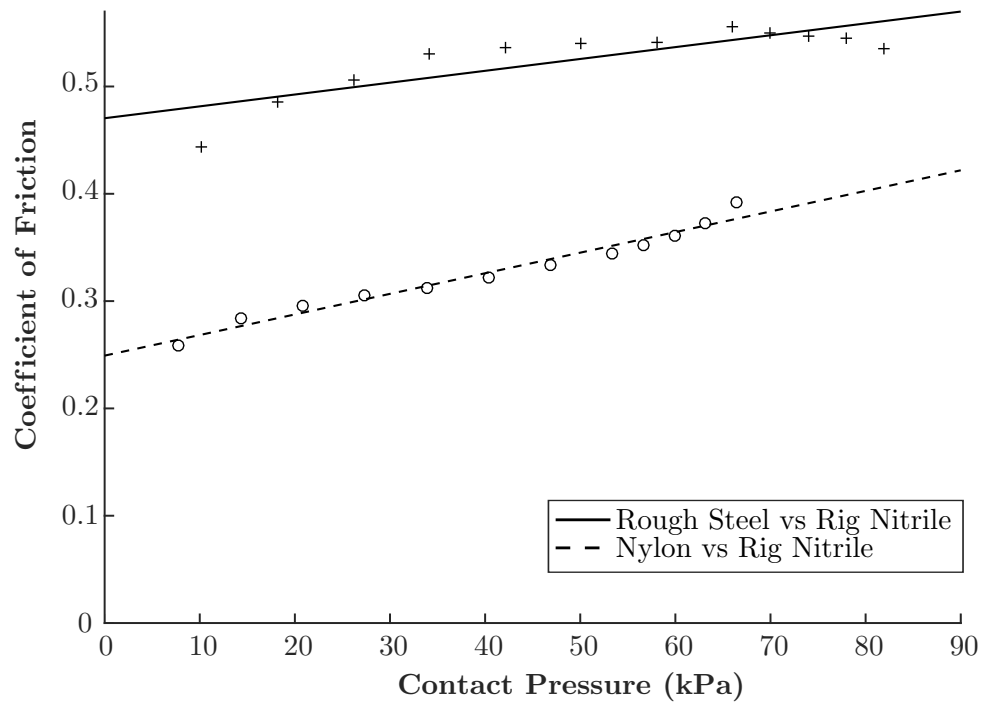


Figure 4.28: Steel vs. Nylon coefficients of friction for the test rig Nitrile rubber belt

This figure reveals there to be a notable difference in the coefficients of friction for the rough steel and Nylon material samples (with the former exhibiting the higher COF as expected). It is also interesting to note the pressure dependence of the friction coefficient exhibited by both roller materials being pushed along the belt sample. In both cases, this pressure dependence is seen to exhibit a positive relationship with contact pressure – that is to say that an increase in contact pressure results in an increase in COF.

At the time of conducting the initial factor screening programme, HDPE rollers (which demonstrate a lower COF than Nylon) had not yet been procured. As a result, Nylon rollers were used for the low COF level in their place. By the time Experimental Programme B was executed, however, HDPE rollers had been obtained. Since the focus from that point on was HDPE and steel rollers, the rest of the additional friction study experiments also focused on these roller materials. The results of these sample tests are summarised in Figure 4.29. This figure shows the variation in COF according to the roller material (smooth/rough steel or HDPE) and the preparation of the rig Nitrile belt sample (clean, dirty or wet)^e.

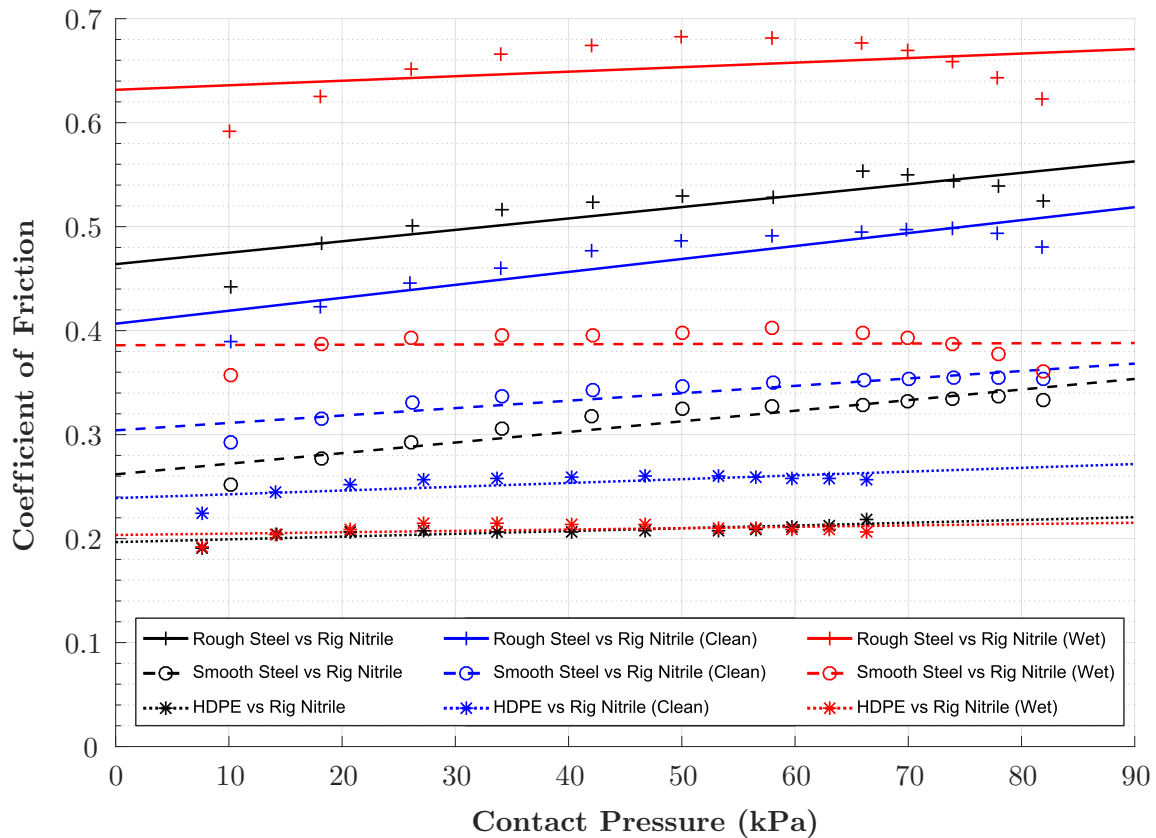


Figure 4.29: Mean coefficients of friction for the test rig Nitrile rubber belt sample

^eThe “dirty” state of the belt sample was considered to be that when the sample was obtained, which just so happened to be dirty and dusty. Experiments were conducted on this “dirty” belt first. Then the belt was cleaned and all experiments run again. Finally, the clean belt was used to run wet tests.

From this graph it may be seen that the rough steel sample demonstrated the highest average COF, with the smooth steel sample coming in second and the HDPE sample exhibiting the lowest COF. The order of these COF trends, which are also observed to exhibit pressure dependence, was expected. It is also worth noting that although non-linear behaviour was exhibited in some cases, only linear curve fits were conducted because the worst cases for belt slip tended closer to linear functions.

The next belt tested was a Nitrile rubber covered, solid woven PVC sample that was taken from a belt previously in operation. One of the reasons for testing a used belt was to determine whether or not the surface properties change significantly enough during continuous operation to result in a change in the associated friction coefficient. The COF data collected for this belt sample is presented in Figure 4.30. Similarly to previous friction graphs, this figure shows rough steel to have the highest average COF, followed by smooth steel and HDPE. This figure was also the first to demonstrate a negative COF pressure dependence (observed when rough and smooth steel samples were wet). Since the intention of the additional friction study was to determine a worst case friction coefficient for each of the relevant material pairs, this anomaly in pressure dependence gradient was deemed to be of no consequence and was therefore not investigated any further.

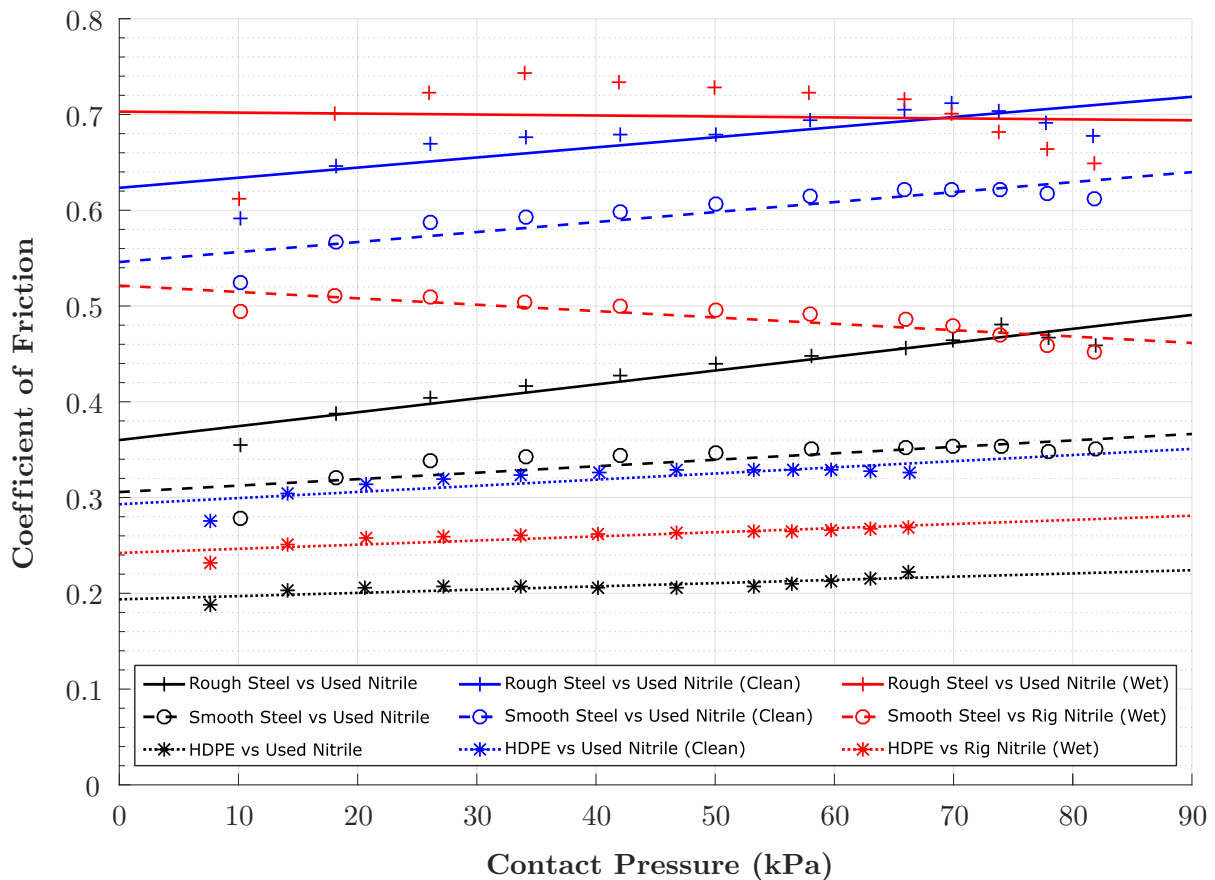


Figure 4.30: Mean coefficients of friction for the used Nitrile rubber belt sample

Following on from this, two more belt samples were studied with the JDST: an unused Nitrile rubber covered solid woven PVC belt and a Type M rubber covered steel cord belt. Unlike the first two belt samples investigated, there was no variation in the surface condition of the new Nitrile or Type M rubber belt samples. The COF data for these belts is presented in Figures 4.31 and 4.32 respectively.

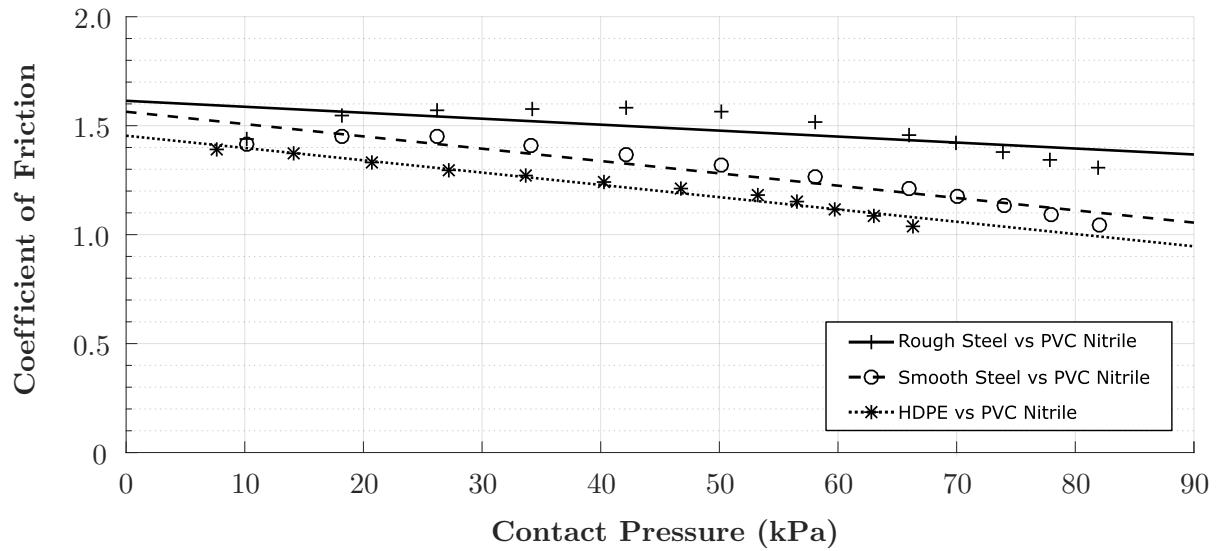


Figure 4.31: Mean coefficients of friction for the PVC Nitrile rubber belt sample

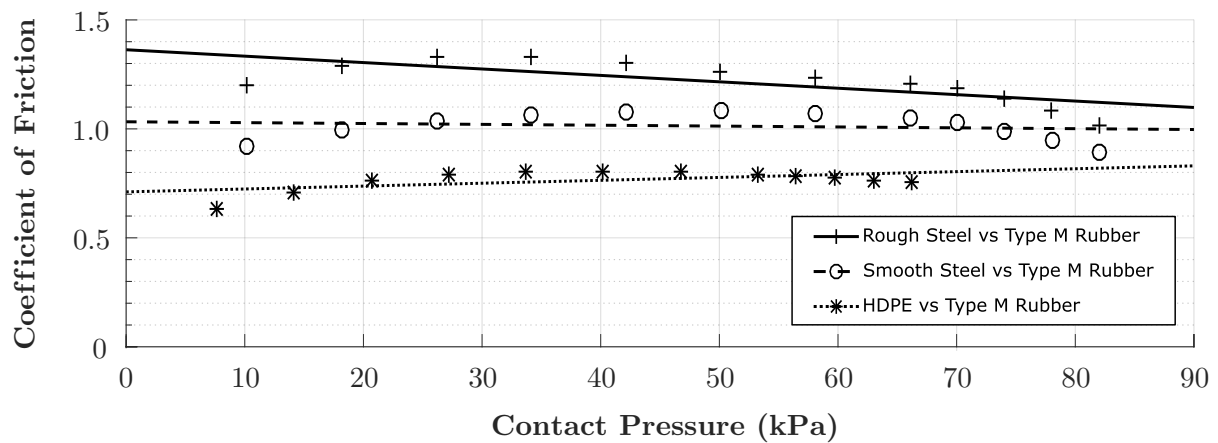


Figure 4.32: Mean coefficients of friction for the Type M rubber belt sample

These figures reaffirm the order of COF associated with the three roller material samples. However, in both cases, the COF magnitude is observed to be greater than that in any previous experiments. Additionally, the direction of the observed COF pressure dependence is also seen to contrast with that from earlier tests, especially in the case of the unused PVC Nitrile rubber sample which exhibits the highest overall COF of all belt samples considered.

From the four tested conveyor belt constructions, the worst cases for each belt and roller material sample pair (for the samples with varied belt preparation) were identified and plotted together in a single graph to summarize and compare friction trends. It is important to reiterate that in the context of this research, a worst case in terms of friction is when the COF is low. The lower the COF, the poorer the friction braking capability of a specific roller material construction. With this in mind, the overall minimum coefficient of friction graph is presented in Figure 4.33.

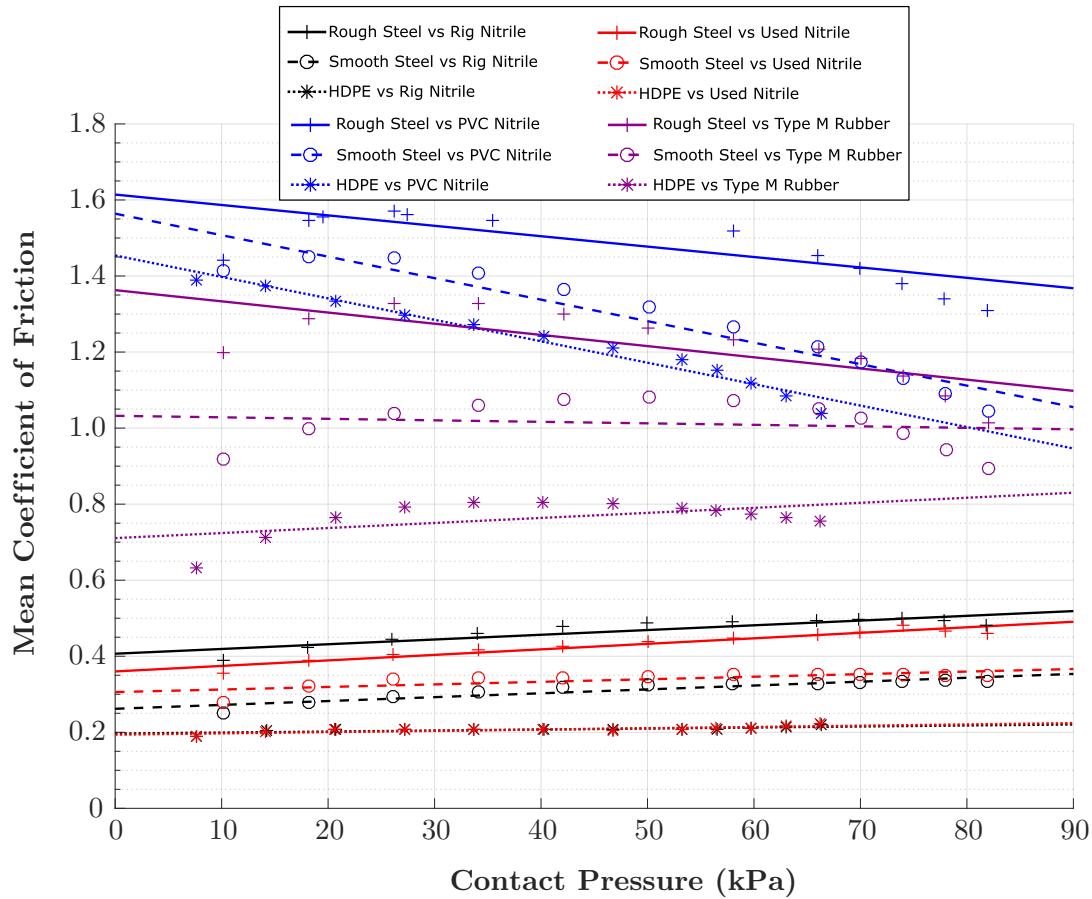


Figure 4.33: Minimum coefficients of friction for various material pairs

This figure summarises and compares some of the findings previously presented in this section. From this graph it may be seen that the COFs experienced when testing the rig Nitrile and used Nitrile belt samples were very similar, with HDPE showing almost no difference at all between these two belt samples. Although the COF magnitude varies slightly in the case of the two steel roller material samples, the corresponding trends observed with contact pressure were very similar in both the rig and used Nitrile belt cases (note the parallelism between the corresponding COF trend lines).

Finally, two more COF graphs were plotted to compare the results of specific conveyor belt samples to each other. The first comparison made was that between the Nitrile rubber belt installed on the UDR Test Rig and a used PVC Nitrile rubber belt. This comparison was made to show the high degree of similarity between these two belt samples. The second comparison considered was between the new PVC Nitrile belt and the used PVC Nitrile belt. This comparison between a new and used conveyor belt of the same construction sought to highlight the effects that belt surface wear over time has on the friction coefficient. These belt sample comparison graphs are shown in Figures 4.34 and 4.35.

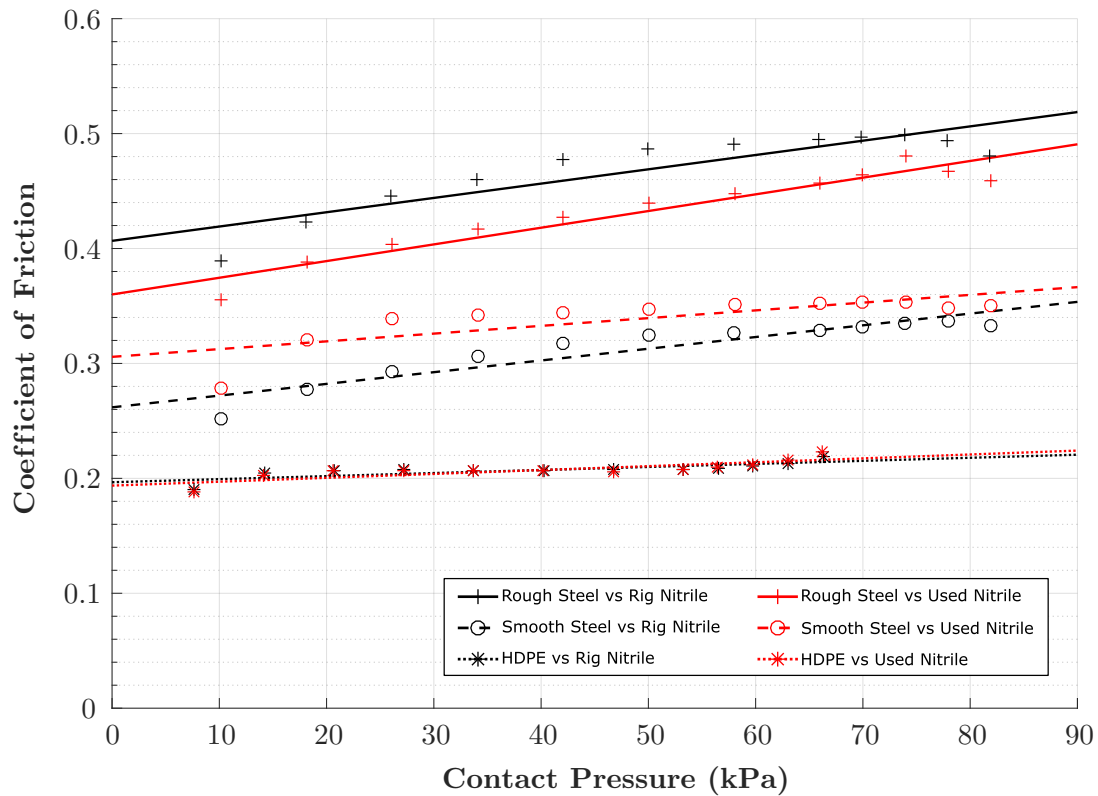


Figure 4.34: Comparison between test rig and used PVC Nitrile rubber friction coefficients

With these findings, the resistive force measurements conducted using the UDR Test Rig could be broken up into friction and non-friction force components. This decoupling allowed the empirically derived resistive force trends observed with the UDR Test Rig to be extrapolated to the generic case. In addition to this, the observed friction coefficients could be applied, using a numerical model derived from experimental results (see Chapter 5), in the unidirectional idler quantity calculation. This enables the collected resistive force data to be used in the design of unidirectional systems with different conveyor belt or roller types to those tested on the UDR Test Rig.

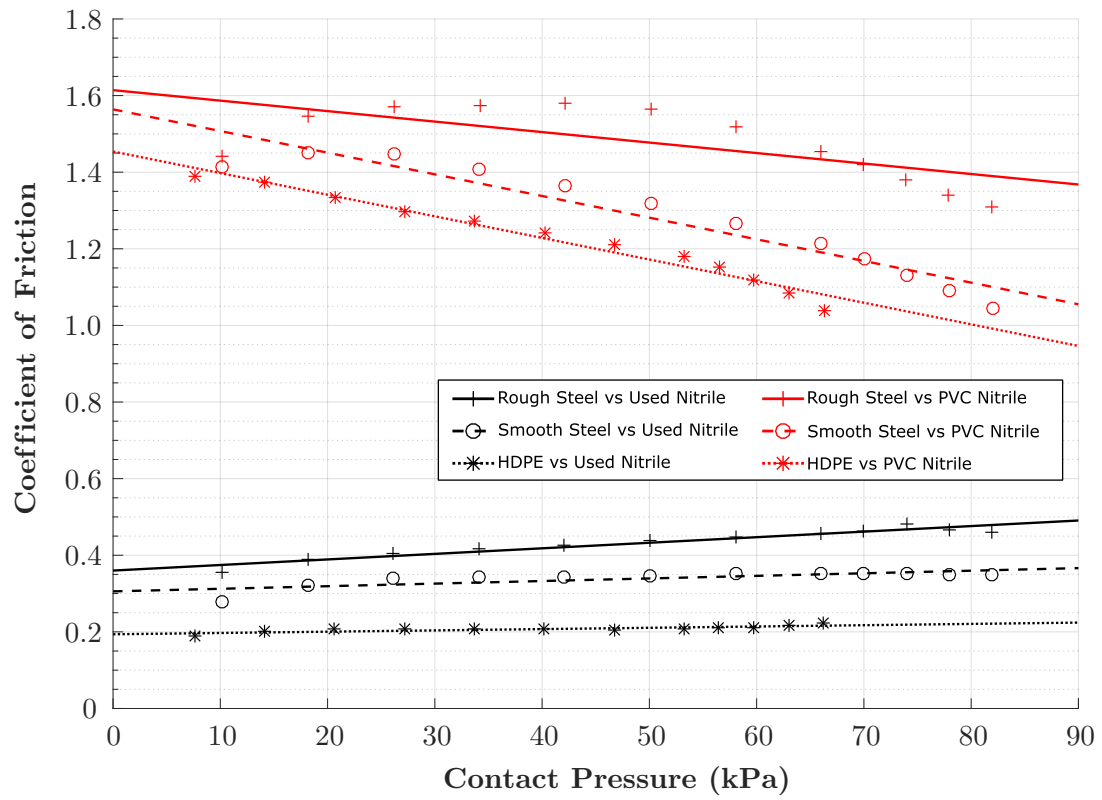


Figure 4.35: Comparison between new and used PVC Nitrile rubber friction coefficients

Chapter 5 Numerical Model for Specifying the Proportion of Unidirectional Idlers

5.1 Introduction

To supplement the results of the unidirectional roller and friction coefficient investigations, a numerical model was derived to specify the required proportion of unidirectional rollers for inclined belt applications. This model took the specific operating conditions of inclined conveyor belts into account and calculated the number of rollers required to be unidirectional in order to prevent a runaway belt from accelerating down the incline. The contents of this chapter explain the steps followed to derive this numerical model.

5.2 Proportion Formula Derivation

Since it was empirically confirmed that the force resisting belt motion over locked idlers was not entirely due to friction (refer to Section 6.3), the conventional friction formula ($F_f = \mu N$) couldn't be used to determine the resistive force acting at the belt-idler interface. However, using a modified version of this formula, the overall resistive force could be related to the roller normal force. This adapted formula^a is given by:

$$F_{res} = \psi N = \lambda \mu N \quad (5.1)$$

where F_{res} is the resistive force and ψ is the resistance coefficient given by the product of the non-friction coefficient (λ) and the friction coefficient μ , such that $\psi = \lambda \mu$.

^aThe conventional friction formula ($F_f = \mu N$) was adapted (see Equation 5.1) by replacing the coefficient of friction term (μ) with a newly defined resistance coefficient (ψ). This constant can be broken up into friction and non-friction coefficients (μ and λ respectively), which contribute to the overall slip resistance at the belt-idler interface. While it is possible to use a resistance term that sums the friction and non-friction effects, the approach chosen here (arbitrarily so) was multiplicative

Knowing the relationship between the friction and non-friction factors and using the minimum recorded friction coefficients as measured with the JDST, the non-friction coefficient (λ) for each experiment conducted on the UDR Test Rig was calculated using:

$$\lambda = \frac{\psi}{\mu} \quad (5.2)$$

The use of these empirically determined non-friction factors (λ) together with the measured coefficient of friction values, allowed for the resistive force developed at a unidirectional idler set to be found, and for a table of non-friction factors at different operating conditions to be drawn up. In order to extend the empirical findings to the theoretical case, it was important to derive the equations for normal force (N) acting at the unidirectional rollers. Before these expressions could be determined, however, it was important to derive the equation of the curve that the material load forms on the belt.

5.2.1 Bulk Material Profile and Cross-Sectional Area

The derivation of the profile that the bulk material forms on the belt, based on the diagram shown in Figure 5.1, is presented below.

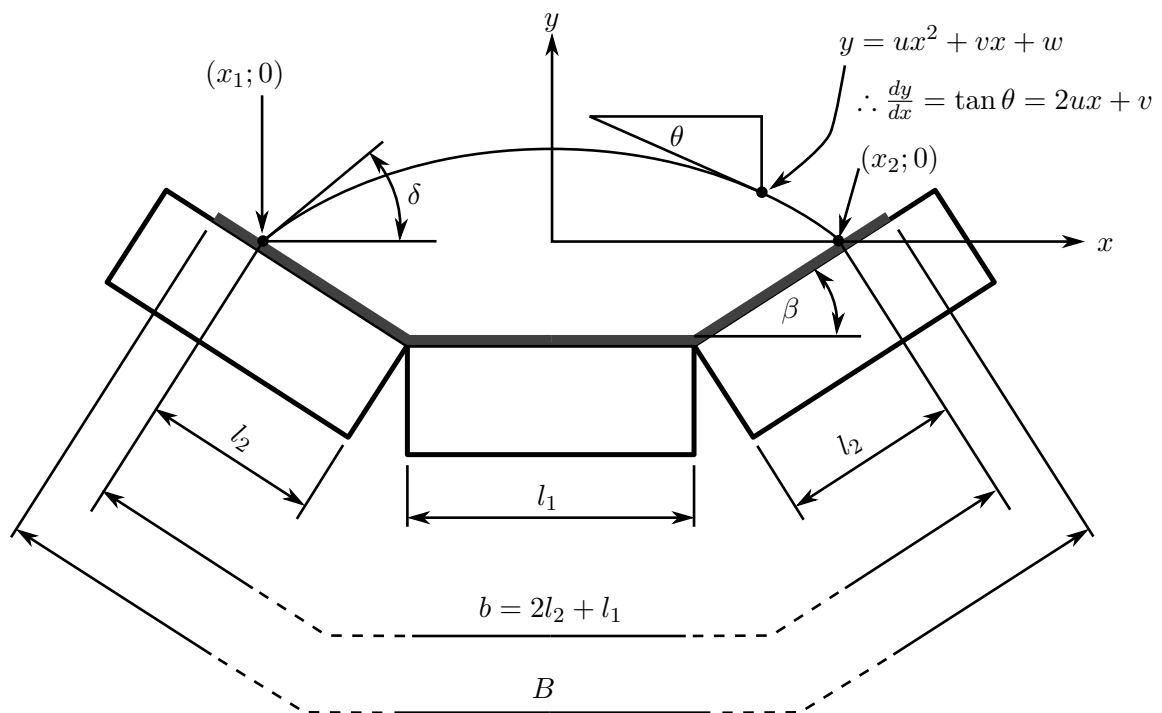


Figure 5.1: Overall bulk material load profile diagram

Using the geometric convention of the x - y axes as shown in Figure 5.1, and assuming the material to follow a parabolic profile [8], the general form of the equation may be stated as:

$$y = ux^2 + vx + w \quad (5.3)$$

Knowing the gradient of the tangent to this curve to be equal to $\tan \theta$, it may be stated for any location x that:

$$\tan \theta = \frac{dy}{dx} = 2ux + v \quad (5.4)$$

From the diagram shown in Figure 5.1:

$$b = 2l_2 + l_1 \quad \implies \quad l_2 = \frac{1}{2}(b - l_1)$$

where b is the usable width of the belt (in metres – [m]) given by:

$$b = 0.9B - 0.05 \quad (5.5)$$

where B is the belt width in metres – [m].

From the diagram:

$$\begin{aligned} x_1 &= -\frac{1}{2}l_1 - l_2 \cos \beta \\ \therefore x_1 &= -\frac{1}{2} \left[l_1 + (b - l_1) \cos \beta \right]; \quad x_1 = -x_2 \end{aligned}$$

Due to the recurring nature of the $\left[l_1 + (b - l_1) \cos \beta \right]$ term, this may be replaced in the calculations that follow with the symbol C , such that:

$$C = l_1 + (b - l_1) \cos \beta$$

With this in mind, the coordinate x_1 may be restated as:

$$x_1 = -\frac{1}{2}C; \quad x_1 = -x_2$$

When $x = x_1$; $\theta = \delta$

$$\therefore \tan \delta = -uC + v \quad \implies \quad v = \tan \delta + uC \quad (5.6)$$

When $x = x_1$; $y = 0$

$$\therefore 0 = \frac{u}{4}C^2 - \frac{v}{2}C + w \quad (5.7)$$

When $x = 0$; $\frac{dy}{dx} = 0$

$$\therefore 0 = 2u(0) + v \quad \implies \quad v = 0 \quad (5.8)$$

Substituting Equation 5.8 into 5.6:

$$0 = \tan \delta + uC \quad \Rightarrow \quad u = -\frac{\tan \delta}{C} \quad (5.9)$$

Substituting Equations 5.8 and 5.9 into 5.7:

$$0 = -\frac{1}{4} \tan \delta C + w \quad \Rightarrow \quad w = \frac{1}{4} \tan \delta C \quad (5.10)$$

The formula of the parabola that the material load exhibits may thus be stated as:

$$y = -\frac{\tan \delta}{C} x^2 + \frac{1}{4} \tan \delta C \quad (5.11)$$

where $C = l_1 + (b - l_1) \cos \beta$.

With the equation of this profile known, the cross-sectional area of carried material load directly above the centre roller was determined. This area is important since the magnitude of the normal force acting at the centre roller is approximately equal to the distributed weight of the material and belt directly above the centre roller [13]. The centre roller area equation was derived according to the diagram shown in Figure 5.2.

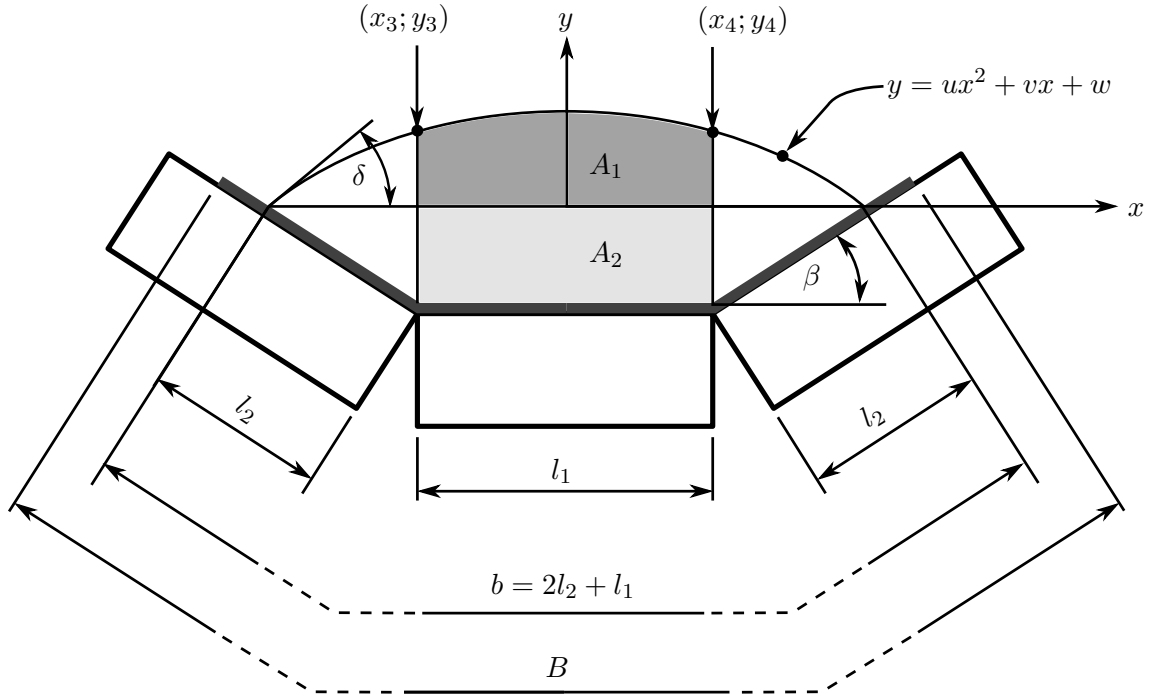


Figure 5.2: Bulk material load profile diagram used to calculate the cross-sectional area of material directly above the centre roller

From this diagram:

$$x_4 = \frac{1}{2}l_1; \quad x_3 = -x_4$$

Using Equation 5.11, the area of section A_1 may be calculated as follows:

$$\begin{aligned} A_1 &= 2 \int_0^{\frac{1}{2}l_1+1} \left(-\frac{\tan \delta}{C}x^2 + \frac{1}{4}\tan \delta C \right) dx \\ &= 2 \left(-\frac{x^3 \tan \delta}{3C} + \frac{x}{4}\tan \delta C \right) \Big|_0^{\frac{1}{2}l_1} \\ &= -\frac{l_1^3 \tan \delta}{12C} + \frac{l_1 \tan \delta C}{4} \\ \therefore A_1 &= \frac{\tan \delta (3l_1 C^2 - l_1^3)}{12C} \end{aligned} \tag{5.12}$$

Using the formula for the area of a rectangle, it was calculated that $A_2 = \frac{1}{2}l_1(b - l_1) \sin \beta$. Adding this to the parabolic area A_1 , the total bulk material cross-sectional area above the centre roller may be written as:

$$A_c = \frac{\tan \delta (3l_1 C^2 - l_1^3)}{12C} + \frac{l_1(b - l_1) \sin \beta}{2} \tag{5.13}$$

Following this, the area above the wing rollers (A_w) was derived in a similar fashion. This equation is given by:

$$A_w = \frac{\tan \delta (2C^3 + l_1^3 - 3l_1 C^2)}{12C} + \frac{(b - l_1)^2 \sin \beta \cos \beta}{4} \tag{5.14}$$

Finally, the total cross-sectional area of conveyed material (A) was determined by adding the areas above the centre and wing rollers and substituting $C = l_1 + (b - l_1) \cos \beta$ into Equations 5.13 and 5.14. The resulting equation is given by:

$$A = \frac{1}{6} \tan \delta \left[l_1 + (b - l_1) \cos \beta \right]^2 + \frac{1}{4} (b - l_1) \sin \beta \left[2l_1 + (b - l_1) \cos \beta \right] \tag{5.15}$$

5.2.2 Idler Roller Normal Force Equations

With the individual load areas determined, the resulting normal forces could be obtained. Approximating the overall normal load of an idler system to be equal to the weight of the belt and the conveyed material pushing down on the idler frame. This force may be calculated using the following equation:

$$N = a(q_b + q_m)g = a(q_b + \rho A)g \quad (5.16)$$

where a is the idler pitch, q_b is the mass per metre of the conveyor belt, ρ is the bulk density of the conveyed material and g is the acceleration due to gravity.

By substituting Equation 5.15 into 5.16, the expected normal force from a full material load may be calculated using:

$$N = ag \left\{ q_b + \rho \left[\frac{1}{6} \tan \delta \left(l_1 + (b - l_1) \cos \beta \right)^2 + \frac{1}{4} (b - l_1) \sin \beta \left(2l_1 + (b - l_1) \cos \beta \right) \right] \right\} \quad (5.17)$$

In the case of the centre roller, the normal load is approximately equal to the weight of the conveyor belt and load immediately above the centre roller (see Figure 5.2) and may thus be written as:

$$N_c = a \left(q_b \left(\frac{l_1}{B} \right) + \rho A_c \right) g \quad (5.18)$$

where l_1/B is the fraction of the belt width directly above the centre roller.

In the case of a troughed conveyor belt, however, previous research [13] showed that the normal load on a wing roller is greater than gravity acting alone. That is to say that the normal force acting at the wing roller is greater than the combined weight of the conveyed material and belt directly above the roller. Previous research showed that the wing roller normal force is 1.2 and 1.9 times greater than the associated combined weight for a 30° and 45° troughed idler roller respectively [13]. Assuming a linear relationship between these normal coefficients (ω), the loading condition for a 35° troughed wing roller was approximated to be 1.43 times greater than gravity. Using these coefficients, the normal force at the two wing rollers can be calculated according to:

$$N_w = a\omega \left(q_b \left(1 - \frac{l_1}{B} \right) + \rho A_w \right) g \quad (5.19)$$

where:

$$\omega = \begin{cases} 1.00, & \text{if } \beta = 0^\circ \\ 1.43, & \text{if } \beta = 35^\circ \\ 1.90, & \text{if } \beta = 45^\circ \end{cases} \quad (5.20)$$

With these normal force equations determined, a calculation for determining the required proportion of unidirectional idlers could be generated. In order to specify this proportion, it was important to consider the mechanics of the system.

5.2.3 Inclined Conveyor Belt Mechanical Model

Based on the preliminary mechanical model presented in the Literature Review, a more in-depth analysis was conducted. The unidirectional idler proportion equation derived in Section 3.2 assumed friction to be the only resistive force at play. Knowing this not to be the case, the proportion formula (Equation 3.8) could be re-written (according to Equation 5.1) as:

$$\eta = \frac{1}{\psi} \tan \alpha \quad (5.21)$$

where η is the proportion of rollers required to be unidirectional, ψ is the newly defined resistance coefficient and α is the angle of inclination.

Although the above equation considers the effects of non-friction factors on the braking performance of unidirectional rollers, it still does not take the differences between wing and centre roller resistive forces into account. Instead, it approximates the resistance of a three-roll unidirectional idler set as a single locked roller (see Figure 3.2). In order to address this issue, a more detailed mechanical model of the system was considered and two unidirectional idler proportion models were derived (see Sections 5.2.3.1 and 5.2.3.2).

5.2.3.1 Model 1 – Proportion Calculating Method based on the Different Resistive Properties of Centre and Wing Rollers

Figure 5.3 illustrates the forces acting at the centre and wing rollers of a length of belt at a specific angle of inclination in the limiting case where belt slip is about to occur. From this free body diagram, and based on previous derivations (see Equations 3.4 and 3.6), the force balance in the longitudinal direction may be stated as:

$$\sum F_z = 0 \quad \implies \quad \eta \frac{L}{a} (\psi_c N_c + \psi_w N_w) = (q_b + q_m) g L \sin \alpha \quad (5.22)$$

where:

$$\psi_c N_c = F_c$$

$$\psi_w N_w = \psi_w (N_{w,l} + N_{w,r}) = F_{w,l} + F_{w,r} = F_w$$

From Equation 3.7:

$$(q_b + q_m) g L = \frac{L N}{a \cos \alpha} \quad (5.23)$$

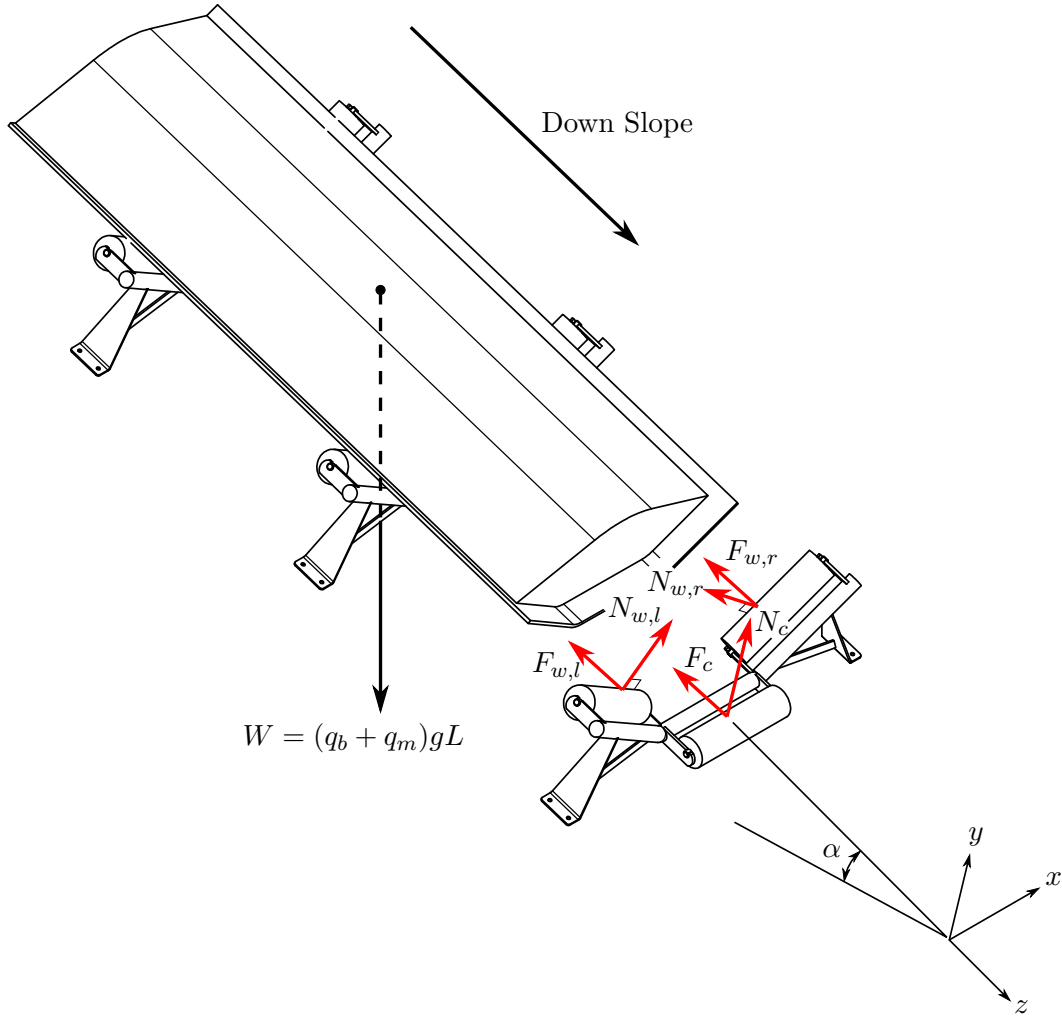


Figure 5.3: Three-dimensional free body diagram of an inclined conveyor belt

Substituting Equation 5.23 into 5.22 gives:

$$\begin{aligned} \eta \frac{L}{a} (\psi_c N_c + \psi_w N_w) &= \frac{L}{a} N \tan \alpha \\ \therefore \eta &= \frac{N}{\psi_c N_c + \psi_w N_w} \tan \alpha \end{aligned} \quad (5.24)$$

This represents the proportion of rollers that are required to be unidirectional so as to prevent the down slope slipping motion of an inclined conveyor belt. Finally, by introducing a safety factor (n), the specified proportion of unidirectional idlers (η_s) may be calculated according to:

$$\eta_s = \frac{nN}{\psi_c N_c + \psi_w N_w} \tan \alpha \quad (5.25)$$

where η_s is the proportion of unidirectional idlers required, taking a factor of safety (n) into account.

Although Equation 5.25 takes the differing resistive conditions of centre and wing rollers into account, it still only specifies the proportion relative to the total number of idler sets (not individual rollers) that need to be locked. If, for example, it is calculated that $\eta_s = 50\%$ for an inclined belt with 100 three-roll idlers, then 50 of the idler sets need to be installed with unidirectional centre and wing rollers.

5.2.3.2 Model 2 – Order Specific Proportion Calculating Method based on the Different Resistive Properties of Centre and Wing Rollers

Considering the significant difference in resistive force at the centre roller as compared with the wing rollers (particularly when the belt is loaded), it follows that Model 1 is wasteful. In order to prevent the installation of excessive quantities of expensive unidirectional rollers, centre rollers should be specified as unidirectional first and only when the required proportion of these exceeds 100% should unidirectional wing rollers be designated^b. The required proportion of unidirectional centre rollers ($\eta_{c,s}$) may be determined (in reference to Equation 5.25) by following the steps below:

$$\epsilon = \frac{nN}{\psi_c N_c} \tan \alpha \quad (5.26)$$

where ϵ is the unidirectional centre roller proportion factor (taking a factor of safety, n , into account), N is the overall normal force acting at the three-roll idler set, N_c is the normal force acting at the centre roller, ψ_c is the centre roller resistance coefficient, and α is the conveyor belt inclination angle.

Following this, the required proportion of unidirectional centre rollers ($\eta_{c,s}$) may be specified according to:

$$\eta_{c,s} = \begin{cases} \epsilon & \text{if } \epsilon < 1 \\ 1 & \text{if } \epsilon \geq 1 \end{cases} \quad (5.27)$$

where $\eta_{c,s}$ is the proportion of unidirectional centre rollers required (taking a factor of safety, n , into account).

If installing unidirectional centre rollers into all frames is not sufficient to prevent runaway (i.e. $\epsilon > 1$), then the required proportion of unidirectional wing rollers may be calculated:

$$\eta_{w,s} = (\epsilon - 1) \frac{\psi_c N_c}{\psi_w N_w}; \quad \text{if } \epsilon > 1 \quad (5.28)$$

where $\eta_{w,s}$ is the proportion of unidirectional wing rollers required (taking a factor of safety, n , into account), N_w is the normal force acting at the wing rollers of an idler set, and ψ_w is the wing roller resistance coefficient.

^bNote that the specification of unidirectional idlers does not consider the maintenance aspect of the belt system and the need to ensure correct replacement of these devices according to manufacturers instructions.

5.3 Numerical Model Results

With all empirical data collected, the attention of the research turned to developing a numerical model for the specification of unidirectional idler proportions. The mathematics followed in deriving the two unidirectional idler proportion models of interest (Model 1 and 2) is presented in Section 5.2. One of the defining characteristics of the inclined conveyor belt mechanical model used in the numerical analysis was the use of a modified friction formula (Equation 5.1). This new equation assumed that the resistance to belt slip was not solely due to friction, but a result of a combination of friction and non-friction effects. In this equation, a resistance coefficient (broken up into friction and non-friction coefficients) was defined and substituted in place of the conventional coefficient of friction.

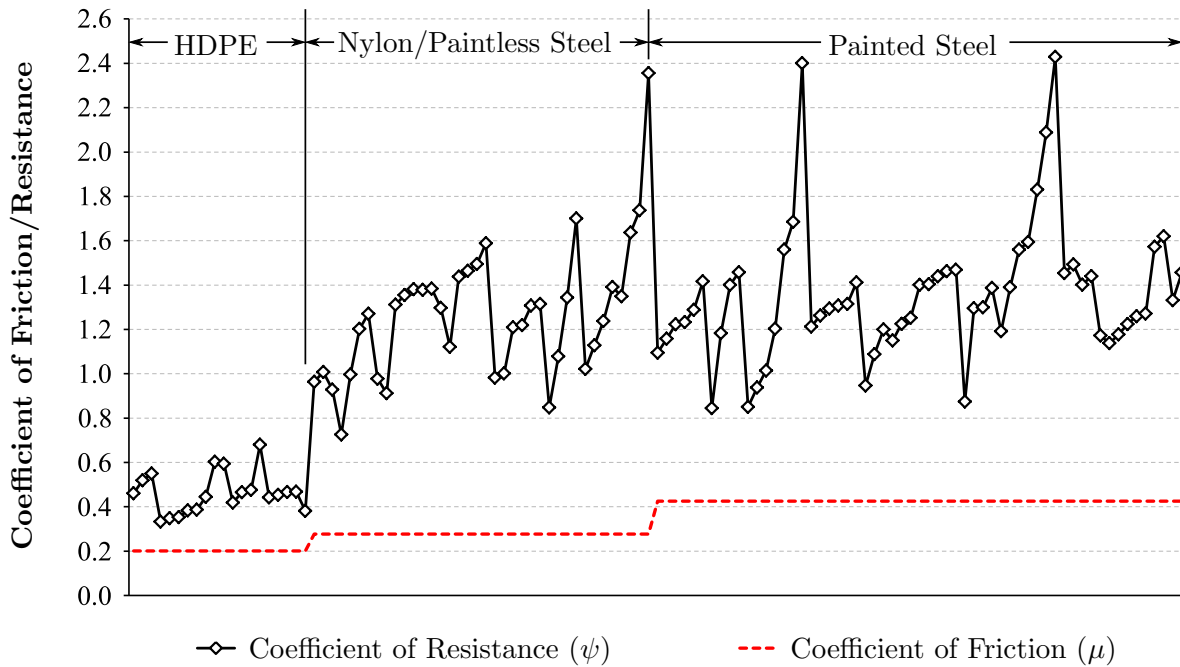
To test the assumption that belt slip resistance affected by a combination of both friction and non-friction components, it was necessary to compare the calculated resistance coefficients with the coefficient of friction data collected from the additional friction study. With measured slip forces and the normal loading conditions known for each UDR experiment, the corresponding resistance coefficients could be determined using the modified friction formula (Equation 5.1). From the friction experiments, the worst case (i.e. lowest coefficient of friction) for each roller material sample was identified. With the slight pressure dependence of friction in mind, these worst case values were calculated using a contact pressure of 15 kPa - corresponding to a 900 mm wide, 15 kg/m empty conveyor belt supported by $\varnothing 152$ mm rollers separated by a pitch of 1.2 m and demonstrating a roller wrap angle of 10° . These parameters were chosen in determining the worst case friction coefficients (see Table 5.1) because they result in a very low contact pressure, whilst still constituting realistic operating conditions.

Table 5.1: Minimum observed coefficients of friction (μ) at 15 kpa of contact pressure

Roller Type			
HDPE	Paintless Steel	Nylon	Painted Steel
$\mu = 0.20$	$\mu = 0.27$	$\mu = 0.28$	$\mu = 0.43$

Once the resistive and friction coefficients had been determined, a comparison graph was plotted as presented in Figure 5.4. This graph shows a scatter plot of the resistance coefficients calculated for every single unidirectional roller experiment that was run, grouped according to the roller in use and compared to the expected worse case coefficient of friction according to the additional friction study. The sequence of scatter data within each

roller material group is ordered in experimental run order. This figure shows that the modified resistance coefficient (ψ) is higher than the pure friction coefficient (μ) for every test case.



All Unidirectional Roller Experiments Conducted

Figure 5.4: Comparison between the inferred friction and resistance coefficients

With all resistive and friction coefficients known for each measured belt slip force and knowing the relationship between these two factors (Equation 5.2), the non-friction coefficient (λ), which accounts for the resistive effects of indentation resistance and belt sag, was calculated for each experiment. The process followed to determine these non-friction coefficients is summarised by the schematic diagram presented in Figure 5.5. This diagram reiterates how the slip force measurements and known loading conditions of each unidirectional roller experiment were used to determine resistance coefficients, which were then used, together with the results of the additional friction study, to infer resistance coefficients.

The use of empirically determined non-friction factors together with the worst case coefficients of friction, allows for the resistive force developed at the unidirectional idlers of a similar inclined belt set-ups to be found. To simplify the exercise of choosing a non-friction factor value for a particular inclined belt configuration, these factors were condensed into minimum recorded values corresponding to the varied levels of the three most significant belt conveying factors (belt load, idler pitch and troughing angle). The resulting non-friction coefficients are summarised in Table 5.2.

Unidirectional Roller Experiments

F – measured using load cells
 N – estimated based on known loading conditions

Additional Friction Study

$\mu_k = \mu_s = \mu$ – inferred from Shear Tester measurements

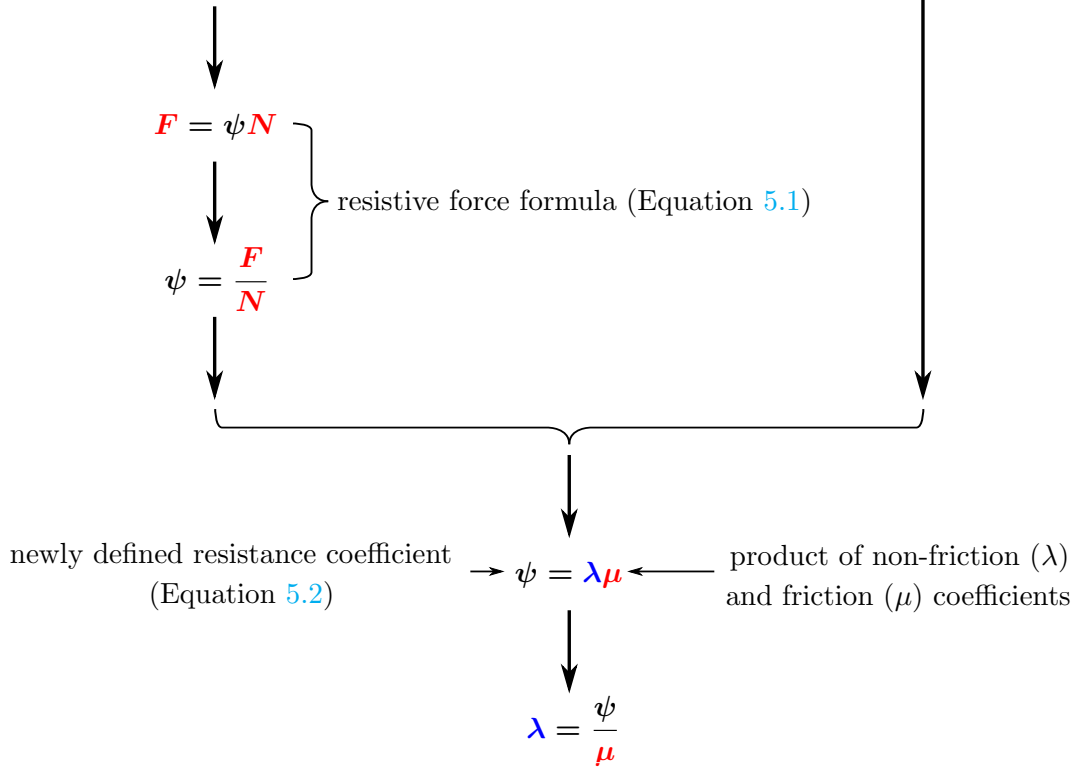


Figure 5.5: Schematic diagram of the relationship between empirical results and the unidirectional idler proportion specifying technique

Table 5.2: Minimum non-friction factors (λ) based on pitch, trough and loading condition

		Idler Pitch					
		1200 mm		1500 mm		1800 mm	
β	0°	$q_m = 0$	$\lambda = 2.30$	$q_m = 0$	$\lambda = 2.85$	$q_m = 0$	$\lambda = 2.70$
		$q_m = FL$	$\lambda = 3.42$	$q_m = FL$	$\lambda = ?$	$q_m = FL$	$\lambda = 3.29$
	35°	$q_m = 0$	$\lambda = 1.66$	$q_m = 0$	$\lambda = 3.07$	$q_m = 0$	$\lambda = 2.06$
		$q_m = FL$	$\lambda = 2.21$	$q_m = FL$	$\lambda = ?$	$q_m = FL$	$\lambda = 1.90$
	45°	$q_m = 0$	$\lambda = 2.00$	$q_m = 0$	$\lambda = 2.23$	$q_m = 0$	$\lambda = 2.80$
		$q_m = FL$	$\lambda = 3.70$	$q_m = FL$	$\lambda = ?$	$q_m = FL$	$\lambda = 3.13$

* FL = Full Load

This table summarises the lowest observed non-friction coefficients (λ) corresponding to the various levels of idler pitch (a), troughing angle (β) and mass per metre of material load (q_m). It should be noted that only empty conveyor belt experiments were conducted at an idler pitch of 1500 mm and as such certain table entries do not have non-friction coefficient values (denoted as $\lambda = ?$). Also note that in this table, the acronym *FL* stands for Full Load.

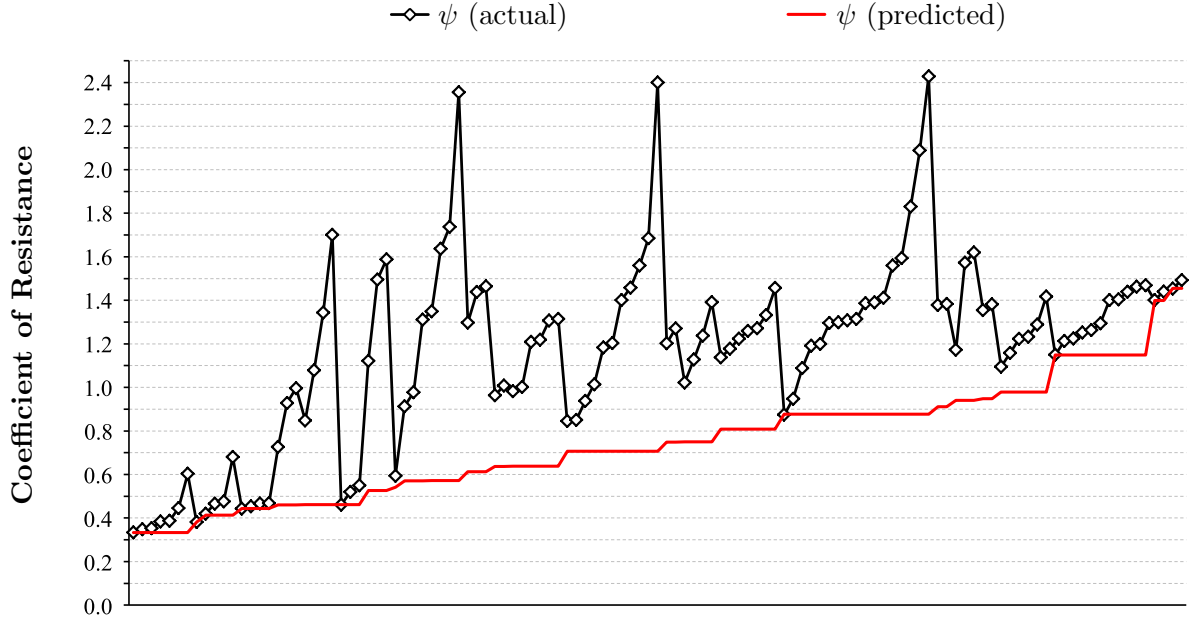
The minimum non-friction factors listed in Table 5.2 relate to specific configurations of idler pitch and troughing angle, and not the general case. As such, this table was modified into an eight cell matrix of non-friction values associated with sizeable factor level intervals that can be used for a broader variety of idler pitch and troughing angle combinations. This modified matrix is presented in Table 5.3.

Table 5.3: Condensed minimum non-friction factor matrix

	$1200 \leq a < 1500$ mm		$a \geq 1500$ mm	
$0^\circ \leq \beta < 35^\circ$	$q_m = 0$	$\lambda = 2.30$	$q_m = 0$	$\lambda = 2.70$
	$q_m = FL$	$\lambda = 3.42$	$q_m = FL$	$\lambda = 3.29$
$35^\circ \leq \beta \leq 45^\circ$	$q_m = 0$	$\lambda = 1.66$	$q_m = 0$	$\lambda = 2.06$
	$q_m = FL$	$\lambda = 2.21$	$q_m = FL$	$\lambda = 1.90$

The non-friction coefficients listed in this table were then used together with Equation 5.2 and the minimum observed coefficients of friction to predict the resistance coefficients ($\psi = \lambda\mu$) for each experimental set-up. These were then plotted together with experimentally determined resistance factors, as shown in Figure 5.6. This graph consists of a scatter plot of all the experimentally determined resistance coefficients re-ordered according to the ascending order of the resistance coefficient prediction model (indicated by the solid line).

Up until this point, the resistive forces used to determine the non-friction factors and derive the prediction models were based on measured forces averaged over one idler set. The difference between the resistive forces developed at the centre roller as compared to the wing rollers was not considered. The importance of taking this difference into account may be appreciated when considering the added costs incurred when installing unidirectional rollers in place of conventional rollers. With reference to the research of Liu et al. [14] (Section 2.2.2) it follows that in a three-roll idler system in which up to 74% of the idler normal force acts on the centre roller, the slight increase in braking force gained from installing unidirectional centre and wing rollers, as opposed to just unidirectional centre rollers, comes at an additional cost that is three times greater than if unidirectional centre rollers were used only.



Variation in the Levels of α , β , a , μ , D , T and q_m (All UDR Experiments)

Figure 5.6: Predicted vs actual resistance factors based on minimum values of λ

Adapting the current model to incorporate these differences was achieved by using the separately measured forces at the contact surface of centre and wing rollers. For details of the centre and wing roller normal force derivations used together with the separate roller slip force data to calculate centre and wing roller non-friction coefficients (λ_c and λ_w), refer to Sections 5.2.2 and Appendix E.1. Based on this, separate non-friction factor tables were drawn up for the centre and wing roller cases. The corresponding minimum non-friction coefficients for centre and wing rollers are summarised in Tables 5.4 and 5.5 respectively.

Table 5.4: Minimum non-friction factor values for locked centre rollers

	1200 $\leq a < 1500$ mm		$a \geq 1500$ mm	
$0^\circ \leq \beta < 35^\circ$	$q_m = 0$	$\lambda_c = 2.42$	$q_m = 0$	$\lambda_c = 2.70$
	$q_m = FL$	$\lambda_c = 4.45$	$q_m = FL$	$\lambda_c = 3.99$
$35^\circ \leq \beta \leq 45^\circ$	$q_m = 0$	$\lambda_c = 1.55$	$q_m = 0$	$\lambda_c = 1.57$
	$q_m = FL$	$\lambda_c = 1.71$	$q_m = FL$	$\lambda_c = 1.57$

Table 5.5: Minimum non-friction factor values for locked wing rollers

	$1200 \leq a < 1500$ mm		$a \geq 1500$ mm	
$0^\circ \leq \beta < 35^\circ$	$q_m = 0$	$\lambda_w = 2.21$	$q_m = 0$	$\lambda_w = 2.52$
	$q_m = FL$	$\lambda_w = 1.80$	$q_m = FL$	$\lambda_w = 2.23$
$35^\circ \leq \beta \leq 45^\circ$	$q_m = 0$	$\lambda_w = 1.19$	$q_m = 0$	$\lambda_w = 1.37$
	$q_m = FL$	$\lambda_w = 2.14$	$q_m = FL$	$\lambda_w = 1.80$

Using these tables, resistance coefficient prediction curves were plotted against actual coefficients for both cases. The comparison plots for the centre and wing cases are presented in Figures 5.7 and 5.8 respectively. Similarly to the overall idler set coefficient of resistance graph, these figures summarise the centre and wing roller coefficients of friction respectively in the form of scatter plots of all experimentally determined coefficients re-ordered according to the resistance coefficient prediction model (indicated by the solid line).

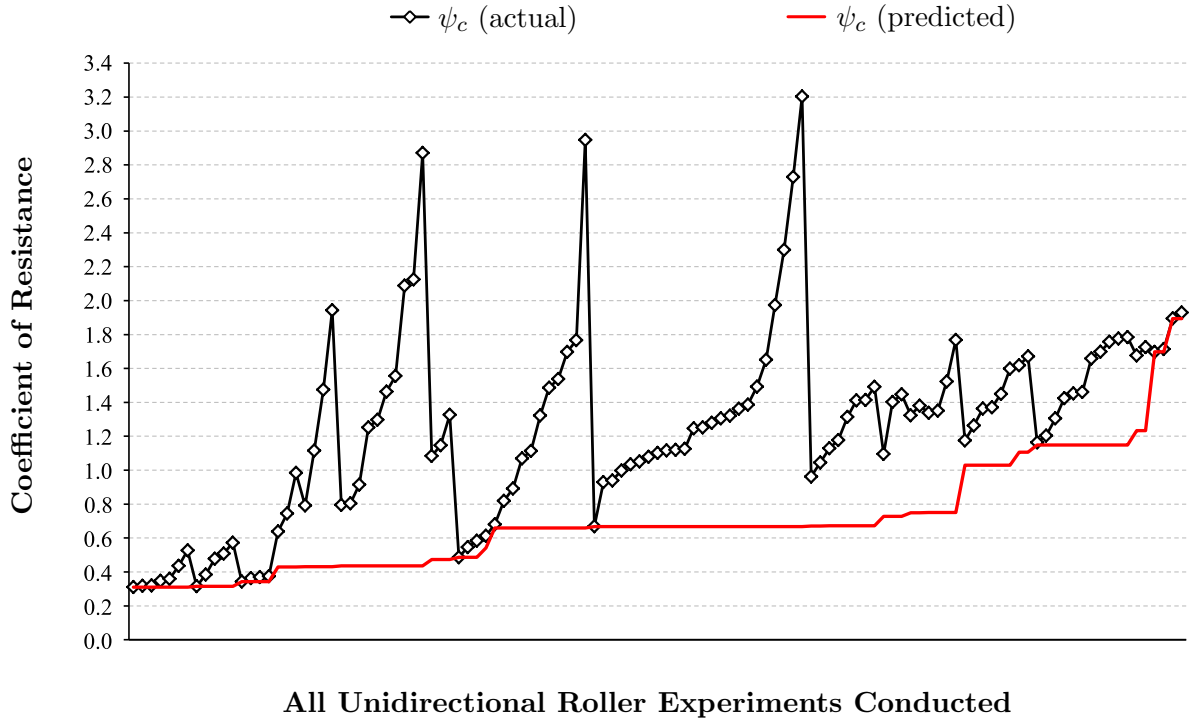


Figure 5.7: Predicted vs actual centre roller resistance factors based on minimum values of λ_c

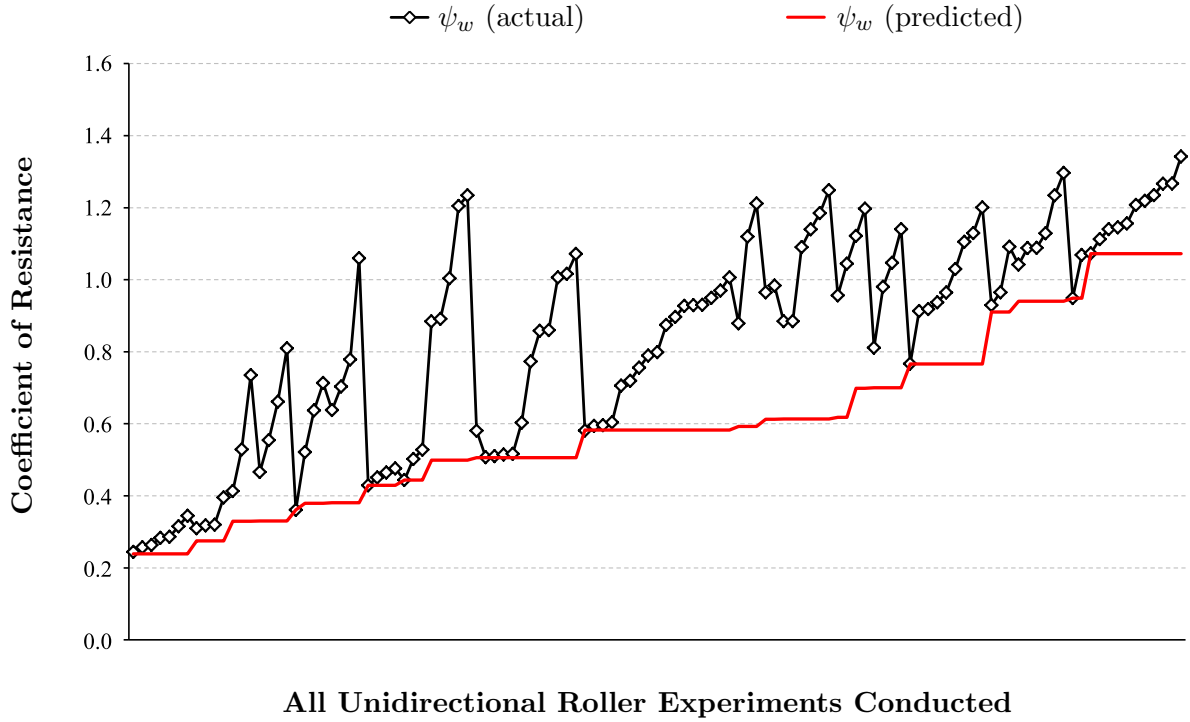


Figure 5.8: Predicted vs actual wing roller resistance factors based on minimum values of λ_c

5.3.1 Comparative Unidirectional Idler Specification Method Results

Now that the origin of the non-friction coefficients has been explained, the results of the two derived proportion prediction methods may be presented. Recall that while both models take the differing resistive conditions of centre and wing rollers into account, Model 1 specifies the required proportion of unidirectional rollers as a percentage of the total number of idler sets while Model 2 specifies centre rollers as unidirectional first before making use of any unidirectional wing rollers. For more details regarding the derivation of these models, refer to Section 5.2.3.

Considering these two models, as well as the preliminary mechanical analysis in Section 3.2, a comparison study was conducted for a wide variety of inclined belt operating conditions. The parameter values used for this study were based on an informal survey of conveyor design companies and encompasses the vast majority of installation parameter ranges. The parameters varied in this analysis included inclination angle (α), bulk material density (ρ), angle of surcharge (δ), belt width (B), troughing angle (β), idler pitch (a) and roller type.

The comparison of these three unidirectional idler proportion specifying methods was achieved with a numerical program that was designed to cycle through various levels of key conveyor belt factors and calculate the unidirectional idler proportions for the different models. The various factors and corresponding factor levels cycled through this program are summarised

in Table 5.6. A total of 3 240 different belt conveying scenarios were considered by varying these factors one at a time, at the levels indicated. The factors chosen to be varied in this case were all those that have a direct affect on the resistance coefficients and normal loads, as well as the principal factor of inclination angle.

Table 5.6: Factors cycled through the numerical simulation

	Factors						
	$\alpha(^{\circ})$	ρ (kg/m ³)	$\delta(^{\circ})$	B (m)	$\beta(^{\circ})$	a (m)	Roller Type
Levels	6	0	15	0.8	0	1.2	HDPE
	9	1000	20	1.4	35	1.5	Steel
	12	2000	25	2.0	45	1.8	
	15	3000					
	18						

It should be noted that when calculating required proportions of steel unidirectional rollers, the worst case coefficient of friction corresponding to a worn roller was used (i.e. $\mu = 0.27$ – see Figure 4.29). It should also be noted that a safety factor of 1.15 was used for all models.

The results of the numerical simulation are presented in Figure 5.9. This figure illustrates the unidirectional idler proportion predictions relating to the two derived models (Model 1 and 2), as well the Preliminary Model presented in Section 3.2^c. In this figure, the results are grouped by levels of inclination angle and arranged according to the ascending order of unidirectional idler proportion values as calculated using Model 2. Refer to Equations 3.9, 5.25, 5.27 and 5.28 to see how the various model unidirectional proportions were calculated.

Comparing the results of all three models, it may be seen that the incorporation of non-friction effects into the proportion prediction calculation (i.e. Models 1 and 2) typically results in a smaller number of unidirectional idlers required to provide the same braking capability as that calculated using the Preliminary Model (which is based purely on friction). However, it may also be noted that under certain circumstances, the required proportion of unidirectional idlers as specified by Models 1 and 2 is greater than that of the Preliminary Model. This is specifically the case when considering the lower friction coefficient of HDPE rollers as compared with that of steel.

^cRecall that this model only considered the effects of contact friction on the resistive braking performance of unidirectional idlers.

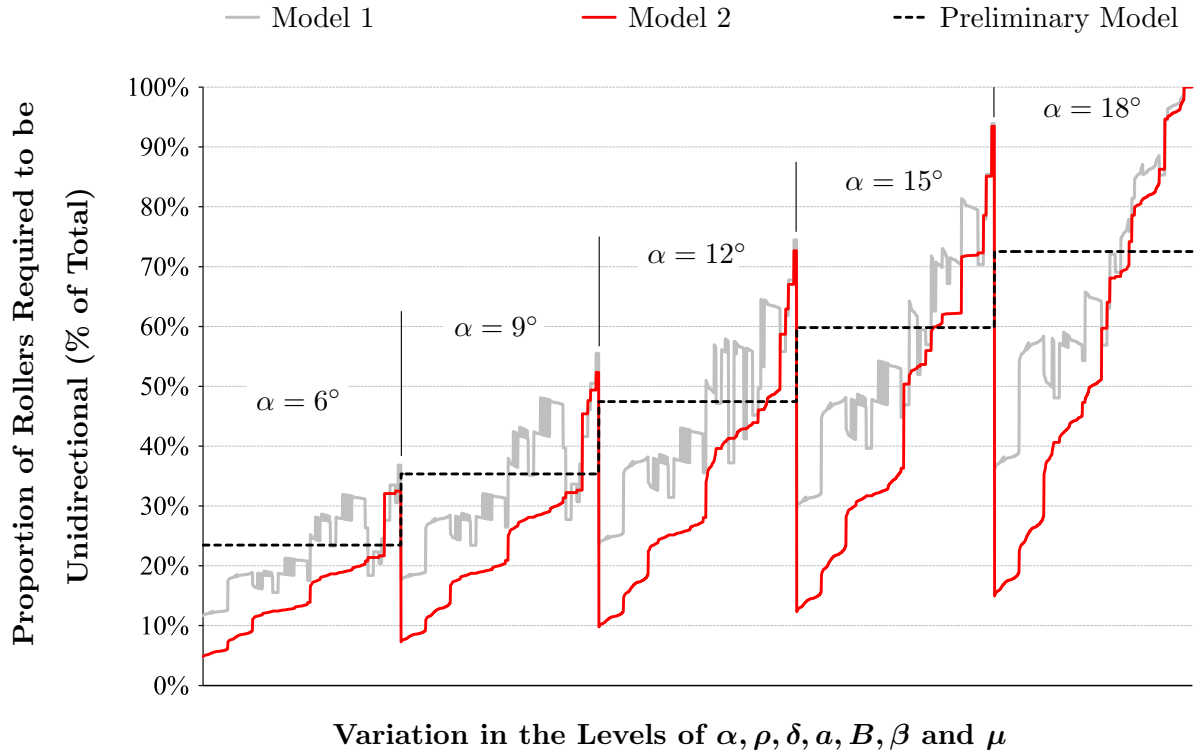


Figure 5.9: Comparison of unidirectional idler proportion prediction models

In addition to these differences with the Preliminary Model, the contrast between Models 1 and 2 may also be appreciated by referring to Figure 5.9. It may be seen that the method of specifying centre idler rollers as unidirectional first, and only making use of unidirectional wing rollers when all centre rollers are unidirectional (as prescribed in Model 2), results in typically smaller quantities of these devices being required. It may therefore be said that unidirectional idler roller proportions are best specified using Model 2, as this model ensures that enough unidirectional rollers will be installed without over-specifying these expensive devices.

At this point it is worth reiterating how the empirical results collected during experimentation may be used, together with the chosen unidirectional idler proportion specification technique (Model 2), to determine the unidirectional idler needs of a given inclined conveyor belt system. This relationship between empirical results and the proportion predicting technique is summarised by the schematic diagram presented in Figure 5.10. This figure illustrates how the friction and non-friction coefficient values, measured and inferred using the Jenike Direct Shear Tester and Unidirectional Roller Test Rig respectively, may be substituted into the various equations (together with other relevant conveyor belt parameter values) used to calculate the proportion of unidirectional idlers required to inhibit conveyor belt slip.

Unidirectional Roller Experiments

F – measured using load cells
 N – estimated based on known loading conditions

Additional Friction Study

$\mu_k = \mu_s = \mu$ – inferred from Shear Tester measurements

$$F = \psi N$$

$$\psi = \frac{F}{N}$$

resistive force formula (Equation 5.1)

newly defined resistance coefficient
(Equation 5.2)

$$\psi = \lambda \mu$$

product of non-friction (λ) and friction (μ) coefficients

$$\lambda = \frac{\psi}{\mu}$$

Unidirectional Roller Proportion Calculation (Model 2)

$$\eta = f(\alpha, \beta, \delta, \lambda, \mu, \omega, \rho, a, B, g, l_1, q_b)$$

← Equations 5.27 and 5.28

belt conveying parameters (see Nomenclature for definitions)

the proportion of rollers required to be unidirectional to prevent belt slip

Figure 5.10: Schematic diagram of process followed to calculate non-friction coefficients

Although the numerical simulation depicted in Figure 5.9 assessed a wide range of operating conditions, many of the scenarios considered are unlikely to occur in reality. The two factors that this particularly applies to are troughing angle and idler roller type. It is uncommon to design inclined conveyor belts without at least some degree of troughing [38]. It is also rare to find inclined belts installed with HDPE rollers; steel rollers being the standard [38].

With this in mind, the more common operating conditions were applied to the numerical simulation. The results of this condensed study, which only considered steel rollers and troughing angles larger than 30° , are shown in Figure 5.11. This figure presents a comparison between Model 2 (which was chosen as the best proportion specifying method) and the Preliminary Model (which considered the braking force of unidirectional idlers to be purely friction based). Similarly to the previous numerical model comparison graph, the results in this figure are grouped according to inclination angle and arranged in ascending proportion order.

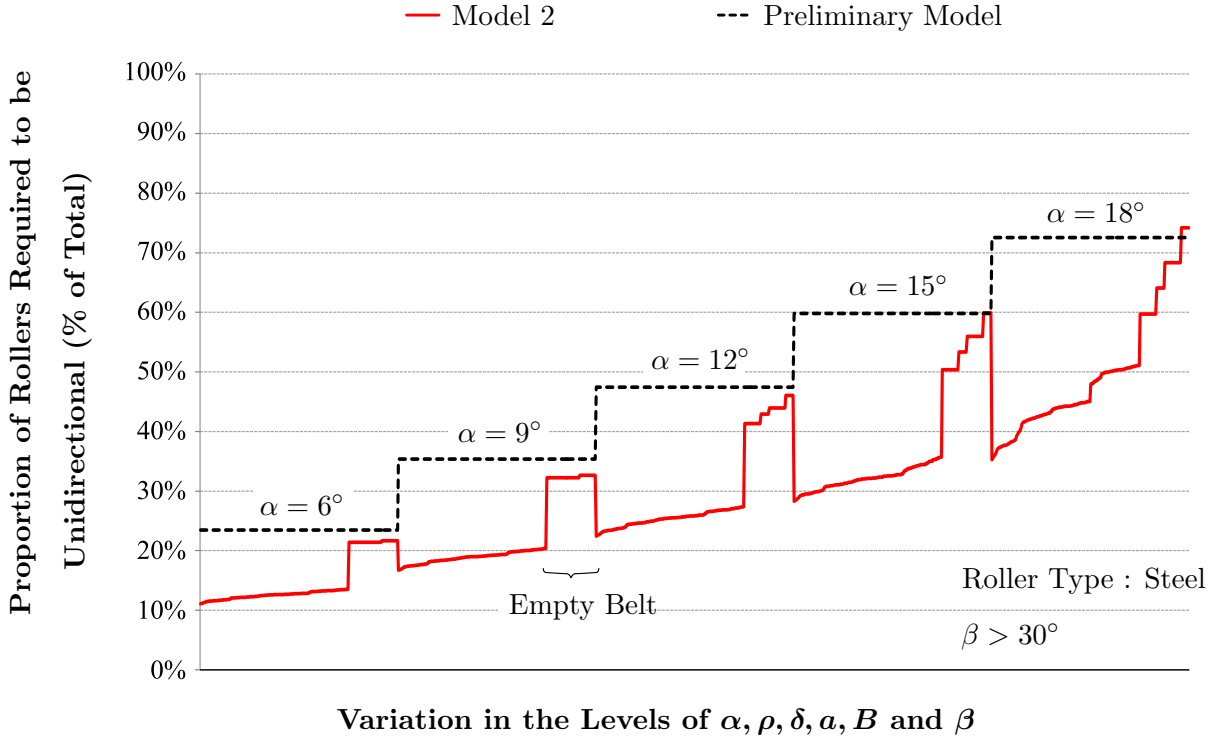


Figure 5.11: Comparison of proportion prediction models for common operating conditions

This figure shows that the required proportion of unidirectional rollers for common inclined belt operating conditions as determined using Model 2 is almost always lower than that calculated with the Preliminary Model. This is true for inclination angles below 18° . Additionally, Figure 5.11 reveals empty conveyor belts to constitute a worst-case scenario in terms of belt slip resistance.

Therefore, when using Model 2 to specify the required proportion of unidirectional rollers, calculations should be conducted assuming an empty conveyor belt. With this in mind, it may be noted that applying Model 2 to the worst-case scenario of an empty belt results in the specification of very similar unidirectional roller proportions to that of the Preliminary Model when the inclination angles are relatively small (less than 9°).

To expand on these findings, the unidirectional idler proportions of an empty belt (worst case scenario) supported by steel rollers (most common roller type) and with specific levels of idler pitch and troughing angle were calculated using the two models and compared, as presented in Figure 5.12. This figure shows that when levels of idler pitch and troughing angle are high, the Preliminary Model becomes more and more wasteful in its proportion specification with increasing inclination (as indicated by the growing deviation between the Preliminary Model and Model 2).

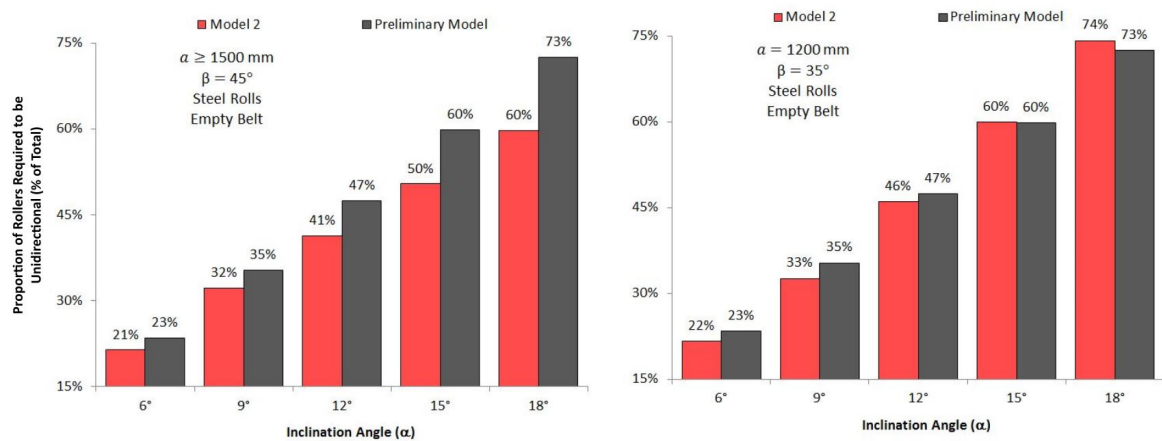


Figure 5.12: Comparison of proportion prediction models for specific operating conditions

Conversely to this, when levels of idler pitch and troughing angle are low, the relative proportion specification difference between the two models actually decreases with increasing inclination angle, until the model predictions actually converge (at $\alpha = 15^\circ$) and cross over (Figure 5.12). This crossing over of the two models, as demonstrated by inclination angles of 18° , is characterised by greater proportions of unidirectional idlers specified by Model 2 as compared to that of the Preliminary Model.

5.3.2 Case Study Results

To demonstrate how Model 2 may be used to determine the required proportion of unidirectional idlers (taking a factor of safety of 1.15 into consideration), three existing inclined belt installations were considered as case studies. The following section presents examples of the calculations used to specify unidirectional idler proportions based on several key belt conveying parameters. Refer to Chapter 5 for the detailed derivation of the proportion specifying method used (Model 2). Also refer to Appendix B for a step-by-step summary of the procedure to follow (including relevant tables and equations) when calculating the required proportion of unidirectional idlers for a specific installation.

Case Study 1

Consider an inclined conveyor belt with the operating parameters as listed in Table 5.7. Two scenarios will be compared for this first case study: a belt that is loaded to capacity with coal and an empty belt.

Table 5.7: Belt conveying parameters (Case Study 1)

Parameter	Symbol	Value	Unit
Inclination Angle	α	10.5	$^{\circ}$
Idler Pitch	a	2.25	m
Roller Type	-	Steel	-
Number of Rollers per Idler Set	-	3	-
Troughing Angle	β	45	$^{\circ}$
Belt Type	-	Steel Cord	-
Belt Width	B	1.8	m
Roller Face Length	l_1	0.67	m
Bulk Material	-	Coal	-
Bulk Density	ρ	900	kg/m ³
Surcharge Angle	δ	25	$^{\circ}$
Inclined Belt Length	L	450	m

Fully Loaded Belt

In order to determine the resistance coefficients (ψ), the non-friction coefficients (λ), friction coefficients (μ) and normal forces (N) needed to be calculated. The non-friction factors were found by referring to the Non-Friction Factor Tables. Using Tables 5.4 and 5.5, it was found that for the fully loaded case with a 2.25 m idler pitch and a 45° trough:

$$\lambda_c = 1.57 \quad \lambda_w = 1.80$$

Assuming a worst case steel roller friction coefficient of 0.27 (the minimum observed coefficient observed for a smooth steel sample at 15 kPa of contact pressure), the resistance coefficients were determined. These were calculated using Equation 5.2 as follows:

$$\psi_c = \lambda_c \cdot \mu = 1.57 \times 0.27 = 0.42$$

$$\psi_w = \lambda_w \cdot \mu = 1.80 \times 0.27 = 0.49$$

The idler set normal load was calculated using Equation 5.17:

$$N = ag \left\{ q_b + \rho \left[\frac{1}{6} \tan \delta \left(l_1 + (b - l_1) \cos \beta \right)^2 + \frac{1}{4} (b - l_1) \sin \beta \left(2l_1 + (b - l_1) \cos \beta \right) \right] \right\}$$

Before this equation could be used, q_b and b needed to be determined. An estimate for the belt mass per metre (q_b) was determined through the use of a belting catalogue [39] to be:

$$q_b = 40 \text{ kg/m}$$

The usable width of the belt (b) was calculated using Equation 5.5 as:

$$b = 0.9B - 0.05 = 0.9(1.8) - 0.05 = 1.57 \text{ m}$$

Now the normal force could be calculated using Equation 5.17:

$$N = (2.25)(9.79) \left\{ 40 + 900 \left[\frac{1}{6} \tan 25^\circ \left(0.67 + (1.57 - 0.67) \cos 45^\circ \right)^2 + \frac{1}{4} (1.57 - 0.67) \sin 45^\circ (2(0.67) + (1.57 - 0.67) \cos 45^\circ) \right] \right\} = 9.74 \text{ kN}$$

In order to determine the magnitude of the normal force acting at each centre roller, the area of material directly above these rollers (A_c) needed to be determined. This was calculated using Equation 5.13 as:

$$A_c = \frac{\tan \delta \left\{ 3l_1 \left[l_1 + (b - l_1) \cos \beta \right]^2 - l_1^3 \right\}}{12 \left[l_1 + (b - l_1) \cos \beta \right]} + \frac{1}{2} l_1 (b - l_1) \sin \beta$$

$$= \frac{\tan 25^\circ \left\{ 3(0.67) \left[0.67 + (1.57 - 0.67) \cos 45^\circ \right]^2 - (0.67)^3 \right\}}{12 \left[0.67 + (1.57 - 0.67) \cos 45^\circ \right]} + \frac{1}{2} (0.67)(1.57 - 0.67) \sin 45^\circ$$

$$\therefore A_c = 0.306 \text{ m}^2$$

Then the normal force at the centre rollers could be calculated using Equation 5.18 as:

$$N_c = a \left(q_b \left(\frac{l_1}{B} \right) + \rho A_c \right) g = 2.25 \left(40 \left(\frac{0.67}{1.8} \right) + 900(0.306) \right) (9.79) = 6.39 \text{ kN}$$

Considering the case of the wing rollers troughed at 45° , the normal coefficient was specified as $\omega = 1.9$ (from Equation 5.20) and the corresponding normal force was found using Equation 5.19 according to:

$$N_w = a\omega \left(q_b \left(1 - \frac{l_1}{B} \right) + \rho A_w \right) g = a\omega \left(q_b \left(1 - \frac{l_1}{B} \right) + \rho(A - A_c) \right) g$$

$$\therefore N_w = 2.25(1.9) \left(40 \left(1 - \frac{0.67}{1.8} \right) + 900(0.447 - 0.306) \right) (9.79) = 6.36 \text{ kN}$$

Finally, the proportion of idlers required to be unidirectional according to Prediction Model 2, including a factor of safety of 1.15, was determined using Equations 5.26 and 5.27 as follows:

$$\epsilon = \frac{nN}{\psi_c N_c} \tan \alpha = \frac{1.15(9.74)}{0.42(6.39)} \tan 10.5^\circ = 0.77$$

$$\epsilon < 1 \quad \implies \quad \therefore \eta_{c,s} = \epsilon = 0.77$$

This equates to an overall unidirectional idler proportion of 26%.

Empty Belt

In the empty belt scenario, the non-friction factors were found (using Tables 5.4 and 5.5) to be:

$$\lambda_c = 1.57 \quad \lambda_w = 1.37$$

Resistance factors were then calculated as:

$$\psi_c = \lambda_c \cdot \mu = 1.57 \times 0.27 = 0.42$$

$$\psi_w = \lambda_w \cdot \mu = 1.37 \times 0.27 = 0.37$$

The overall, centre and wing roller normal forces needed to determine the required unidirectional roller proportion, were calculated as presented below:

$$N = ag(q_b + \rho A) = (2.25)(9.79)(40 + 0) = 881 \text{ N}$$

$$N_c = a \left(q_b \left(\frac{l_1}{B} \right) + \rho A_c \right) g = (2.25) \left(40 \left(\frac{0.67}{1.8} \right) + 0 \right) (9.79) = 328 \text{ N}$$

$$N_w = a \omega \left(q_b \left(1 - \frac{l_1}{B} \right) + \rho A \right) = (2.25)(1.9) \left(40 \left(1 - \frac{0.67}{1.8} \right) + 0 \right) (9.79) = 1051 \text{ N}$$

Finally, the proportion of idlers required to be unidirectional according to Prediction Model 2, were determined as follows:

$$\epsilon = \frac{nN}{\psi_c N_c} \tan \alpha = \frac{1.15(881)}{0.42(328)} \tan 10.5^\circ = 1.36$$

$$\epsilon > 1 \quad \implies \quad \therefore \eta_{c,s} = 1$$

Since $\epsilon > 1$, specifying all centre rollers as unidirectional was not enough to prevent belt slip and a proportion of wing rollers were required to be unidirectional as well. This proportion was found as follows:

$$\eta_{w,s} = (\epsilon - 1) \frac{\psi_c N_c}{\psi_w N_w} = (1.36 - 1) \frac{0.42(328)}{0.37(1051)} = 0.13$$

This means that in order to prevent belt runaway, all the centre rollers and 13% of the wing rollers need to be specified as unidirectional idlers. Knowing the length of the inclined section of the belt ($L = 450 \text{ m}$) and the idler pitch ($a = 2.25 \text{ m}$), the total number of idler sets in this system is 200. Therefore a total of 200 unidirectional centre rollers and $200 \times 0.13 \times 2 = 52$ unidirectional wing rollers are required. This equates to an overall unidirectional idler proportion of 42%.

Case Study 2

Consider an inclined conveyor belt with the operating parameters as listed in Table 5.8. In this case study, since it has been demonstrated that an unloaded conveyor belt is the worst case when it comes to belt slip, only the empty belt scenario was considered. In this scenario, the non-friction factors were found (using Tables 5.4 and 5.5) as:

$$\lambda_c = 1.57 \quad \lambda_w = 1.37$$

Resistance factors could then be calculated as:

$$\psi_c = \lambda_c \cdot \mu = 1.57 \times 0.27 = 0.42$$

$$\psi_w = \lambda_w \cdot \mu = 1.37 \times 0.27 = 0.37$$

Before the various normal forces could be calculated, the mass of the conveyor belt (q_b) needed to be known. An estimate for the belt mass per metre was determined through the use of a belting catalogue [39] to be:

$$q_b = 25 \text{ kg/m}$$

Table 5.8: Belt conveying parameters (Case Study 2)

Parameter	Symbol	Value	Unit
Inclination Angle	α	12.5	°
Idler Pitch	a	1.5	m
Roller Type	-	Steel	-
Number of Rollers per Idler Set	-	3	-
Troughing Angle	β	35	°
Belt Type	-	Steel Cord	-
Belt Width	B	1.05	m
Roller Face Length	l_1	0.39	m
Bulk Material	-	Iron Ore	-
Inclined Belt Length	L	197	m

The overall, centre and wing roller normal forces needed to determine the required unidirectional roller proportion, could then be calculated as presented below:

$$N = ag(q_b + \rho A) = (1.5)(9.79)(25 + 0) = 367 \text{ N}$$

$$N_c = a \left(q_b \left(\frac{l_1}{B} \right) + \rho A_c \right) g = (1.5) \left(25 \left(\frac{0.39}{1.05} \right) + 0 \right) (9.79) = 136 \text{ N}$$

$$N_w = a \omega \left(q_b \left(1 - \frac{l_1}{B} \right) + \rho A \right) = (1.5)(1.43) \left(25 \left(1 - \frac{0.39}{1.05} \right) + 0 \right) (9.79) = 330 \text{ N}$$

Finally, the proportion of idlers required to be unidirectional according to Prediction Model 2, were determined as follows:

$$\begin{aligned}\epsilon &= \frac{nN}{\psi_c N_c} \tan \alpha = \frac{1.15(367)}{0.42(136)} \tan 12.5^\circ = 1.64 \\ \epsilon &> 1 \quad \implies \quad \therefore \eta_{c,s} = 1 \\ \eta_{w,s} &= (\epsilon - 1) \frac{\psi_c N_c}{\psi_w N_w} = (1.64 - 1) \frac{0.42(136)}{0.37(330)} = 0.30\end{aligned}$$

This means that in order to prevent belt runaway, all the centre rollers and 30% of the wing rollers need to be specified as unidirectional idlers. Knowing the length of the inclined section of the belt ($L = 197$ m) and the idler pitch ($a = 1.5$ m), the total number of idler sets in this system is approximately 130. Therefore a total of 130 unidirectional centre rollers and $130 \times 0.3 \times 2 = 78$ unidirectional wing rollers are required. This equates to an overall unidirectional idler proportion of 53%.

Case Study 3

Consider an inclined conveyor belt with the operating parameters as listed in Table 5.9. In this case study, as in the previous one, only the empty belt scenario was considered.

Table 5.9: Belt conveying parameters (Case Study 3)

Parameter	Symbol	Value	Unit
Inclination Angle	α	14.7	$^\circ$
Idler Pitch	a	1.4	m
Roller Type	-	Steel	-
Number of Rollers per Idler Set	-	3	-
Troughing Angle	β	35	$^\circ$
Belt Type	-	Fabric Carcass	-
Belt Width	B	0.75	m
Roller Face Length	l_1	0.29	m
Bulk Material	-	Coal Ore	-
Inclined Belt Length	L	38	m

The non-friction factors were found (using Tables 5.4 and 5.5) as:

$$\lambda_c = 1.55 \quad \lambda_w = 1.19$$

Resistance factors could then be calculated as:

$$\psi_c = \lambda_c \cdot \mu = 1.55 \times 0.27 = 0.42$$

$$\psi_w = \lambda_w \cdot \mu = 1.19 \times 0.27 = 0.32$$

Before the various Normal forces could be calculated, the mass of the conveyor belt (q_b) needed to be known. An estimate for the belt mass per metre was determined through the use of a belting catalogue [39] to be $q_b = 10$ kg/m.

The overall, centre and wing roller normal forces needed to determine the required unidirectional roller proportion, could then be calculated as presented below:

$$N = ag(q_b + \rho A) = (1.4)(9.79)(10 + 0) = 137 \text{ N}$$

$$N_c = a \left(q_b \left(\frac{l_1}{B} \right) + \rho A_c \right) g = (1.4) \left(10 \left(\frac{0.29}{0.75} \right) + 0 \right) (9.79) = 53.0 \text{ N}$$

$$N_w = a \omega \left(q_b \left(1 - \frac{l_1}{B} \right) + \rho A \right) = (1.4)(1.43) \left(10 \left(1 - \frac{0.29}{0.75} \right) + 0 \right) (9.79) = 120 \text{ N}$$

Finally, the required proportion unidirectional idlers was determined as follows:

$$\epsilon = \frac{nN}{\psi_c N_c} \tan \alpha = \frac{1.15(137)}{0.42(53.0)} \tan 14.7^\circ = 1.86$$

$$\epsilon > 1 \quad \implies \quad \therefore \eta_{c,s} = 1$$

$$\eta_{w,s} = (\epsilon - 1) \frac{\psi_c N_c}{\psi_w N_w} = (1.86 - 1) \frac{0.42(53.0)}{0.32(120)} = 0.50$$

This means that in order to prevent belt runaway, all the centre rollers and 50% of the wing rollers need to be specified as unidirectional idlers. Knowing the length of the inclined section of the belt ($L = 38$ m) and the idler pitch ($a = 1.4$ m), the total number of idler sets in this system is approximately 27. Therefore a total of 27 unidirectional centre rollers and $27 \times 0.5 \times 2 = 27$ unidirectional wing rollers are required. This equates to an overall unidirectional idler proportion of 67%.

Hypothetical Case

With reference to the three cases, it was observed that all of these real world examples could safely make use of unidirectional idlers. There were no scenarios considered where more than 100% of the centre and wing rollers were required to be unidirectional. To appreciate the operating conditions that need to be present for a total unidirectional idler proportion just short of 100% to be specified, consider the hypothetical scenario presented in Table 5.10.

Table 5.10: Belt conveying parameters (Hypothetical Case)

Parameter	Symbol	Value	Unit
Inclination Angle	α	16	°
Idler Pitch	a	1.2	m
Roller Type	-	HDPE	-
Number of Rollers per Idler Set	-	3	-
Troughing Angle	β	35	°
Belt Type	-	Fabric Carcass	-
Belt Width	B	0.9	m
Roller Face Length	l_1	0.34	m
Inclined Belt Length	L	120	m

The non-friction factors were found (using Tables 5.4 and 5.5) to be:

$$\lambda_c = 1.55 \quad \lambda_w = 1.19$$

Assuming a worst case HDPE roller friction coefficient of 0.20 (the minimum observed coefficient at 15 kPa of contact pressure), the resistance factors could be calculated as:

$$\psi_c = \lambda_c \cdot \mu = 1.55 \times 0.20 = 0.31$$

$$\psi_w = \lambda_w \cdot \mu = 1.19 \times 0.20 = 0.24$$

Before the various Normal forces could be calculated, the mass of the conveyor belt (q_b) needed to be known. An estimate for the belt mass per metre was determined through the use of a belting catalogue [39] to be:

$$q_b = 15 \text{ kg/m}$$

The overall, centre and wing roller normal forces needed to determine the required unidirectional roller proportion, could then be calculated as presented below:

$$\begin{aligned}
N &= ag(q_b + \rho A) = (1.2)(9.79)(15 + 0) = 176 \text{ N} \\
N_c &= a \left(q_b \left(\frac{l_1}{B} \right) + \rho A_c \right) g = (1.2) \left(15 \left(\frac{0.34}{0.9} \right) + 0 \right) (9.79) = 66.6 \text{ N} \\
N_w &= a \omega \left(q_b \left(1 - \frac{l_1}{B} \right) + \rho A \right) = (1.2)(1.43) \left(15 \left(1 - \frac{0.34}{0.9} \right) + 0 \right) (9.79) = 157 \text{ N}
\end{aligned}$$

Finally, the proportion of idlers required to be unidirectional according to Prediction Model 2, were determined as follows:

$$\begin{aligned}
\epsilon &= \frac{nN}{\psi_c N_c} \tan \alpha = \frac{1.15(176)}{0.31(66.6)} \tan 16^\circ = 2.81 \\
\epsilon &> 1 &\implies \therefore \eta_{c,s} = 1 \\
\eta_{w,s} &= (\epsilon - 1) \frac{\psi_c N_c}{\psi_w N_w} = (2.811 - 1) \frac{0.31(66.6)}{0.24(157)} = 0.99
\end{aligned}$$

This means that in order to prevent belt runaway, all the centre rollers and all of the wing rollers need to be specified as unidirectional idlers. Knowing the length of the inclined section of the belt ($L = 120$ m) and the idler pitch ($a = 1.2$ m), the total number of rollers in this three-roll system is 300. Therefore all 300 rollers are required to be unidirectional.

5.3.3 Condensed Unidirectional Idler Proportion Matrix

After running numerical simulations and studying the various real world cases, it was desired to condense all the findings into a compact matrix that may be used in industry as an initial approximate unidirectional idler proportion specifying tool. This matrix was generated by using the Model 2 formula to calculate the proportion of centre and wing rollers required to be unidirectional at varying angles of inclination for an absolute worst case scenario. This being an empty conveyor belt troughed at 35° and supported by idlers separated by a 1.2 m idler pitch. The combination of a 1.2 m idler pitch and a 35° troughing angle constitutes a worst case scenario because of the very low non-friction coefficient that results from this particular pairing (see Tables 5.3 and 5.4). The results of this exercise are presented in Table 5.11.

Table 5.11: Unidirectional idler proportions required at various inclination angles

α (°)	Steel Unidirectional Idler Proportions			HDPE Unidirectional Idler Proportions		
	Centre Rollers	Wing Rollers	Overall	Centre Rollers	Wing Rollers	Overall
5.0	60%	0%	20%	89%	0%	30%
6.0	72%	0%	24%	100%	3%	36%
7.0	84%	0%	28%	100%	13%	42%
8.0	96%	0%	32%	100%	23%	49%
9.0	100%	5%	37%	100%	33%	55%
10.0	100%	13%	42%	100%	43%	62%
11.0	100%	20%	47%	100%	53%	69%
12.0	100%	28%	52%	100%	64%	76%
13.0	100%	36%	57%	100%	74%	83%
14.0	100%	43%	62%	100%	84%	89%
15.0	100%	51%	67%	100%	95%	96%
16.0	100%	59%	73%	100%	100%	100%
17.0	100%	67%	78%	UNSAFE	UNSAFE	UNSAFE
18.0	100%	75%	83%	UNSAFE	UNSAFE	UNSAFE
19.0	100%	83%	89%	UNSAFE	UNSAFE	UNSAFE
20.0	100%	91%	94%	UNSAFE	UNSAFE	UNSAFE
21.0	100%	100%	100%	UNSAFE	UNSAFE	UNSAFE

Chapter 6 Discussion

6.1 Unidirectional Roller Experiments

Before the results of the factor screening experimental programme are considered, a typical example of the slip force data collected during a UDR Test Rig experiment will be reviewed. Figure 4.1 shows the typical resistive force trends experienced at the left, centre and wing rollers (denoted by pink, red and blue lines respectively). Examining the differences between these three responses it may be seen that the centre roller exhibits the highest resistive force. This finding makes sense when considering the inherently higher normal forces present at the centre rollers, thus resulting in larger friction forces being developed there.

The particular experiment presented in Figure 4.1 refers to the scenario of a conveyor belt without any material load applied. When, however, belt loading is introduced, the contrast observed between centre and wing roller resistances is more appreciable. Figure 4.2 demonstrates the greater difference between centre and wing roller slip that results from the introduction of load. This phenomenon may be explained when considering that in this scenario, the majority of the conveyed material exists in the region directly above the centre rollers, thus resulting in significantly larger normal forces, belt indentation and, ultimately, slip resisting friction forces. This reality was demonstrated in a recent study by Liu et al. [14], which revealed the total percentage of normal force on the centre belt section of a 35° troughed three-roll idler system to vary from 66.2% to 74.2%.

Although the individual centre and wing roller resistive responses provide insight into the interaction between the different rollers at the onset of belt slip, the focus of the UDR experiments was to quantify the overall resistive force at slip. With this in mind, the results discussed from this point on will be related to the total resistive responses recorded during experimentation (indicated by the black line in Figures 4.1 and 4.2).

As mentioned in Section 4.1.1, each UDR experiment consisted of five experimental iterations that allowed for the variability of the force measurements to be inferred. Running five

iterations of each experiment also helped to improve the accuracy of the average slip force estimates. An example of the slip force variation observed in the five runs of a typical UDR experiment is shown in Figure 4.3. From this graph, it may be seen that the typical variation in measured resistive force is relatively small (as confirmed by a standard deviation of 6.0 N in this particular case) and the repeatability of these experiments is therefore favourable. For more evidence of the typically low measured variation relative to the mean slip force, refer to the complete list of test run standard deviation values presented in Table E.2.

Since it has been shown that the variability in the measured resistive force is acceptably small, the mean case for each experiment may be examined. Considering the average resistive force response of a typical UDR experiment (Figure 4.4) a number of key observations may be made. Following the overall resistive force line it may be seen that after the instance of initial belt pull (when time = 0), the average idler set resistive force first increases sharply, then levels off for a fraction of a second, and increases sharply again until a maximum resistive force peak is reached. After this peak, the resistive force levels off to an oscillating, but almost constant, resistive force value that is perceptibly lower than the peak value. This phenomenon is not isolated to the particular experiment presented in the figure, but instead extends to many of the experiments conducted^a. To see the similarities in resistive force relationship between conducted experiments, refer to the complete set of resistive force graphs presented in Appendix E.2.

Figure 4.4 shows three characteristic response features worth discussing: the momentary levelling off of the resistive force, the peak resistance observed and the constant measured force oscillations after the peak. In terms of the “kink” in the graph depicted by the momentary levelling off of the measured resistive force during increasing belt pull, there are two possible explanations. Firstly, the kink (or inflection region) may be the result of the belt slipping at one of the locked idler sets before the other^b. When the belt pulling force increased, it is possible that the belt slipped over the locked idler set closest to the belt pulling end of the test rig first, at the instance of the inflection region, but did not slip over the remaining locked idler set due to the drop in belt tension that would have occurred when the lower belt tension region between the two locked idler sets combined with the higher tension present in the section of belt being pulled. However, this explanation isn’t likely to be the real cause of

^aNote that not all experiments exhibited a distinct friction force peak at the start of belt slip. The reason for the absence of this peak in certain experiments was unknown, however of no consequence to the aim of the friction force measurements – to determine the average kinetic slip force of a conveyor belt being pulled over stationary rollers.

^bIt should be recalled at this point that the rollers of two idler sets were locked in place in order to simulate unidirectional idlers, and that the resistive force trends presented in the various figures thus far depict force values averaged over the two locked idler sets.

the inflection region. Upon closer inspection it was observed that the inflection region was present in the individual resistive force responses of both of the locked idler sets.

The second and most plausible reason for the inflection region may be explained by considering the mechanism of friction and how it interacts with the other sources of belt slip resistance. Before the belt can slip over the locked idlers, all forms of belt resistance need to be overcome. This means that even when the belt friction limit is reached, the belt will not slip until there is enough belt pulling force to overcome the non-friction sources of slip resistance (such as belt indentation and sag). With this in mind, it is likely that the momentary levelling off of the measured resistive force is a result of the belt-roller friction limit being reached, and the succeeding increase in resistive force is a result of the remaining resistance inherent in the conveyor belt which needs to be pulled up and over the idlers before slip can occur.

The period of increasing resistive force ends with a peak force value being reached. This is the point at which the overall resistance limit has just been exceeded and the belt begins to slip over the locked idlers. It is important to note that before this instance in time, the belt is stationary. After this peak value, however, the belt has been set in motion and continues to slip over the locked idlers at a constant speed. The small fluctuations in resistive force that may be seen after the resistive peak are indicative of a typical stick-slip motion of the belt sliding over the locked rollers. It is important to note that the average magnitude of the oscillating stick-slip region of the resistive force curve (equal to 297 N in the example shown in Figure 4.4) is appreciably lower than the peak value that precedes it.

This difference in resistive force at the onset of belt slip as compared to the ensuing continuation of belt slip, is related to the differences in static and kinetic friction. In this case, however, the resistance to slip is not purely frictional. Instead, the distinguishable difference in the peak and oscillatory measured forces may be classified as static and kinetic resistance. When the static resistance limit is exceeded, the state of the belt changes from stationary to slipping and the belt begins to experience kinetic resistance. It is this measure of kinetic resistance (or kinetic slip) that is of pivotal importance to the assessment of unidirectional idler performance. With the magnitude of this element known for a particular operating condition, the unidirectional idler braking capacity may be determined. It is worth noting that kinetic slip was chosen instead of static slip in order to be more conservative when calculating the required proportions of unidirectional idlers. Although belt slip may only occur once the static resistance limit has been exceeded, if design calculations are based on the slightly lower kinetic resistance then the specification of required unidirectional idler proportions will have an additional built in safety factor while still being representative of the true resistive limit.

6.1.1 Factor Screening Experiments

The main purpose of the factor screening set of experiments was to employ statistical Design of Experiment techniques to determine, with sufficient confidence, which belt conveying parameters had the greatest effect on the braking performance of unidirectional idlers. To answer this question, a set of 32 experiments (plus replicates) was conducted using the UDR Test Rig. In this investigation, six belt conveying factors were varied between two levels each (low and high), resulting in a 2_{VI}^{6-1} fractional factorial design (see Table A.1 for details). The kinetic slip force data collected during the course of this investigation (summarised in Figure 4.5) was loaded into a design of experiments software package that was used to perform statistical analysis on the data.

Using this software package, an ANOVA Table was generated (Table 4.1) in order to test for factor significance. The key performance indicator in this table is the column of P-values. Recall that in statistics, the P-value is the probability that the experimentally observed result could occur when the null hypothesis is true. In this case, the null hypothesis states that the various factors, and their interactions, have no effect on the measured outcome. If the P-value is exceptionally low (P-value < 0.05), it means that the observed result is very unlikely to occur when the null hypothesis is true - therefore the null hypothesis must be rejected and the alternative hypothesis may be accepted. In this case the alternative hypothesis is that a particular factor or interaction has a significant effect on the measured belt slip. To put this paragraph simply; if the P-value of a specific factor/interaction is lower than 0.05 then the effect of that factor/interaction may be considered significant.

With this in mind, the P-values presented in Table 4.1 were examined to determine factor significance. To help emphasize which factors had the greatest effect on unidirectional idler performance, the table included asterisks next to the significant factors. From this it may be seen that idler pitch, troughing angle and belt tension were the most consequential factors, the pitch-trough interaction demonstrated a medium significance level and the roller type main effect and roller type-diameter interaction were marginally significant.

Although the ANOVA table summarised the factor significance, the results of the analysis of variance performed were more appreciable when presented graphically by a half-normal plot. Recall that if a source of variation complies with the null hypothesis, then its associated half-normal plot point will fall close to the insignificance line. Knowing this, Figure 4.6 clearly shows just how significant the various factors and interactions were. From this graph it may be seen that the sources of variation that were identified as significant by the ANOVA Table deviated noticeably from the insignificance line, thus confirming their significance.

6.1.1.1 Significant Factors

The exceptional significance of idler pitch, denoted by its substantial deviation from the insignificance line, was particularly appreciable. This high level of consequence makes sense when considering the effect that idler pitch has on belt friction and sag. The greater the longitudinal spacing is between successive idler sets, the greater the loads being supported by each individual idler set will be. This results in a greater friction force being developed according to the friction equation ($F = \mu N$). In addition to this, a larger idler pitch will result in a greater maximum vertical deflection experienced at the midway point between two idler sets. This increase belt deflection results in an increase in belt sag, which means that a greater downward pulling force is required to drag the belt up and over each idler set.

According to Figure 4.6 the belt conveying factor with the second greatest effect on belt slip was troughing angle. Similarly to idler pitch, varying the angle of belt trough affects slip resistance due to belt friction and sag. When a belt is bent into a trough, elastic potential energy is stored in the belt. It is this stored energy that tries to push the belt flat again, thus resulting in greater loading (i.e. normal force) at the wing rollers. This increase in wing loading with increased troughing angle was proven in a study that found wing roller normal force to be 1.2 and 1.9 times greater than gravity acting alone for a 30° and 45° troughed belt respectively [13]. Again, an increase in the normal loading condition results in a higher belt slip resisting friction force at the contact surface of a unidirectional idler.

In terms of non-friction resistance, introducing a troughed belt profile results in a type of three-dimensional belt sag. When a conveyor belt is flat (i.e. $\beta = 0$) the only type of sag present is the conventional vertical deflection that follows a sine-wave-like contour between two successive idler sets. However, when the belt is troughed, there is also a type of belt sag that propagates from the wing rollers as a result of the belt trough opening up and flattening out slightly between adjacent idler sets. In order for the conveyor belt to slip down the incline, the gravitational pull of the belt needs to be sufficient enough to flex the sides of the conveyor belt to the troughing profile at the idler sets.

Figure 4.6 reveals belt tension to be the next greatest contributor to slip resistance. However, although the magnitude of the belt tension main effect may seem reasonable, its direction (depicted by the corresponding positive gradient main effect line presented in Figure 4.7) is a bit more difficult to understand. Before this phenomenon is discussed, the remaining half-normal plot related insights will be explored.

Closer to the insignificance line, the factor of roller type is found together with the interaction pairs of pitch-trough and roller type-diameter. These factors, although significant according to their P-values, had a much smaller influence on belt slip when compared with

idler pitch, troughing angle and belt tension. In addition to these sources of variation, the three-way interaction between inclination angle, idler pitch and belt tension was also seen to be significant. However, as was mentioned in Section 2.3, the main focus of the investigation was to determine which factors and two factor interactions had the greatest impact on unidirectional idler performance (due to the inherent confounding of high-order interactions in fractional factorial designs). As such the significant three factor interaction identified by the half-normal plot is not important in this case and can be ignored.

With regard to the last significant factor (roller type), the comparatively low significance score demonstrated by the half-normal plot (Figure 4.6) came as somewhat of a surprise. It was initially expected that the roller type would contribute much more appreciably to the braking capability of the locked rollers; because of the consequential effect that roller material has on surface friction. Upon closer inspection, however, it was deduced that the range of coefficients of friction tested was fairly small. Even if roller type strongly influenced unidirectional idler performance, if the rollers compared had similar COFs then the differences in measured belt slip will be slight. This assumption was later confirmed by the results of the additional friction study which showed painted steel and Nylon COFs to be 0.4 and 0.3 respectively (see Table 5.1). This finding necessitated the study of HDPE rollers, which demonstrate a much lower coefficient of friction. These rollers were investigated in the focused DOE experiments (Section 6.1.2) that followed the initial factor screening study.

In addition to the effects of a small COF range being tested, the lower than expected significance of roller type is also an indicator of the considerable contribution of non-friction resistances to the overall unidirectional idler resistive force. This distinction between friction and non-friction effects is illustrated by the friction versus resistance coefficient comparison plot presented in Figure 5.4. This graph will be discussed in more detail in Section 6.3.

6.1.1.2 Insignificant Factors

Continuing with the discussion of single factor significance, it is also interesting to note which factors were deemed to be insignificant. These are the factors that can be found on the insignificance line of the half-normal plot. Inspecting Figure 4.6, it can be seen that the effects of **inclination angle** and **roller diameter** are inconsequential when it comes to belt slip. The declaration that inclination angle has no effect on belt slip doesn't seem to make sense. That's because this is not strictly speaking correct. Inclination angle definitely plays a role when it comes to unidirectional idler performance; this has already been shown by the development of the various inclined belt mechanical models (Sections 3.2 and 5.2.3). However, the reason that inclination angle doesn't influence the experimentally measured slip forces is because the test rig was designed to force belt slip.

Many of the experiments conducted on the UDR Test Rig were set up with operating conditions that would never otherwise have experienced slip (such as the horizontal belt case). Despite this, slip was achieved every time by increasing the pulling force until the belt began to move. The belt pulling action did the work that gravity would have done under the right operating conditions - with inclination angle playing the most important role (see Equation 5.21). With this in mind, the belt pulling action of the rig actually simulated the gravitational pull that would have been experienced at the critical inclination angle. Therefore, since inclination angle was already being simulated by the belt pulling action, physically changing the inclination of the rig had no effect on the measured slip force.

Besides inclination angle, idler **roller diameter** was also identified as insignificant. However, this was only true of roller diameter acting alone: when paired with roller type, the interaction effect (denoted by CD on the half-normal plot) was seen to be significant. A possible explanation for this is the slight pressure dependence that the friction coefficients of roller materials demonstrated in the additional friction study (see Figure 4.29). In terms of the insignificance of roller diameter acting alone, the expected effect that diameter related indentation resistance had on belt slip was either an insignificant contributor or the variation range between 127 and 152 mm diameter rollers was too small for any meaningful impact to be realised.

6.1.1.3 Main Effects

Moving on from the half-normal plot, the main effects of the various test factors may be examined. Recall that in design of experiments, the main effect of any factor is the average amount that the measured output (slip force in this case) will increase or decrease by when varying the factor from a low to high level. Numerically, the main effect of a factor is the difference between the average measured output at high and low levels of the factor (see Equation 2.3). The magnitude of this calculated number demonstrates the relative significance of the associated factor, while the sign indicates whether the effect increases or decreases when increasing the factor level (correlating to a positive and negative sign respectively). With this in mind, refer to the main effects plot presented in Figure 4.7.

This Figure shows the variation range of slip force about the overall mean (325 N). Examining the direction of the various effects it may be seen that diameter and inclination angle (the two factors with the smallest main effect) had a negative effect on the slip force. The remaining factors demonstrated positive effects; meaning that the belt slip force will increase with them. Further confirmation of the relative factor significance hierarchy was given by the difference in low-to-high level slip force range (i.e. main effect magnitude) and accompanying line gradient of the various factors. Again, idler pitch, which had the steepest main effect line and highest

value ($Ma = 182$), was evidently the most significant factor. Practically, if a belt conveying factor has a main effect value of 182 it means that if the factor is changed from a low to a high level (with all other factor levels remaining the same) then the resulting slip force will be 182 N greater, on average, than the original slip force.

With reference to the main effects plot (Figure 4.7), the significance of **belt tension** will now be examined. Observing belt tension to have the third most considerable effect on belt slip, it was initially thought that the reason for this had something to do with belt sag. Increasing belt tension is known to reduce belt sag (see Equation 2.2), and since lower belt sag diminishes the non-friction element of slip resistance, it follows that an increase in belt tension should result in a decrease in measured belt slip. However, if this were the case, the main effect of tension should have had a negative value. Examining the main effects plot it may be seen that tension demonstrated a positive influence on belt slip.

Not only was the tension main effect positive, but the belt sag measured throughout the experimentation hardly fluctuated (even with varying belt tensions) and was constantly at a low sag state of about 0.5%. The low, unwavering belt sag observed was likely due to the fact that the belt was consistently empty during the course of factor screening experimentation. With a belt that was always tensioned to at least some degree^c and without the presence of any conveyed material, there was nothing to weigh the belt down. Knowing that belt sag did not vary throughout testing, it was unexpected to see belt tension demonstrating any significance at all.

Initially it was considered that the sizeable positive main effect of tension was possibly as a result of idler set normal force increasing with tension. If stretching the belt resulted in it being pulled down onto the idler sets with greater force, then the associated normal loads (and therefore frictional resistance) would increase. However, if the successive idlers were correctly aligned (which they were confirmed to be) then there should not have been any additional loading imparted onto the rollers. Considering the change in belt surface properties related to varying the tension, two other possible explanations were contemplated. Firstly, if the change in rubber properties caused by a change in tension affect the COF between the belt and the rollers, then slip force will be dependant on belt tension regardless of sag conditions. Secondly, tension may affect belt indentation and associated indentation resistance, thus contributing to slip resistance (by way of a non-friction resistance). However, these possibilities were not tested in this experimental programme. It is therefore suggested that future research into unidirectional idlers tries to shed light on this oddity.

^cThe various UDR experimental programmes did not explore the absolute zero belt tension case. This was because of the operational difficulty associated with the required set-up. However, the conducted experiments analysed worse belt tension cases than in the case of a real belt snap - one characterised by excessive sagging of the belt between idler sets.

6.1.1.4 Interaction Effects

With the main factor effects examined, attention may now be given to the factor interactions that had a meaningful impact on belt slip. As previously mentioned, the significant interaction pairs were **pitch-trough** and **roller type-diameter**. Figure 4.8a shows the relationship between idler pitch and troughing angle in terms of the average measured resistive force at belt slip. From this figure, it may be seen that the effects of idler pitch were more significant at higher levels of troughing angle (indicated by the steeper gradient of the β^+ line). This finding makes sense when considering how trough and pitch affect normal loading and belt sag.

In the case of a flat belt (β^-), increasing the idler pitch from a low to a high level results in an increased kinetic slip force because of its relationship with normal force and belt sag. When, however, the belt is troughed (β^+), not only is the mean slip force greater than the flat belt case (indicated by the β^+ line being above the β^- line), but the effect that an increase in pitch has on the measured slip force is further amplified by the three dimensional sag (associated with a troughed belt) which is sensitive to the distance between successive idler sets. As an example, consider how an increase in idler pitch affects a flat conveyor belt. In this case, there is no increase in wing roller sag (i.e. three dimensional sag) with idler pitch, and therefore the change in slip force experienced will be lower than the troughed belt case where three dimensional sag effects are felt.

Moving onto the **roller type-diameter interaction** effect plot presented in Figure 4.8b, it may be seen that the effect of roller type was greater when larger diameter rollers were used. In the case of a Nylon roller (RT^-), increasing the roller diameter (from d^- to d^+), and therefore lowering contact pressure, resulted in a lower slip force. This makes sense when considering the relationship observed between contact pressure and COF for a Nylon roller - which was seen in Figure 4.28 to increase with contact pressure. When, however, the steel roller (RT^+) was considered, the difference in slip force related to a small versus a large roller diameter was very slight, with the larger roller diameter (d^+) experiencing the slightly higher slip forces on average. Based on this finding, it was expected to see a slight decrease in steel COF with increasing pressure (almost horizontal but slightly negative gradient COF line.) However, Figure 4.28 shows that this was not the case.

Instead, the Nylon vs steel COF comparison plot showed the COF line of a rough steel sample to have a positive gradient (i.e COF increases with contact pressure) that was much the same as that of the Nylon COF line. This may indicate that the rough steel sample used in the additional friction study was not a true representation of the painted steel rollers installed on the UDR Test Rig. It is important to note, that even if this was the case, it does not

discredit the additional friction study. This ancillary investigation was intended as a short look into the effects of belt and roller type on the friction force experienced during kinetic slip.

Now that all significant factors have been identified and their implications discussed, the core findings of the factor screening experimental programme may be summarised. The overall trends of the screening experiments are illustrated by the slip force bar chart presented in Figure 4.9. From this graph it may be seen that an increase in any of the three most significant belt conveying factors (idler pitch, troughing angle and belt tension), resulted in an increase in the resistive force developed at a locked idler set. For any particular combination of these three factors, the variation in inclination angle, roller diameter and roller material construction had little influence on the measured slip force (particularly when there was no belt troughing). This figure also indicates which combinations of significant factors had the most and least desirable impact in terms of unidirectional idler braking performance. With reference to Figure 4.9 it may be seen that the worst inclined belt case (i.e. the one resulting in the lowest resistive force at belt slip) was that where the idler pitch was 1200 mm, there was no troughing of the belt and the system tension was low. The best case scenario (i.e. the one requiring the highest pulling force to initiate belt slip) was experienced at the opposite end of each factor level when the idler sets were separated by a 1800 mm pitch, the belt was troughed to 45° and the system tension was high.

6.1.2 Focused DOE Tests

Following on from the initial factor screening set of experiments, additional tests were conducted so as to gain further insights into the performance of unidirectional idlers, specifically concerning the belt conveying factors that were deemed to have the greatest impact on the resistive braking mechanism. In addition to this, the larger data set that resulted from further experimentation could also be used in the development of the unidirectional idler proportion specification model (refer to Chapter 5).

6.1.2.1 Roller Type and Inclination Angle Effects

The first focused set of tests (**Experimental Programme B**) was designed to gain further insight into the effects of **roller material construction**. For the first time since the start of running experiments, this programme introduced rollers with an HDPE shell. The rationale behind conducting this programme came from the unexpectedly low significance of roller material construction observed during factor screening tests. These initial experiments compared steel and Nylon rollers only, and the resulting friction forces measured for both

roller types varied only slightly. As a result, the analysis of variance conducted for these early experiments showed roller type to have a comparatively small effect on unidirectional braking. However, the results of Experimental Programme B (presented in Figure 4.10) revealed this not to be the case.

This figure summarises the kinetic slip forces of steel, Nylon and HDPE rollers measured under various belt tension states. When comparing steel and Nylon rollers to HDPE ones, the effect of a markedly lower coefficient of friction (i.e. that of HDPE) may be appreciated. This is characterised by the considerably lower belt pulling force required to induce slip at the surface of an HDPE roller as compared to that of a steel or Nylon roller. This observation revealed HDPE rollers to constitute a worst case scenario when it comes to the friction braking component of a unidirectional idler. Based on this finding, further experiments were run using HDPE rollers, a supplementary friction study was conducted and HDPE rollers were considered in the derivation of the unidirectional idler proportion specification model.

The second set of focused tests (**Experimental Programme C**) was conducted in order to verify the earlier finding that varying the **inclination angle** of the UDR Test Rig has little impact on the measured belt slip force. This short experimental investigation varied the inclination of the test rig through four different angles, while keeping all other factors constant, and measured the associated resistance to belt slip. The results of these experiments, summarised in Figure 4.12, confirmed the conclusions drawn from the factor screening experiments – that the inclination angle of the test rig is of no consequence to the measured braking limit. It should, however, be reiterated that this does not mean that inclination angle does not influence the quantity of unidirectional idlers required to keep a snapped belt from accelerating down the incline. Remember that the belt pulling action of the test rig ensured that belt slip occurred every time and actually simulated the gravitational pull that would have been experienced at the critical inclination angle.

6.1.2.2 Roller Smoothing and Material Load Effects

Experimental Programme D was conducted so as to determine the effects of **roller smoothing** and **material load** on the performance of unidirectional idlers. Up to this point, all experiments had been conducted using an empty conveyor belt (for sake of ease) and roller surface wear had not been considered. These two factors were introduced so as to further investigate the effects of the previously determined significant factors (idler pitch, troughing angle and belt tension) when combined with more realistic operating conditions. This is because in reality, the conveyor belt would be loaded with material and the roller shells would quickly be worn down to a smooth surface finish. In the case of painted steel rollers, this equates to the loss of the layer of paint over time, resulting in a now exposed

and smoother “paintless” steel roller. As such, this experimental programme compared the surfaces of newly painted and “paintless” steel rollers.

The corresponding results of this experimental programme are summarised by the half normal plot presented in Figure 4.13. This plot reveals material load to have by far the most significant influence on unidirectional idler performance. This is because the introduction of material load onto the belt greatly affects the magnitude of supporting normal loads at each of the rollers, thus increasing the frictional braking potential of these devices as a result of increased surface friction, as well as increased indentation of the belt on the idler. In addition to this, a loaded conveyor belt experiences significantly more sag than an empty one. Consequently, the effort required to pull the loaded belt up and over each idler set is greater when there is material on the belt.

Similarly to the analysis of variance conducted using the factoring screening observations (see Figure 4.6), the half normal plot from this programme (Figure 4.13) shows the points corresponding to idler pitch and troughing angle to sit notably far away from the insignificance line, thus reiterating the significance of these factors. What is, however, interesting to note about these factors is that the order of their significance differs between the two studies (Experimental Programmes A and D). In this case, troughing angle demonstrates greater significance than idler pitch. This is because of the strong interaction between belt load and trough (as indicated by the position of this two-factor interaction point relative to the insignificance line).

This is better understood with reference to the corresponding interaction plot presented in Figure 4.15a. In this figure, the notable difference in effect line gradient emphasises the strength of the interaction between load and trough. The β^+ and β^- lines show that adding material to an empty belt has a greater impact on the belt slip force when troughed belts (β^+) are used as opposed to flat belts (β^-). This is interpreted from the steeper β^+ line and may be explained by considering how a fully loaded troughed belt would demonstrate a much greater resistance than that of a fully loaded flat belt, due to the higher slip impeding belt sag and friction that results from the heavier loading condition.

Keeping the focus on interaction effects, the relationship between material load and idler pitch (which was also deemed to have a significant effect on braking performance) may be explained. The effect plot of this two-factor interaction (presented in Figure 4.15b) shows that the effect that material load has on the critical belt slip force is greater when larger idler pitches are used (as indicated by the steeper a^+ effect line). This makes sense when considering the formula for belt sag (Equation 2.2), which proves that the increase in slip-restricting belt sag that results from adding material load (W) to a belt is further intensified when the horizontal distances between successive idler sets (a) is larger.

Referring back to the half normal plot derived from this experimental programme (Figure 4.13), it is also interesting to note which belt conveying factors were found to be insignificant. The half normal plot shows that the main effects of roller type^d and belt tension are inconsequential. With regards to the roller type factor, this finding shows that the loss of the paint layer and the smoothing of the shell surface that results from the wear subjected to idlers during continuous operations has no meaningful effect on the performance of a unidirectional idler during braking. However, this is not to say that the state of the roller surface does not have some effect on braking performance. Figures 4.16 and 4.17 reveal that a smoother steel roller without any paint on its surface almost always results in a lower critical slip point. This is particularly appreciable in Figure 4.16, which summarises the slip forces recorded in the unloaded belt case. When material load is introduced into the system, Figure 4.17 reveals surface finish to have some, but ultimately very little, impact on measured slip force (hence the aforementioned insignificance of this factor).

It is also interesting to note that, similarly to roller surface finish, the effects of **belt tension** are pronounced when the belt is empty (Figure 4.16) but almost totally void when there is material load on the belt (Figure 4.17). This graphically reiterates the findings of the analysis of variance, which categorised belt tension as insignificant. Ultimately, these observations prove that although tension effects are felt when the conveyor belt is empty, when material load is considered, the specific level of belt tension (within the range tested of 500 to 2500 kgf) is of no meaningful consequence. Despite this, it is still interesting to note the effect that belt tension has on sag under certain operating conditions.

With reference to Figure 4.17, it may be seen that the greatest force required to induce belt slip occurs when the belt is loaded, the successive idlers are separated by a pitch of 1800 mm and the belt is troughed to an angle of 45°. This is because, in addition to the development of higher normal forces at the idler sets promoting greater surface friction, there is a greater degree of conventional and three dimensional sag. The greater sagging of the belt results from the larger distances between successive idlers, the steeper troughed profile that the belt is required to bend into as it passes through the idlers and the greater carrying capacity of the belt (due to large troughing angle). It would thus be expected that in this case, an increase in belt tension would alleviate some of this excessive sag, and the resulting pulling force required to induce belt slip would be less. This is confirmed by the observations of Experimental Programme D (Figure 4.17) which determined the critical slip force of the aforementioned operating conditions to decrease with increasing belt tension (from 4.7 kN to 4.3 kN when increasing tension from 500 kgf to 2500 kgf, for example).

^dRecall that in this experimental programme, roller type was varied about two levels: painted steel rollers and steel rollers whose paint layers had been polished off.

6.1.2.3 Focused Analysis of Significant Belt Conveying Factors

The next set of experiments to be conducted (**Experimental Programme E**) expanded on the main findings of the original factor screening investigation. This was achieved by running focused experiments that varied the three most significant factors (as identified through Experimental Programme A) by **three levels** instead of the original two. This meant that medium factor levels of idler pitch, troughing angle and belt tension were introduced for experimentation ($a = 1500$ mm, $\beta = 35^\circ$ and $T = 1500$ kgf). The results of this focused study are summarised by the three-dimensional dot plot presented in Figure 4.18. This plot shows belt tension to have a noticeably less significant effect on slip force than idler pitch or troughing angle, which confirms the previously determined order of significance. As a result of this comparatively lower significance, as well as the findings from Experimental Programme D which revealed belt tension to be of no consequence when material load is introduced, the effects of belt tension in this particular investigation could be disregarded.

Instead, the focus was dedicated to idler pitch and troughing angle, and measured slip forces were averaged over the three tension levels. These averages, which are summarised in Table 4.2, were then used to plot a kinetic slip force surface (Figure 4.19). This figure illustrates the slip force trends related to idler pitch and troughing angle in an easy to visualise format. From this graph it may be seen that an increase in idler pitch results in an increase in critical slip force. This trend was expected and has already been previously explained by the increase in idler set normal load and belt sag that results from larger idler pitches. In addition to this, however, it is interesting to note that in the case of the 0° and 35° troughed belts, increasing the idler pitch by equal amounts (resolution of 300 mm) results in an almost perfectly linear increase in the resulting slip force.

However, in the case of the 45° troughed belt, the increasing slip force trend with idler pitch is not seen to be linear, but instead increases noticeably more when moving from idler pitches of 1500 mm to 1800 mm than that when moving from 1200 mm to 1500 mm (as indicated by the relative line gradients). This is most likely because when the conveyor is troughed to such a steep angle ($\beta = 45^\circ$), the increase in three-dimensional belt sag^e that results from increasing the idler pitch is more pronounced than in cases with smaller troughing angles, and more sensitive to larger troughing angles.

In other words, the effects of three-dimensional sag are not likely prominent at the lower end of the troughing angle scale and the increase in slip force experienced when $\beta = 0^\circ$ or 35° is therefore the result of friction, conventional belt sag and other non-friction factors (but not

^eRecall that the three-dimensional sag between successive idler sets is that which propagates from the wing rollers of a troughed belt as a result of the belt trough opening up and flattening out slightly between adjacent idler sets.

three-dimensional sag). In the case of the 45° troughed belt, however, three-dimensional sag effects are appreciable and since this type of sag does not scale linearly with increasing idler pitch, the resulting slip force profile is seen to be more sensitive to larger idler pitches. This finding is confirmed by the pitch-trough interaction plot derived from the factor screening experimental results (Figure 4.8a), which shows the effects of idler pitch to be more significant at higher levels of troughing angle.

Considering the change in slip force that results from increasing the troughing angle, it may be seen from Figure 4.19 that slip force increases consistently when moving from a flat belt to one with a troughing angle of 35°. This reiterates the findings of the factor screening programme, which showed that an increase in troughing angle results in improved unidirectional braking due to the increased normal loads (and therefore friction forces) at the wing rollers and the greater degree of three-dimensional sag. This trend is also observed when the idler pitch is 1800 mm and the troughing angle is increased from 35° to 45°. However, when idler pitches of 1200 mm and 1500 mm are considered, Figure 4.19 shows the levels of slip force to *decrease* when moving from a troughing angle of 35° to one of 45°.

Although this observation opposes the previously described trend, this decrease in slip force may be explained by considering how idler pitch and belt trough interact with each other. Recall from the factor screening experiments that the most significant two-factor interaction was that between idler pitch and troughing angle. When the idler pitch is small, not only are the effects of belt friction lower, but the non-friction effect of belt sag (including three-dimensional sag) is lower. When the belt is troughed to 45° and the idler pitch is relatively small, not only is there lower three-dimensional sag due to the short distances between idler sets, but there is also likely to be less conventional sag as a result of the greater degree of longitudinal belt stiffness gained through the curved profile of a troughed belt.

Figure 4.19 also illustrates the best and worst empty belt set up scenarios in terms of resulting belt slip^f. This figure reveals the best case to be when idler pitch and troughing angle are at their highest (1800 mm and 45° respectively), and the worst case to be when the idler pitch is 1200 mm and the belt troughing angle is 45°.

The next investigation conducted using the UDR Test Rig (**Experimental Programme F**) was used to study the effects of **belt loading**, incorporated into a focused study using the three most significant factors (as identified during the initial factor screening process). The observations of this investigation are summarised by the column chart presented in Figure 4.20. The observations gathered in this testing procedure did not generate any new

^fIt is important to reiterate that in the context of this research; a low slip force is not desirable as it represents a situation of potential runaway in the event of a belt snap. It is therefore preferable to maximise slip forces when possible.

understanding of unidirectional idler performance, but rather reiterated some of the findings already presented in this chapter. Similarly to previous findings, Figure 4.20 shows material load to be the most significant factor by far, idler pitch and troughing angle to have the second or third highest significance level (depending on whether the belt is loaded or not) and belt tension to have the least effect. This figure also reiterates that when the belt is empty, idler pitch has a greater effect on unidirectional performance than troughing angle and an increase in belt tension results in an increase in slip force, but when the belt is loaded the opposite is true. The explanation of these trends has already been presented and will therefore not be addressed again here.

Following this, a number of additional mini-investigations were conducted in order to gain further insight into the effects of roller material construction, the interaction between inclination angle and belt load, and the previously understudied medium troughing angle level of 35° . In addition to the increased functional understanding gained from running these focused experiments, the collection of more slip force observations also allowed for a larger data set to be established and then drawn from when deriving the unidirectional idler proportion specification model.

6.1.2.4 Mini-Investigation Results

The first of the mini-investigations, **Experimental Programme G**, reiterated the **roller type** related findings of Experimental Programmes B and D: that the difference in slip force of a painted steel roller compared to a “paintless” one is inconsequential, and that HDPE rollers result in noticeably lower slip forces than steel ones. What was, however, interesting to note in this particular investigation was just how much more pronounced the difference in slip force was between steel and HDPE rollers. This may be seen by the associated column chart (presented in Figure 4.21) which shows HDPE roller slip force to be 34% that of steel rollers on average when the belt was loaded (versus 44% when the belt was empty).

This phenomenon may be explained by considering the effect that contact pressure has on friction coefficient. The observations of an additional friction study (which will be discussed in greater depth later on) found that the friction coefficient of the specific type of conveyor belt used for all full scale experiments (a Nitrile rubber covered three-ply fabric belt) exhibits a positive pressure dependence. That is to say that the friction coefficients of the various roller-belt material pairs increase with increasing contact pressure. Figure 4.29 demonstrates this positive pressure dependence. With reference to this figure, it may also be noted that although the friction coefficients of steel and HDPE both increase with contact pressure, steel COF does so at a greater rate (as indicated by the steeper regression line gradient). This means that when the belt is loaded with material, the resulting increase in contact pressure

at the idlers results in an increase in friction coefficient that is greater for steel rollers than HDPE ones, and the different coefficients thus diverge even further from one another. This explains the more noticeable difference in slip force when comparing steel and HDPE rollers supporting a loaded belt (as previously highlighted in Figure 4.21).

The final two mini-investigations conducted, **Experimental Programmes H and I**, revisited the factor of **inclination angle**. These studies differed from Experimental Programme C in their incorporation of material load and lower friction generating HDPE idler rollers. The results from these two investigations are presented in Figures 4.22 and 4.23. These figures show that, unlike the results of Experimental Programme C, there are some instances where the critical slip force of the conveyor belt decreases with increasing inclination angle (see Figures 4.22b and 4.23a). However, the change in critical slip force that resulted from increasing the inclination angle by 5° was only 4% lower on average, and is therefore still negligible in the greater scheme of things.

6.1.2.5 Overall Observations

After all the experimental programmes had been conducted, further analysis could be realised with reference to the overall results. One such analysis that was considered was the effect that the cumulative belt and roller wear had on the measured slip forces. This was inspected due to the experimental observation that the test rig belt and roller surfaces became smoother after repeated use. Two graphs were generated to illustrate the effect of surface wear on the experiments conducted (Figures 4.24 and 4.25).

The first of these graphs, presented in Figure 4.24, focuses on six tests from Experimental Programme E (an investigation that was conducted long after Programme A). Two of these tests (namely E3 and E7) were not run concurrently with Programme E because the same experimental set-up had already been examined in Programme A. For this reason, the data obtained from the equivalent experiments in the factor screening trial were substituted into Programme E. Figure 4.24 compares these two substituted tests with commonly set up experiments (but at different belt tension levels) from Programme E. This figure shows the slip forces of Programme E experiments with equivalent factor screening counterparts (E3 and E7) to be uncharacteristically higher than the general trend of increasing slip force with increasing belt tension that was expected (according to previous observations). Since experiments A9 and A11 were conducted much earlier on than Experimental Programme E (and before belt loading had been introduced, for that matter), the discrepancy between anticipated and actual slip forces for the substituted experiments is indicative of a change in slip resistant properties with consistent belt wear.

The second graph addressing the consequence of surface wear is presented in Figure 4.25. This graph compares the slip forces of four repeated experiments. These experiments were repeated (instead of substituting the original result in place of the later occurring equivalent test) after it was discovered that the surface wear over repeated use of the equipment was a potential source of influence. This figure confirms the speculation made in reference to Figure 4.24 - that belt wear over time affects the critical slip force of a conveyor belt sliding over locked unidirectional idlers. From this column chart it may be seen that the repeated experiments demonstrated consistently (and considerably) lower slip forces than their original duplicates.

6.1.2.6 Summary

Now that all the focused DOE tests have been discussed, the core findings of the various experimental programmes may be summarised. The first of the focused studies (Experimental Programme B) found that although the small difference in coefficient of friction between steel and Nylon rollers has a comparably low impact on kinetic slip force, HDPE rollers have a markedly lower coefficient of friction of these two material types, and the use of HDPE rollers in their place does have a significant effect on unidirectional idler performance. In this case, the effect is a negative one, since HDPE rollers required the smallest pulling force to induce belt slip and thus constitute a worst case roller material when it comes to braking functionality.

The next investigation conducted (Experimental Programme C) confirmed the conclusions of the original factor screening tests - that, because belt slip is “forced” by the belt pulling action of the test apparatus, the inclination angle of the test rig has no meaningful effect on the measured force at slip. In addition to this programme, two other inclination angle studies were conducted (Experimental Programmes H and I), this time with the inclusion of belt load and HDPE rollers. The results from these tests didn’t quite conform to that of Experimental Programmes A and C, but rather did show some sensitivity to inclination angle (decreasing slip force with increasing inclination). However, the changes in observed slip force with inclination angle were very slight, and ultimately deemed insignificant. These findings thus supported the decision to run all other experimental programmes with the test rig in the horizontal position (thus reducing the number of required experiments to be run and decreasing the average time taken to run a single experiment).

Experimental Programme D introduced material load and smoothed steel rollers for the first time. From this study it was determined that material load has by far the greatest influence on unidirectional idler performance, whilst roller smoothing due to wear is inconsequential (especially when the belt is loaded). This study also reaffirmed the significance of idler pitch

and troughing angle, whose order of significance changes when material load is introduced (with troughing angle becoming more influential). It was also determined that the interaction pairs of trough-load and pitch-load to be strongly significant, with both idler pitch and troughing angle demonstrating a greater impact on measured slip force when the belt is loaded as opposed to empty. Finally, this experimental programme also revealed belt tension to have no meaningful effect on unidirectional braking when material load is introduced.

Following this, Experimental Programme E introduced medium levels of key belt conveying factors to the mix. From this study it was determined that when the troughing angle is relatively low (between 0° and 35°), increasing idler pitch results in a linear increase in measured slip force. When, however, the troughing angle is high (45°), the increase in three-dimensional sag that results from increasing the idler pitch is more pronounced and the rate at which slip force increases with idler pitch therefore also grows with increasing pitch (i.e. non-linear increase). In addition to this, Programme E found that when the troughing angle is large (45°) and idler pitch is small (1200 mm), not only is there less three-dimensional sag, but there is also less conventional sag as a result of the greater longitudinal belt stiffness provided by the steeply troughed belt profile. This explains why the lowest observed slip force in this investigation was obtained from a belt with a troughing angle of 45° supported by idler sets with a pitch of 1200 mm.

Finally, Experimental Programmes F and G reiterated the order of significance of material load, idler pitch and troughing angle for both loaded and unloaded belt cases, and also reaffirmed that HDPE rollers constitute a worst case scenario in terms of roller type and that the surface wear of steel idlers is of no meaningful consequence. Besides these repeated findings, Experimental Programme G did also provide an additional insight. From this study it was shown that the difference in slip force experienced by an HDPE roller versus a steel one is more pronounced when the conveyor belt is loaded (because of the difference in contact pressure dependence experienced by these two materials).

With reference to all the observations made throughout the various experimental programmes, it was also discovered that the continued conveyor belt wear over time directly affects the braking performance of unidirectional idlers, which is seen to deteriorate with increasing surface wear (i.e. lower resistive forces at the instance of belt slip). In addition to this, an assessment of the entire body of data found the worst unidirectional braking scenario (i.e. lowest observed slip force) to occur when an empty conveyor belt, troughed to 45° , is supported by idler sets containing HDPE rollers separated by a pitch of 1200 mm.

6.2 Additional Friction Study

The additional friction experiments that were conducted in the TBSA Laboratory, sought to determine the friction properties of relevant roller and conveyor belt material pairs. The observations made during the course of this study allowed friction force components to be isolated from the overall resistance forces measured using the UDR Test Rig (Experimental Programmes A to J). The decoupling of friction and non-friction forces, facilitated by this additional friction study, ultimately allowed non-friction coefficients^g to be derived from measured slip forces, thus enabling empirically derived trends to be extrapolated to the generic case. This process is summarised by the schematic diagram presented in Figure 5.10.

This extrapolation was realised by means of the numerically derived unidirectional idler proportion specification model (refer to Chapter 5), which enabled the collected resistive force data to be used in the design of unidirectional systems with different conveyor belt or roller types to those tested on the UDR Test Rig. A focused discussion of this proportion specifying model is presented in Section 6.3.

Returning to the friction study discussion, it may be recalled that the combinations of four roller materials and four belt samples (see Tables 3.1 and 3.2) were investigated in a total of 25 experiments. The first two of these experiments measured the friction coefficients of steel and Nylon with a conveyor belt sample that was cut from the same belt installed on the UDR Test Rig (a Nitrile rubber covered three-ply fabric belt). From the results of these experiments, the difference in COF for the two different roller material types may be appreciated. With reference to Figure 4.28, it may be seen that the rough steel sample exhibited a notably higher COF than that of the Nylon sample. This coincides with measurements captured using the UDR Test Rig which consistently demonstrated lower slip forces for experiments using Nylon rollers as opposed to steel.

In addition to these experiments revealing the COF values of the two samples, they also demonstrated the pressure dependence of conveyor belt COF for the first time. This positive pressure dependence was observed for all roller material samples paired with the test rig Nitrile rubber belt sample (see Figure 4.29), and was used to explain the greater difference in slip force between steel and HDPE rollers in the case of a fully loaded conveyor belt as compared to that of an empty belt (see Section 6.1.2).

^gRecall that the conventional friction formula ($F_f = \mu N$) was adapted (see Equation 5.1) by replacing the coefficient of friction term (μ) with a newly defined resistance coefficient (ψ). This constant may be broken up into friction and non-friction coefficients (μ and λ respectively), which contribute to the overall slip resistance at the belt-idler interface.

6.2.1 Effects of Conveyor Belt Preparation on Coefficient of Friction

Another interesting observation that was made when investigating the test rig Nitrile rubber belt was the varied effects of belt preparation on COF measurements. Recall from Figure 4.29 that the rough steel sample demonstrated the highest average COF, with the smooth steel sample coming in second and the HDPE sample exhibiting the lowest COF. The order of these COF trends was expected. What wasn't expected, however, was the way in which belt sample preparation affected the measured friction. It was initially assumed that running experiments with a wet belt, for example, would induce lower COF regardless of roller material sample type. But the experiments actually showed a wet belt to exhibit higher COF properties in the case of the steel samples, and the same COF in the case of HDPE.

Additionally, cleaning the surface of the conveyor belt sample (which was expected to reduce surface friction) had a varied effect on COF. In the case of the rough steel material sample, the cleaned belt demonstrated a lower COF as anticipated. In the cases of the smooth steel and HDPE samples, however, cleaning the belt before testing actually resulted in greater observed friction. However confusing the effects of belt preparation on COF may be, Figure 4.29 does at least demonstrate the spread of COF values that may be expected when pairing a Nitrile rubber covered belt with one of the three tested roller material samples. With this in mind, it may be noted that the rough steel sample demonstrated the greatest COF spread, and HDPE the lowest.

The anomaly of belt preparation effects was also observed during the testing of the next belt sample in the series: the used Nitrile rubber covered, solid woven PVC belt. Similarly to the previous experiments, the results from these tests (Figure 4.30) revealed rough steel to have the highest average COF, followed by smooth steel and HDPE. It may also be noted that, much like the case of the test rig belt sample, the effects of belt preparation varied between roller material samples.

The unexpected belt sample preparation trends (or seemingly lack thereof) present a very clear opportunity for additional study in order to better understand the friction mechanism affecting unidirectional idlers. It is therefore recommended that future research into the performance of unidirectional idlers endeavours to explain the friction limits of relevant material pairs subjected to a range of expected operating conditions. To properly explore these friction limits, it is also recommended that a greater sample size of different conveyor belt construction types are tested. Although the additional friction study conducted within this research facilitated an improved understanding of the friction developed between a conveyor belt and an idler, it was never the intention of this mini-investigation to completely explain the complex mechanics of rubber friction under varied conditions. Instead, the study

achieved what it sought out to do: to approximate the coefficients of friction of the different material types of interest and to use these measurements to determine the typical proportion of unidirectional idler slip force that results from surface friction alone.

6.2.2 Identification of the Worst Friction Case Scenario

The last two belt samples to be examined were an unused Nitrile rubber covered solid woven PVC belt and a Type M rubber covered steel cord belt. The friction measurements of these belt samples (Figures 4.31 and 4.32) conform to the established COF order associated with sample roller material type. However, the friction coefficient magnitudes recorded for both these samples were notably greater than that of the previously tested samples. With this in mind, it may be concluded that the first two belt samples tested (the rig Nitrile and used Nitrile belts) constitute a worst case scenario when it comes to unidirectional idler braking performance. That is to say that less effort is required to overcome the friction limits of these two samples as compared with higher COF demonstrating PVC Nitrile and Type M Rubber belts. This is better appreciated with reference to the overall minimum coefficient of friction summary graph presented in Figure 4.33.

This figure reveals that the belt choice resulting in the highest friction coefficient is the unused PVC Nitrile belt, whilst the lowest COF results from the use of either the rig or used Nitrile rubber belts. Drilling down into the specific material combinations it may be seen that a rough steel roller (analogous to a newly painted steel roller) paired with the brand new PVC Nitrile belt constitutes a best case scenario in terms of friction braking ability, whilst an HDPE roller supporting either the rig or used Nitrile belts constitutes a worst case scenario. It is also interesting to note the total spread of inferred friction coefficients presented in this graph, with the lowest and highest overall COF values observed through experimentation being approximately 0.19 and 1.58 respectively.

In addition to this, Figure 4.33 reveals the COF magnitudes experienced when testing the rig Nitrile and used Nitrile belt samples to be very similar, with HDPE showing almost no difference at all between these two belt samples. This similarity is better observed in Figure 4.34 which more clearly demonstrates the close proximity and parallelism between the various belt-to-roller sample material pairs. Remembering that the rig and used Nitrile rubber belt samples demonstrated the lowest friction properties overall, it may be said that the specific conveyor belt installed on the UDR Test Rig is analogous to that of a Nitrile rubber covered PVC belt that has undergone appreciable surface wear after continual use in operation.

This conclusion is important because it means that the unidirectional idler proportion specifying technique which was derived from full scale experiments on a belt analogous to that

of a used Nitrile rubber covered PVC belt, is based on a worst case scenario when it comes to the friction properties of the roller-to-belt material type combination. This introduces an additional factor of safety to the calculation of the required proportion of the safety critical unidirectional idlers, and therefore means that the derived proportion specifying model is conservative.

The conservative friction properties of a used conveyor belt are better appreciated when comparing the COF results between the new and used PVC Nitrile rubber belt samples as presented in Figure 4.35. This figure illustrates the change in surface friction properties that may occur throughout the life of a conveyor belt. Both belt samples presented in this figure have the same construction. They both consist of a PVC impregnated solid woven carcass with Nitrile rubber covers. The only difference between these samples is that one of them was brand new, whilst the other was taken from a belt that was previously in operation. With this in mind, it may be concluded that the continued use of a conveyor belt in operation results in the degradation of the rubber surface which consequently leads to a decrease in the friction coefficient. This conclusion directly corresponds to the previous UDR Test Rig experimental finding that an increase in belt wear over time results in lower critical slip forces.

In addition to the stark difference in COF magnitude observed, Figure 4.35 also seems to indicate that increasing rubber wear over time results in a change in the pressure dependence of the belt, which is seen to migrate from a positive to a negative trend within the lifetime of the belt.

Using the COF findings discussed in this section, the resistive forces measured with the UDR Test Rig could be broken up into friction and non-friction components. This allowed the UDR Test Rig observed trends to be extrapolated to the generic case. In addition to this, the observed friction coefficients could be applied, using the numerical model presented in Section 6.3, to the unidirectional idler proportion calculation.

6.2.3 Summary

Before the results of this unidirectional idler proportion specifying model are discussed in the next section, it is worth summarising the key findings from the additional friction study. From the various experiments conducted using the Jenike Direct Shear Tester, the following was determined:

- HDPE and rough steel (correlating to a newly painted roller) demonstrate the lowest and highest average COF respectively, regardless of the conveyor belt type.

- Belt friction is slightly pressure dependent, with the majority of tests showing an increase in COF with increasing contact pressure.
- The best (high friction) and worst (low friction) belt-to-roller material pairs were determined to be a rough steel roller combined with a brand new PVC Nitrile rubber belt, and an HDPE roller paired either with the rig or used Nitrile rubber belts respectively.
- The friction coefficients observed throughout all conducted experiments ranged from approximately 0.19 to 1.58.
- The continuous wearing away of the conveyor belt rubber cover over time leads to a reduction in observed COF.

6.3 Numerical Model

To facilitate the use of the various empirical findings in practical belt conveyor design considerations, a numerical model was derived. This model, characterised by Equations 5.27 and 5.28, may be used by in industry to approximate the proportion of centre and wing rollers in a three-roll idler system that are required to be unidirectional, in order to inhibit belt runaway in the catastrophic event of a belt snap.

The final version of the unidirectional idler proportion specifying model (Model 2) assumed the resistance to belt slip to result from a combination of friction and non-friction effects. To test this assumption, resistance coefficients calculated from UDR Test Rig experiments were compared to the friction coefficients measured in the TBSA laboratory (see Figure 5.4). This comparison, which shows the pure friction coefficient (μ) to be lower than the empirically inferred resistance coefficient (ψ) for every test case, proves that the resistance to conveyor belt slip results from both friction and non-friction forces.

6.3.1 Non-Friction Coefficient Approximation

With all resistive and friction coefficients known for each measured belt slip force and knowing the relationship between these two factors (see Equation 5.2), the non-friction coefficients (λ) were calculated for each experiment (as summarised by the schematic diagram presented in Figure 5.5). These factors were then condensed into a list of the minimum recorded values corresponding to the varied levels of the three most significant belt conveying factors (belt load, idler pitch and troughing angle). The resulting non-friction coefficients (calculated for centre rollers, wing rollers and idler sets) are summarised in Tables 5.3, 5.4 and 5.5.

The non-friction coefficients listed in these tables were then used together with Equation 5.2 and the minimum observed coefficients of friction to predict the centre roller, wing roller and overall idler set resistance coefficients (ψ_c , ψ_w and ψ) for each experimental set-up. These predictions were then plotted together with the actual resistance factors, as presented in Figures 5.6, 5.7 and 5.8. These comparison plots illustrate the relationship between measured slip forces and the derived prediction model. In each of these plots, the prediction line is seen to intersect the minimum resistance coefficients corresponding to the standard operating condition ranges categorised in Tables 5.3, 5.4 and 5.5.

Knowing that lower coefficients of resistance are detrimental to the functionality of unidirectional idlers, Figures 5.6, 5.7 and 5.8 show that the prediction method used is rather conservative because it predicts coefficients that are never higher than actual values^h (especially if a safety factor is used). In summary, these figures show that if the Resistance Model (refer to Equations 5.27 and 5.28) is used to predict the resistance coefficient of a particular inclined belt operating scenario, the model will always err on the side of caution and use the worst observed case as the datum for predicting required proportions of unidirectional idlers.

These conservative non-friction coefficient values were used in both unidirectional idler proportion specifying methods (Models 1 and 2). The difference between the two models resulted from the order in which unidirectional idlers were prescribedⁱ. Recall from Section 5.2.3 that whereas Model 1 specified the required proportion of unidirectional rollers as a percentage of the total number of idler sets^j, Model 2 specified centre rollers as unidirectional first, before making use of any unidirectional wing rollers.

6.3.2 Proportion Specification Model Comparison

To determine which of the two unidirectional idler proportion specification methods would be most suitable for use in industry, Models 1 and 2 were compared against the Preliminary Model (derived in Section 3.2) as shown in Figure 5.9. This figure shows that the newly derived proportion specifying methods (Models 1 and 2) typically recommend unidirectional idler proportions that are less than that as calculated using the Preliminary Model. However,

^hIt is important to stress that the numerical model results presented in Section 5.3 are based on the experiments of a single conveyor belt type (a Nitrile rubber covered three-ply fabric belt), and three roller material types (steel, Nylon and HDPE). For a meaningful analysis of any other conveyor belt and roller combinations, the corresponding friction coefficients would need to be determined.

ⁱAlthough these models differ in their prescribed order of unidirectional rollers being specified (centre and wing rollers together versus centre rollers first), the pattern of this installation is not important, provided the correct quantities of centre and wing unidirectional rollers are installed.

^jIf, for example, Model 1 specifies a unidirectional idler proportion of $\eta_s = 50\%$ for an inclined belt with 100 three-roll idlers, then 50 of the idler sets need to be installed with unidirectional centre and wing rollers.

it may also be noted that under certain circumstances, the required proportion of unidirectional idlers as specified by Models 1 and 2 is greater than that of the Preliminary Model. This is specifically the case when considering the lower friction coefficient of HDPE rollers as compared with that of steel.

When considering the difference in required unidirectional roller proportions as prescribed by Models 1 and 2 it may be seen from Figure 5.9 that Model 2 specifies consistently lower proportions than Model 1. Bearing in mind that both models use the same friction and non-friction coefficients to calculate the resistive force limits of the system, it follows that the specification method used in Model 1 is wasteful. When the high cost of unidirectional idlers is considered, it may be concluded that unidirectional proportions are best specified using Model 2.

Although the study presented in Figure 5.9 assessed a wide range of inclined belt operating conditions, many of the scenarios considered are very unlikely to occur in reality. The two factors that this particularly applies to are troughing angle and roller type. It is uncommon to design inclined conveyor belts without at least some degree of troughing (i.e. flat belts are unexpected). It is also rare to find inclined belts installed with HDPE rollers; steel being the standard. With this in mind, the more common operating conditions were applied to the numerical simulation. The results of this condensed study, which only considered steel rollers and troughing angles larger than 30° , were shown in Figure 5.11.

This figure shows the required proportion of unidirectional steel rollers as specified by the Preliminary Model to depend on inclination angle alone and is not affected by the idler pitch, belt trough, belt width or material load. This is in accordance with the model equation presented in Section 3.2 (Equation 3.8). The proportion of these safety devices according to Model 2, on the other hand, is seen to change with changing levels of inclination angle as well as with idler pitch, troughing angle, belt width and material loading conditions.

With reference to Model 2, it is also interesting to note the spike in required unidirectional idler proportions that occurs when the conveyor belt is empty. This is because an empty belt does not experience as much resistance from belt sag or belt cover indentation as one that is loaded with material. In addition to this, the increase in COF experienced with increased contact pressure (see Section 4.2), means that loaded belts exhibit additional resistance to belt slip as a result of contact friction. One of the important conclusions that may be drawn from this observation is that an empty conveyor belt constitutes a worst-case scenario in terms of belt slip resistance. This finding was reiterated by Case Study 1 (see Section 5.3.2) which used the proportion specifying equations of Model 2 to determine the overall unidirectional idler ratios of an empty and fully loaded belt under the same operating conditions to be 42% and 26% respectively.

From this it follows that the specification of the required proportion of unidirectional idlers for an installation must be calculated for an empty belt, not a loaded one. If an inclined conveyor belt system has enough unidirectional idlers to prevent the runaway of an empty belt then it will also be able to hold the belt in place when loaded with bulk material.

Comparing the two models presented in Figure 5.11 it may also be seen that the model derived from the experimentally determined friction and non-friction factors (Model 2) almost always specifies lower proportions of unidirectional idlers than the Preliminary Model. In fact, only at an inclination angle of 18° does Model 2 begin to specify larger proportions for certain combinations of inclined belt parameters (when the belt is empty and the idler pitch and troughing angle are minimised).

Knowing Model 2 to already be conservative in its estimation of required unidirectional idler proportions, the comparison plot presented in Figure 5.11 shows the Preliminary Model to be somewhat wasteful – particularly in operating conditions that exhibit a high degree of conventional and three-dimensional belt sag (i.e. when the idler pitch and/or troughing angle is large). This is confirmed by the focused comparison plot presented in Figure 5.12, which presents the unidirectional idler proportions of an empty belt supported by steel idlers paired with specific levels of idler pitch and troughing angle.

From this figure it may be seen that when large troughing angles ($\beta = 45^\circ$) are paired with large idler pitches ($a \geq 1500$ mm), the Preliminary Model specifies proportionally larger and larger amounts of unidirectional idlers with increasing inclination angle than that of the derived model (Model 2). This is because the Preliminary Model, which only considered the effects of contact friction on braking performance, does not take into account any of the non-friction resistances that are so influential when empty belts are troughed to steep angles and idler spacings are large. Recall from Experimental Programme E that the best case scenario for unidirectional idler performance (i.e. the scenario that requires the largest downward belt pulling force to induce slip) was that with an idler pitch of 1800 mm and a troughing angle of 45° (as illustrated in Figure 4.19).

However, this over-specification of unidirectional idlers by the Preliminary Model only occurs when idler pitches and troughing angles are large. When smaller levels of these parameters are prescribed, Figure 5.12 shows the converse to be true. That is to say that as inclination angle increases, the relative difference in unidirectional idler proportion decreases between the two prediction techniques until such point as the models converge, and eventually cross over. In this way, it may be said that the Preliminary Model specifies larger proportions of unidirectional idlers than Model 2 for inclination angles up to, and including, 12° . For inclinations greater than this, Model 2 specifies larger proportions, and the steeper the inclination angle, the greater the disparity between the two calculation techniques. It may

also be observed that the greater the inclination angle, the greater the disparity between the two models, and therefore the more serious the cost and/or safety implications of assuming unidirectional idlers to resist belt slip through contact friction alone, instead of taking into account the effects of non-friction resistances as well.

6.3.3 Case Study Results

Following the derivation and validation of Model 2, three existing inclined belt installations were considered as case studies and the full unidirectional idler proportion calculations for each of these cases were carried out as shown in Section 5.3.2. From the exploration of these case studies, a specified unidirectional idler proportion range of 42% to 67% was calculated. These proportions corresponded to inclination angles 10.5° - 14.7° , idler pitches 1400 - 2250 mm, belt widths 750 - 1800 mm, and troughing angles 35° - 45° . All case studies made use of steel rollers.

From these results it may be said that for the practical range of belt conveying operating conditions that varied in the three case studies, the maximum proportion of unidirectional idlers required to inhibit belt runaway was just under 70% of the total number of rollers installed along the inclined belt section, and therefore unidirectional idlers are a viable option for all case studies considered. But when do unidirectional idlers become inadequate in the prevention of belt runaway? To answer this question, a hypothetical scenario was considered that flexed the various conveyor belt operating conditions to the limit of what unidirectional idlers can safely operate in (see Section 5.3.2 and Table 5.10).

This hypothetical case showed that all idlers would be required to be unidirectional for a 900 mm wide conveyor inclined to an angle of 16° , troughed to 35° and supported by HDPE rollers separated by an idler pitch of 1200 mm. In this case, increasing the inclination angle past 16° would result in unidirectional idlers becoming inadequate as a means to belt slip prevention. In reality, however, this is very unlikely to occur because HDPE rollers are very seldom prescribed in inclined systems and inclination angles greater than 16° are rare.

6.3.4 Condensed Unidirectional Idler Proportion Matrix

Finally, it was desired to condense all the findings into a compact unidirectional idler proportion matrix that may be used in industry as an initial approximate proportion specifying tool. This matrix was generated by using the Model 2 formulae (Equations 5.27 and 5.28) to calculate the proportion of centre and wing rollers required to be unidirectional at varying angles of inclination for a worst-case scenario. This being an empty conveyor belt troughed

at 45° and supported by idlers separated by a 1200 mm pitch. The combination of a 1200 mm idler pitch and a 45° troughing angle constitutes a worst case scenario because of the very low non-friction coefficient that results from this particular pairing (see Tables 5.4 and 5.5). The results of this exercise, which was conducted for both steel and HDPE rollers, and which used a safety factor of 1.15, were presented in Table 5.11. This table shows that it is unsafe to use steel unidirectional idlers for conveyor belts inclined at an angle greater than 21° and it is unsafe to use HDPE unidirectional idlers for inclines greater than 16° . Knowing that inclination angles greater than 18° are uncommon in practice, it is justified to assume steel unidirectional idlers to be suitable runaway preventing safety devices regardless of operating conditions. Unidirectional HDPE rollers, on the other hand, should be used with caution.

Chapter 7 Conclusions and Recommendations

7.1 Conclusions

An investigation into the performance of unidirectional idlers was conducted so as to determine the braking limits of these devices under various operating conditions. Full scale experiments were carried out, using the UDR Test Rig, to measure the resistive force acting at the contact between locked idlers and the conveyor belt at the instance of belt slip. A wide range of conveyor belt operating conditions were tested by varying the levels of inclination angle, troughing angle, idler pitch, roller type, roller diameter, belt tension and material loading. Following this, a mini-investigation into the friction properties of various roller and conveyor belt material pairs was conducted in the TUNRA Bulk Solids Africa Laboratory at the University of the Witwatersrand. This study quantified the friction coefficients of relevant material pairs under varied loading conditions. To supplement the results of these two investigations, a numerical model was derived, based on empirical findings, that may be used by industry to specify the required proportion of unidirectional rollers for any given inclined conveyor belt installation. The following sections present the conclusions drawn from each of the sections discussed in Chapter 6.

7.1.1 Factor Screening Experiments

The overall trends of the factor screening experiments (see Figure 4.9) demonstrated that an increase in any of the three most significant belt conveying factors (identified to be idler pitch, troughing angle and belt tension), led to an increase in the resistive force developed at a locked idler set. For any particular combination of these three factors, the variation in inclination angle, roller diameter and roller material construction had little influence on the measured slip force (particularly when there was no belt troughing). It was also observed that the worst inclined belt case (i.e. the one resulting in the lowest resistive force at belt slip) was that where the idler pitch was 1200 mm, there was no troughing of the belt and the system tension was low. The best case scenario (i.e. the one requiring the highest pulling

force to initiate belt slip) was experienced at the opposite end of each factor level when the idler sets were separated by a 1800 mm pitch, the belt was troughed to 45° and the system tension was high.

7.1.2 Focused DOE Tests

Experimental Programme B found the low friction coefficients of HDPE rollers to significantly affect the performance of unidirectional idlers and may be considered a worst case roller material scenario when it comes to braking functionality.

Experimental Programme C confirmed the conclusions of the factor screening programme – that the inclination angle of the test rig had no meaningful effect on the measured force at slip.

Experimental Programme D showed material load to have had by far the greatest influence on unidirectional idler performance, whilst roller smoothing due to wear was inconsequential. It was also determined that both idler pitch and troughing angle demonstrated a greater impact on measured slip force when the belt was loaded as opposed to empty. Finally, belt tension was revealed to have had no meaningful effect when material load was introduced.

Experimental Programme E proved that when the troughing angle is relatively low, increasing idler pitch results in a linear increase in measured slip force. When, however, the troughing angle is high, the increase in three-dimensional sag that results from increasing idler pitch is more pronounced and the rate at which slip force increases with idler pitch therefore intensifies. In addition to this, it was found that when large troughing angles are paired with small idler pitches, the greater longitudinal belt stiffness that results diminishes the typical level of conventional belt sag between successive idler sets.

Programmes F and G reiterated the order of significance of material load, idler pitch and troughing angle for both loaded and unloaded belt cases, and also reaffirmed that HDPE rollers constitute a worst case scenario in terms of roller type and that the surface wear of steel idlers was of no meaningful consequence. Besides these repeated findings, Experimental Programme G also found that the difference in slip force experienced by an HDPE roller versus a steel one is more pronounced when the conveyor belt is loaded.

It was also discovered that conveyor belt wear over time directly affects the braking performance of unidirectional idlers, which is seen to deteriorate with increasing surface wear. In addition to this, an assessment of all the experiments conducted with the UDR Test Rig found

the worst unidirectional braking scenario to occur when an empty conveyor belt, troughed to 45° , is supported by idler sets containing HDPE rollers separated by a pitch of 1200 mm.

7.1.3 Additional Friction Study

The results of the additional friction study showed that HDPE and rough steel (analogous to a newly painted roller) demonstrated the lowest and highest average friction coefficients respectively, regardless of the conveyor belt type. It was also found that belt friction was slightly pressure dependent, with the majority of tests showing an increase in COF with increasing contact pressure. The belt-to-roller material pairs that demonstrated the highest and lowest friction coefficients were determined to be a rough steel roller combined with a brand new PVC Nitrile rubber belt, and an HDPE roller paired either with the rig or used Nitrile rubber belts respectively. This study also showed that the continuous wearing away of the conveyor belt rubber cover over time led to a reduction in observed COF. These findings echoed those of the unidirectional roller experiments – that HDPE rollers provide the least resistance to belt slip and that belt wear over time results in the reduction of unidirectional idler braking capability.

7.1.4 Numerical Model

A conservative method for specifying the required proportion of unidirectional rollers to prevent belt runaway in the event of a belt snap was derived using empirical findings. This model, which separated the effects of friction and non-friction resistive forces, may be used to calculate the required proportion of unidirectional centre and wing rollers for belt conveyor systems with various levels of inclination angle, roller type, idler pitch, troughing angle and loading conditions. The final version of the proportion specification method selected (Model 2) prioritises the specification of the larger load supporting centre rollers as unidirectional first before assigning any unidirectional wing rollers, thus minimising the total number of these expensive devices^a needed to provide the required cumulative braking force. Refer to Appendix B for a step-by-step summary of the procedure to follow (including relevant equations) when calculating the required proportion of unidirectional rollers using Model 2.

Comparison studies found that using a purely friction based unidirectional idler proportion specification method (such as the Preliminary Model) in place of the empirically derived technique may lead to the over/under-specification of these devices, particularly when conveyor belts are inclined to angles greater than 15° . In instances where low levels of idler pitch and

^aThe relative sale prices provided by a local idler manufacturer showed that unidirectional rollers may cost between 2.1 and 2.9 times that of equivalent conventional rollers [7].

troughing angle are present, using pure friction based calculations will result in the wasteful over-specification of unidirectional idlers. Whilst in systems with high levels of idler pitch and belt trough, the assumption of pure friction braking may result in inadequate proportions of unidirectional idlers being specified.

By varying the levels of the relevant belt conveying parameters, Model 2 was used to determine the worst case unidirectional idler performance scenarios (i.e. the scenario requiring the largest proportion of unidirectional idlers to resist belt slip). This worst case was identified as a 900 mm wide empty conveyor belt, troughed to an angle of 35° and supported by HDPE idlers separated by a pitch of 1200 mm. Under these conditions, even if 100% of the HDPE rollers installed in the system are unidirectional, the prevention of belt runaway in the event of a belt snap is not possible if the belt inclination angle is greater than 16° .

When the same exercise was performed for steel rollers it was found that a maximum inclination angle of 21° was allowable under worst case conditions. Knowing that inclination angles greater than 18° are uncommon in practice, it may be concluded that steel unidirectional idlers are suitable runaway preventing devices regardless of operating conditions. Unidirectional HDPE rollers, on the other hand, should be used with caution.

7.2 Recommendations

The research presented in this dissertation tackled a mammoth task: to determine and quantify, for the first time, the effects that seven different belt conveying parameters have on the braking performance of unidirectional idlers. In order to make a start, statistical design of experiment techniques were used to strategically plan a realistic number of experiments (within the almost limitless set of conveyor belt factor level combinations) that could provide some insight into the performance of unidirectional idlers.

Given the extensive inclined conveyor belt configuration possibilities, many opportunities exist for continued research work. Based on the findings of this dissertation, the following list of future work is recommended:

- The research conducted showed the effects of belt tension to contradict expected outcomes when testing an empty conveyor belt. In this case an increase in belt tension (which was expected to reduce belt sag and therefore slip resistance) was seen to result in an increase in slip force. This anomaly may be explained through a focused investigation into the effects of tension on belt surface properties. It is anticipated different levels

of belt tension will affect the friction coefficient and indentation resistance properties of the rubber belt.

- Due to the physical constraints of the UDR Test Rig used, the conveyor belt installed on the rig was always tensioned to at least some degree. By redesigning the experimental apparatus to accommodate zero tension belt states, conveyor belt operating conditions that are more representative of those after a belt snap may be considered in future work.
- The additional friction study conducted during this research venture was able to approximate the friction coefficients of relevant material pairs. However, this was just a mini-investigation and as such there is a lot of potential for future research in this field. A more in-depth investigation into the friction properties of a wider variety of roller and conveyor belt material types subjected to a range of realistic operating states (such as high temperatures, dusty conditions or wet conveyor belts) would help to better explain the actual friction braking limits of unidirectional idlers.
- The experiments conducted using the UDR Test Rig considered a quasi-static system. One in which the conveyor belt being tested is assumed to have snapped, released its stored energy and come to rest just before it begins to accelerate down the incline (if the operating conditions of the system allow this). In reality, however, the event of a belt snap is a lot more dynamic, and at this stage, unpredictable. There is therefore the potential for future research to explore the transient behaviour of the conveyor belt at the instance of snap and its impact on unidirectional idler performance. This may be realised/supplemented through the development of a dynamic simulation.
- The total set of tests carried out through the course of this research venture varied a wide variety of conveyor belt parameters at multiple levels so as to develop as big a picture of unidirectional idler performance as possible. There were, however, a number of factors that weren't explored due to practical and time constraints. It is therefore recommended that future work endeavours to explain the effects that the following list of parameters has on unidirectional braking:
 - Conveyor belt width.
 - Conveyor belt material construction
 - Quantity of rollers per idler set (e.g. three-roll vs five-roll idlers)
 - Idler set layout type (e.g. in-line vs offset vs belt friendly)

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Appendix A Experimental Programmes

A.1 Unidirectional Roller Experiments

Table A.1: Factor screening experiments (Programme A)

Test	α (°)	Pitch (mm)	Roller	D (mm)	Trough (°)	Tension	Load
A1	0	1200	Steel	127	0	low	NO
A2	15	1200	Steel	127	0	high	NO
A3	0	1800	Steel	127	0	high	NO
A4	15	1800	Steel	127	0	low	NO
A5	0	1200	Nylon	127	0	high	NO
A6	15	1200	Nylon	127	0	low	NO
A7	0	1800	Nylon	127	0	low	NO
A8	15	1800	Nylon	127	0	high	NO
A9	0	1200	Steel	152	0	high	NO
A10	15	1200	Steel	152	0	low	NO
A11	0	1800	Steel	152	0	low	NO
A12	15	1800	Steel	152	0	high	NO
A13	0	1200	Nylon	152	0	low	NO
A14	15	1200	Nylon	152	0	high	NO
A15	0	1800	Nylon	152	0	high	NO
A16	15	1800	Nylon	152	0	low	NO
A17	0	1200	Steel	127	45	high	NO
A18	15	1200	Steel	127	45	low	NO
A19	0	1800	Steel	127	45	low	NO
A20	15	1800	Steel	127	45	high	NO
A21	0	1200	Nylon	127	45	low	NO
A22	15	1200	Nylon	127	45	high	NO
A23	0	1800	Nylon	127	45	high	NO
A24	15	1800	Nylon	127	45	low	NO

Test	α (°)	Pitch (mm)	Roller	D (mm)	Trough (°)	Tension	Load
A25	0	1200	Steel	152	45	low	NO
A26	15	1200	Steel	152	45	high	NO
A27	0	1800	Steel	152	45	high	NO
A28	15	1800	Steel	152	45	low	NO
A29	0	1200	Nylon	152	45	high	NO
A30	15	1200	Nylon	152	45	low	NO
A31	0	1800	Nylon	152	45	low	NO
A32	15	1800	Nylon	152	45	high	NO

Table A.2: Roller type and belt tension experiments (Programme B)

Test	α (°)	Pitch (mm)	Roller	D (mm)	Trough (°)	Tension	Load
B1	0	1200	Nylon	152	0	low	NO
B2	0	1200	Nylon	152	0	med	NO
B3	0	1200	Nylon	152	0	high	NO
B4	0	1200	HDPE	152	0	low	NO
B5	0	1200	HDPE	152	0	med	NO
B6	0	1200	HDPE	152	0	high	NO
B7	0	1200	Steel	152	0	low	NO
B8	0	1200	Steel	152	0	med	NO
B9	0	1200	Steel	152	0	high	NO

Table A.3: Effect of incline confirmation experiments (Programme C)

Test	α (°)	Pitch (mm)	Roller	D (mm)	Trough (°)	Tension	Load
C1	0	1800	Steel	152	0	high	NO
C2	5	1800	Steel	152	0	high	NO
C3	10	1800	Steel	152	0	high	NO
C4	15	1800	Steel	152	0	high	NO

Table A.4: Loaded steel rollers with and without paint (Programme D)

Test	α (°)	Pitch (mm)	Roller	D (mm)	Trough (°)	Tension	Load
D1	0	1200	Steel	152	0	low	NO
D2	0	1200	Steel	152	0	high	NO
D3	0	1800	Steel	152	0	low	NO
D4	0	1800	Steel	152	0	high	NO
D5	0	1200	Steel (NP)	152	0	low	NO
D6	0	1200	Steel (NP)	152	0	high	NO
D7	0	1800	Steel (NP)	152	0	low	NO
D8	0	1800	Steel (NP)	152	0	high	NO
D9	0	1200	Steel	152	0	low	YES
D10	0	1200	Steel	152	0	high	YES
D11	0	1800	Steel	152	0	low	YES
D12	0	1800	Steel	152	0	high	YES
D13	0	1200	Steel (NP)	152	0	low	YES
D14	0	1200	Steel (NP)	152	0	high	YES
D15	0	1800	Steel (NP)	152	0	low	YES
D16	0	1800	Steel (NP)	152	0	high	YES
D17	0	1200	Steel	152	45	low	NO
D18	0	1200	Steel	152	45	high	NO
D19	0	1800	Steel	152	45	low	NO
D20	0	1800	Steel	152	45	high	NO
D21	0	1200	Steel (NP)	152	45	low	NO
D22	0	1200	Steel (NP)	152	45	high	NO
D23	0	1800	Steel (NP)	152	45	low	NO
D24	0	1800	Steel (NP)	152	45	high	NO
D25	0	1200	Steel	152	45	low	YES
D26	0	1200	Steel	152	45	high	YES
D27	0	1800	Steel	152	45	low	YES
D28	0	1800	Steel	152	45	high	YES
D29	0	1200	Steel (NP)	152	45	low	YES
D30	0	1200	Steel (NP)	152	45	high	YES
D31	0	1800	Steel (NP)	152	45	low	YES
D32	0	1800	Steel (NP)	152	45	high	YES

Table A.5: Significant factor focused experiments (Programme E)

Test	α (°)	Pitch (mm)	Roller	D (mm)	Trough (°)	Tension	Load
E1	0	1200	Steel	152	0	low	NO
E2	0	1200	Steel	152	0	med	NO
E3	0	1200	Steel	152	0	high	NO
E4	0	1500	Steel	152	0	low	NO
E5	0	1500	Steel	152	0	med	NO
E6	0	1500	Steel	152	0	high	NO
E7	0	1800	Steel	152	0	low	NO
E8	0	1800	Steel	152	0	med	NO
E9	0	1800	Steel	152	0	high	NO
E10	0	1200	Steel	152	35	low	NO
E11	0	1200	Steel	152	35	med	NO
E12	0	1200	Steel	152	35	high	NO
E13	0	1500	Steel	152	35	low	NO
E14	0	1500	Steel	152	35	med	NO
E15	0	1500	Steel	152	35	high	NO
E16	0	1800	Steel	152	35	low	NO
E17	0	1800	Steel	152	35	med	NO
E18	0	1800	Steel	152	35	high	NO
E19	0	1200	Steel	152	45	low	NO
E20	0	1200	Steel	152	45	med	NO
E21	0	1200	Steel	152	45	high	NO
E22	0	1500	Steel	152	45	low	NO
E23	0	1500	Steel	152	45	med	NO
E24	0	1500	Steel	152	45	high	NO
E25	0	1800	Steel	152	45	low	NO
E26	0	1800	Steel	152	45	med	NO
E27	0	1800	Steel	152	45	high	NO

Table A.6: Loaded significant factor experiments (Programme F)

Test	α (°)	Pitch (mm)	Roller	D (mm)	Trough (°)	Tension	Load
F1	0	1200	Steel	152	0	low	NO
F2	0	1200	Steel	152	0	low	YES
F3	0	1200	Steel	152	0	high	NO
F4	0	1200	Steel	152	0	high	YES
F5	0	1800	Steel	152	0	low	NO
F6	0	1800	Steel	152	0	low	YES
F7	0	1800	Steel	152	0	high	NO
F8	0	1800	Steel	152	0	high	YES
F9	0	1200	Steel	152	45	low	NO
F10	0	1200	Steel	152	45	low	YES
F11	0	1200	Steel	152	45	high	NO
F12	0	1200	Steel	152	45	high	YES
F13	0	1800	Steel	152	45	low	NO
F14	0	1800	Steel	152	45	low	YES
F15	0	1800	Steel	152	45	high	NO
F16	0	1800	Steel	152	45	high	YES

Table A.7: Roller type experiments at a constant 35° trough (Programme G)

Test	α (°)	Pitch (mm)	Roller	D (mm)	Trough (°)	Tension	Load
G1	0	1200	HDPE	152	35	low	NO
G2	0	1800	HDPE	152	35	low	NO
G3	0	1200	Steel (NP)	152	35	low	NO
G4	0	1800	Steel (NP)	152	35	low	NO
G5	0	1200	Steel	152	35	low	NO
G6	0	1800	Steel	152	35	low	NO
G7	0	1200	HDPE	152	35	low	YES
G8	0	1800	HDPE	152	35	low	YES
G9	0	1200	Steel (NP)	152	35	low	YES
G10	0	1800	Steel (NP)	152	35	low	YES
G11	0	1200	Steel	152	35	low	YES
G12	0	1800	Steel	152	35	low	YES

Table A.8: Loaded incline experiments at a constant 1800 mm idler pitch (Programme H)

Test	α (°)	Pitch (mm)	Roller	D (mm)	Trough (°)	Tension	Load
H1	0	1800	Steel	152	35	high	YES
H2	5	1800	Steel	152	35	high	YES
H3	10	1800	Steel	152	35	high	YES
H4	15	1800	Steel	152	35	high	YES

Table A.9: Loaded incline experiments at a constant 1200 mm idler pitch (Programme I)

Test	α (°)	Pitch (mm)	Roller	D (mm)	Trough (°)	Tension	Load
I1	0	1200	HDPE	152	35	low	NO
I2	5	1200	HDPE	152	35	low	NO
I3	10	1200	HDPE	152	35	low	NO
I4	15	1200	HDPE	152	35	low	NO
I5	0	1200	HDPE	152	35	low	YES
I6	5	1200	HDPE	152	35	low	YES
I7	10	1200	HDPE	152	35	low	YES
I8	15	1200	HDPE	152	35	low	YES

Table A.10: Supplementary empty belt experiments (Programme J)

Test	α (°)	Pitch (mm)	Roller	D (mm)	Trough (°)	Tension	Load
J1	0	1200	HDPE	152	0	low	NO
J2	0	1200	Steel (NP)	152	0	low	NO
J3	0	1200	HDPE	152	35	low	NO
J4	0	1200	HDPE	152	35	medium	NO
J5	0	1200	HDPE	152	35	high	NO
J6	0	1200	HDPE	152	45	low	NO
J7	0	1200	Steel (NP)	152	45	low	NO
J8	0	1800	HDPE	152	0	low	NO
J9	0	1800	Steel (NP)	152	0	low	NO
J10	0	1800	HDPE	152	35	low	NO
J11	0	1800	HDPE	152	35	medium	NO
J12	0	1800	HDPE	152	35	high	NO
J13	0	1800	HDPE	152	45	low	NO
J14	0	1800	Steel (NP)	152	45	low	NO

A.2 Additional Friction Study

Table A.11: Additional friction study experimental programme

Test No.	Roller Material Sample	Conveyor Belt Sample	Preparation
1	Rough Steel	Rig Nitrile	None
2	Smooth Steel	Rig Nitrile	None
3	Nylon	Rig Nitrile	None
4	HDPE	Rig Nitrile	None
5	Rough Steel	Rig Nitrile	Belt Cleaned
6	Smooth Steel	Rig Nitrile	Belt Cleaned
7	HDPE	Rig Nitrile	Belt Cleaned
8	Rough Steel	Rig Nitrile	Wet Belt
9	Smooth Steel	Rig Nitrile	Wet Belt
10	HDPE	Rig Nitrile	Wet Belt
11	Rough Steel	Used Nitrile	None
12	Smooth Steel	Used Nitrile	None
13	HDPE	Used Nitrile	None
14	Rough Steel	Used Nitrile	Belt Cleaned
15	Smooth Steel	Used Nitrile	Belt Cleaned
16	HDPE	Used Nitrile	Belt Cleaned
17	Rough Steel	Used Nitrile	Wet Belt
18	Smooth Steel	Used Nitrile	Wet Belt
19	HDPE	Used Nitrile	Wet Belt
20	Rough Steel	PVC Nitrile	None
21	Smooth Steel	PVC Nitrile	None
22	HDPE	PVC Nitrile	None
23	Rough Steel	Type M	None
24	Smooth Steel	Type M	None
25	HDPE	Type M	None

Appendix B Unidirectional Idler Proportion Calculating Procedure

The following steps summarise the procedure to follow when calculating the required proportion of unidirectional rollers for a specific installation.

1. Determine the installation specific values of the following list of belt conveying parameters:

α	angle of inclination [°]
β	troughing angle [°]
δ	angle of surcharge [°]
ρ	bulk material density [kg/m ³]
a	idler pitch [m]
B	belt width [m]
l_1	idler roller face length
q_b	mass per metre of the conveyor belt [kg/m]

2. Select the relevant centre and wing roller non-friction coefficients (λ_c and λ_w respectively) from the corresponding non-friction factor tables (Tables 5.4 and 5.5):

Table 5.4: Minimum non-friction factor values for locked centre rollers

	$1200 \leq a < 1500$ mm		$a \geq 1500$ mm	
$0^\circ \leq \beta < 35^\circ$	$q_m = 0$	$\lambda_c = 2.42$	$q_m = 0$	$\lambda_c = 2.70$
	$q_m = FL$	$\lambda_c = 4.45$	$q_m = FL$	$\lambda_c = 3.99$
$35^\circ \leq \beta \leq 45^\circ$	$q_m = 0$	$\lambda_c = 1.55$	$q_m = 0$	$\lambda_c = 1.57$
	$q_m = FL$	$\lambda_c = 1.71$	$q_m = FL$	$\lambda_c = 1.57$

Table 5.5: Minimum non-friction factor values for locked wing rollers

	1200 ≤ a < 1500 mm		a ≥ 1500 mm	
$0^\circ \leq \beta < 35^\circ$	$q_m = 0$	$\lambda_w = 2.21$	$q_m = 0$	$\lambda_w = 2.52$
	$q_m = FL$	$\lambda_w = 1.80$	$q_m = FL$	$\lambda_w = 2.23$
$35^\circ \leq \beta \leq 45^\circ$	$q_m = 0$	$\lambda_w = 1.19$	$q_m = 0$	$\lambda_w = 1.37$
	$q_m = FL$	$\lambda_w = 2.14$	$q_m = FL$	$\lambda_w = 1.80$

3. Select the relevant roller friction coefficient (μ). This may be done using the minimum observed coefficients as determined in this study (Table 5.1), using other sources of roller and conveyor belt friction literature, or by measuring desired friction-pair coefficients in a similar manner to that presented in Section 3.4.

Table 5.1: Minimum observed coefficients of friction (μ) at 15 kpa of contact pressure

Roller Type			
HDPE	Paintless Steel	Nylon	Painted Steel
$\mu = 0.20$	$\mu = 0.27$	$\mu = 0.28$	$\mu = 0.43$

4. Calculate the centre and wing roller resistance coefficients (ψ_c and ψ_w respectively):

$$\begin{aligned}\psi_c &= \lambda_c \cdot \mu \\ \psi_w &= \lambda_w \cdot \mu\end{aligned}\tag{5.2}$$

5. Calculate the usable width of the belt (b):

$$b = 0.9B - 0.05\tag{5.5}$$

6. Calculate the idler set normal load (N). In this equation, the g term is the acceleration due to gravity (assumed to be 9.79 m/s² in this work):

$$N = ag \left\{ q_b + \rho \left[\frac{1}{6} \tan \delta \left(l_1 + (b - l_1) \cos \beta \right)^2 + \frac{1}{4} (b - l_1) \sin \beta \left(2l_1 + (b - l_1) \cos \beta \right) \right] \right\}\tag{5.17}$$

7. Calculate the cross-sectional area of material load directly above the centre rollers (A_c):

$$A_c = \frac{\tan \delta \left\{ 3l_1 \left[l_1 + (b - l_1) \cos \beta \right]^2 - l_1^3 \right\}}{12 \left[l_1 + (b - l_1) \cos \beta \right]} + \frac{l_1(b - l_1) \sin \beta}{2} \quad (5.13)$$

8. Calculate the normal force acting at the centre rollers (N_c):

$$N_c = a \left(q_b \left(\frac{l_1}{B} \right) + \rho A_c \right) g \quad (5.18)$$

9. Select the normal coefficient (ω) according to the conveyor belt troughing angle (β):

$$\omega = \begin{cases} 1.00, & \text{if } \beta = 0^\circ \\ 1.43, & \text{if } \beta = 35^\circ \\ 1.90, & \text{if } \beta = 45^\circ \end{cases} \quad (5.20)$$

10. Calculate the normal force acting at the wing rollers (N_w):

$$N_w = a\omega \left(q_b \left(1 - \frac{l_1}{B} \right) + \rho A_w \right) g \quad (5.19)$$

11. Calculate the unidirectional centre roller proportion factor (ϵ), including a factor of safety (n):

$$\epsilon = \frac{nN}{\psi_c N_c} \tan \alpha \quad (5.26)$$

12. Determine the proportion of centre rollers required to be unidirectional ($\eta_{c,s}$) to prevent conveyor belt rollback/runaway:

$$\eta_{c,s} = \begin{cases} \epsilon & \text{if } \epsilon < 1 \\ 1 & \text{if } \epsilon \geq 1 \end{cases} \quad (5.27)$$

13. If installing unidirectional centre rollers into all idler frames is not sufficient to prevent belt runaway (i.e. $\epsilon > 1$), then calculate the proportion of wing rollers required to be unidirectional ($\eta_{w,s}$):

$$\eta_{w,s} = (\epsilon - 1) \frac{\psi_c N_c}{\psi_w N_w}; \quad \text{if } \epsilon > 1 \quad (5.28)$$

14. If $\eta_{w,s} > 1$, then it is not safe to use the chosen unidirectional idlers in the specific inclined conveyor belt system under consideration.

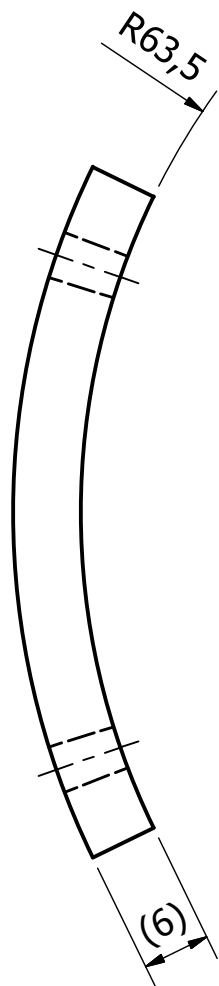
Appendix C Manufacturing Drawings

The following Appendix presents the manufacturing drawings of the belt clamps and load cell mounting brackets designed for the UDR Test Rig. Table C.1 lists these drawings in order of appearance.

Table C.1: List of manufacturing drawings in order of appearance

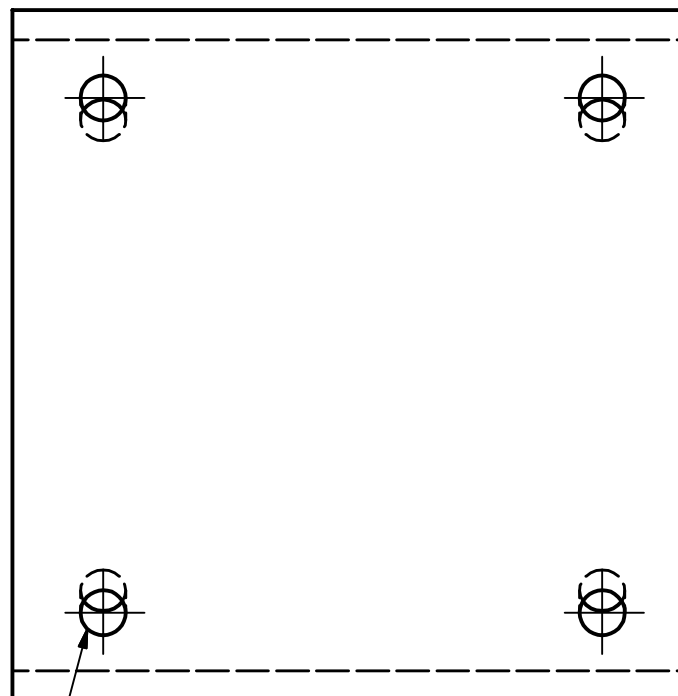
Drawing Number	Drawing Description
ARB001	127 mm Roll Attachment Plate
ARB002	152 mm Roll Attachment Plate
ARB003	Clevis Head
ARB004	Clevis Pin
ARB005	127 mm Roll Torque Arm Assembly
ARB006	152 mm Roll Torque Arm Assembly
LCP001	0° Mounting Plate
LCP002	35° Mounting Plate
LCP003	45° Mounting Plate
LCP004	Positioning Bar
LCP005	Attachment Angle Iron
LCP006	Clamping Angle Iron
LCP007	Spacer Plate
LCP008	Parts List
LCP009	0° Mount Assembly
LCP010	35° Mount Assembly
LCP011	45° Mount Assembly

Drawing Number	Drawing Description
BCL001	Clamping Angle Iron
BCL002	Short Clamping Angle Iron
BCL003	0° Belt Support Angle Iron
BCL004	Mounting Base Plate
BCL005	0° Mounting Plate
BCL006	900 mm Belt Width, 35° Mounting Plate
BCL007	1200 mm Belt Width, 35° Mounting Plate
BCL008	1500 mm Belt Width, 35° Mounting Plate
BCL009	900 mm Belt Width, 45° Mounting Plate
BCL010	1200 mm Belt Width, 45° Mounting Plate
BCL011	1500 mm Belt Width, 45° Mounting Plate
BCL012	1500 mm Belt Width, 45° Support Angle Iron
BCL013	1200 mm Belt Width, 45° Support Angle Iron
BCL014	900 mm Belt Width, 45° Support Angle Iron
BCL015	1500 mm Belt Width, 35° Support Angle Iron
BCL016	1200 mm Belt Width, 35° Support Angle Iron
BCL017	900 mm Belt Width, 35° Support Angle Iron
BCL018	Large Stiffener Plate
BCL019	Small Stiffener Plate
BCL020	0° Belt Clamp Base
BCL021-026	35° and 45° Belt Clamp Bases (All Widths)
BCL027-033	Belt Clampe Full Assembly Drawings



Ø4 THRU
(Guide Holes for Self Tapping Screws)
Cut BEFORE Rolling
4 PLCS

Rolled Plate 01: QTY 6 OFF



SCALE: 1.5:1

NOTES:
SURFACE FINISH $\sqrt{3.2}$ UNLESS
OTHERWISE STATED.
DIMENSIONS IN mm.
DRILL & TAP CENTRE HOLE
ONLY AFTER PLATE IS
ROLLED

APPROVED:

CHECKED:

DATE: 08/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: ARB001

REV NO. 0



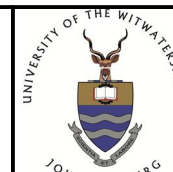
SHEET 1 OF 4

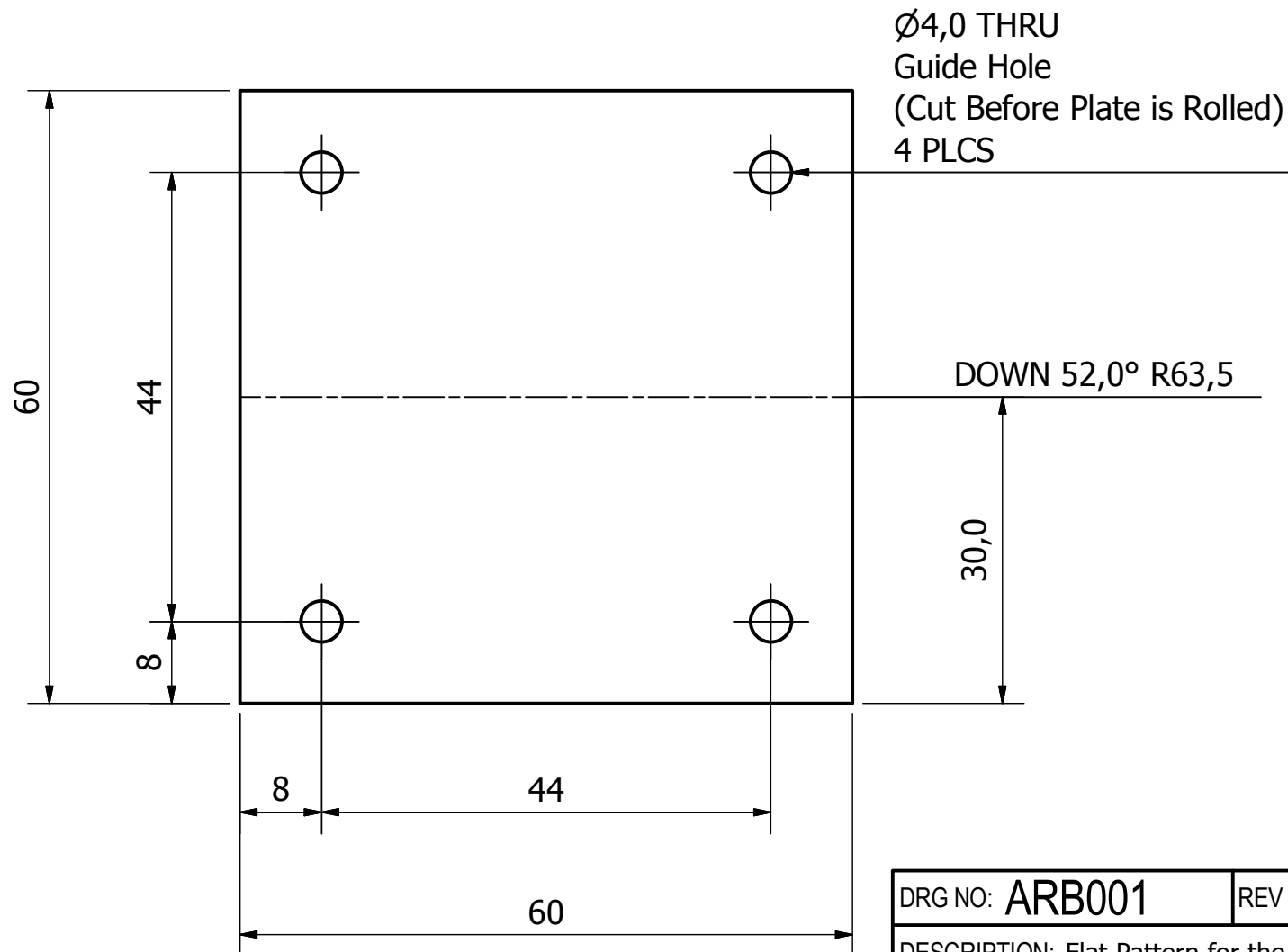
DESCRIPTION: 127 mm Roll Attachment Drawings

MATERIAL: 355JR x 6 mm Steel Plate



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SCALE: 1.5:1

NOTES:
SURFACE FINISH $\sqrt{3.2}$ UNLESS
OTHERWISE STATED.
DIMENSIONS IN mm.
DRILL & TAP CENTRE HOLE
ONLY AFTER PLATE IS
ROLLED

APPROVED:

CHECKED:

DATE: 08/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: ARB001

REV NO. 0



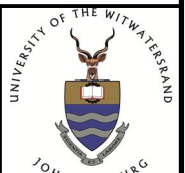
SHEET 2 OF 4

DESCRIPTION: Flat Pattern for the 127 mm Rolled Plate 01

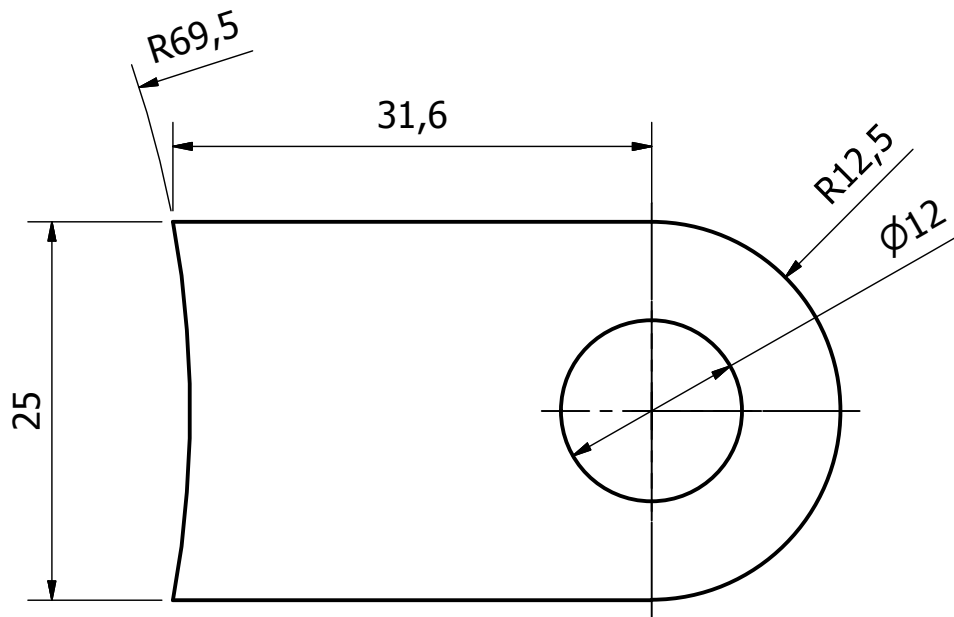
MATERIAL: 355JR x 6 mm Steel Plate



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Rolled Plate 01: QTY 6 OFF



SCALE: 2:1

NOTES: 3.2/✓
 SURFACE FINISH UNLESS
 OTHERWISE STATED.
 DIMENSIONS IN mm.
 DRILL & TAP CENTRE HOLE
 ONLY AFTER PLATE IS
 ROLLED

APPROVED:

CHECKED:

DATE: 08/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: **ARB001**

REV NO. 0



SHEET 3 OF 4

DESCRIPTION: Attachment Lug 01 Manufacturing Drawing

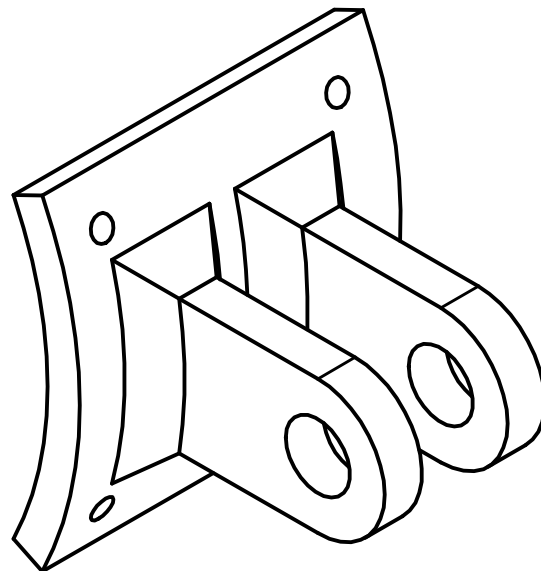
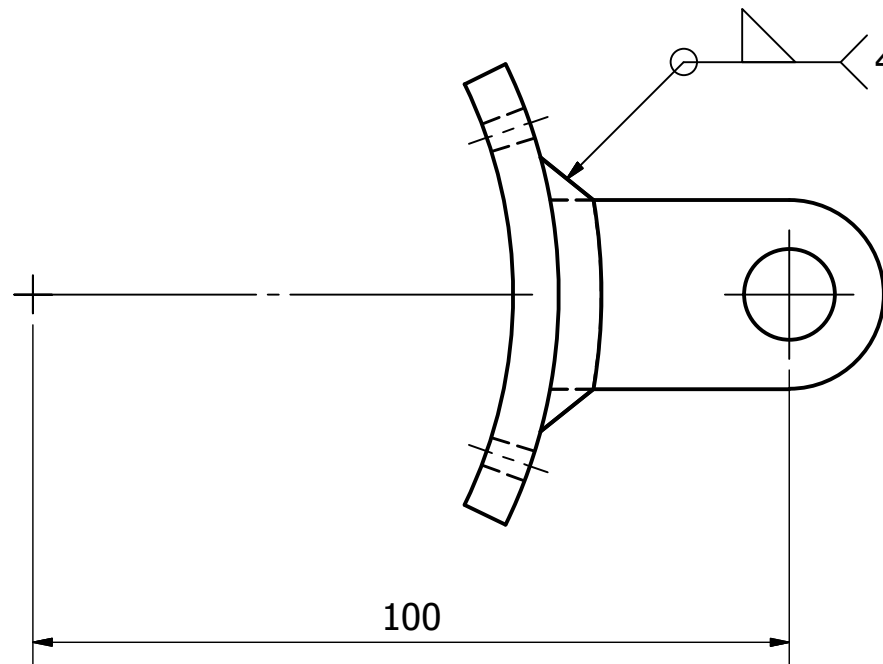
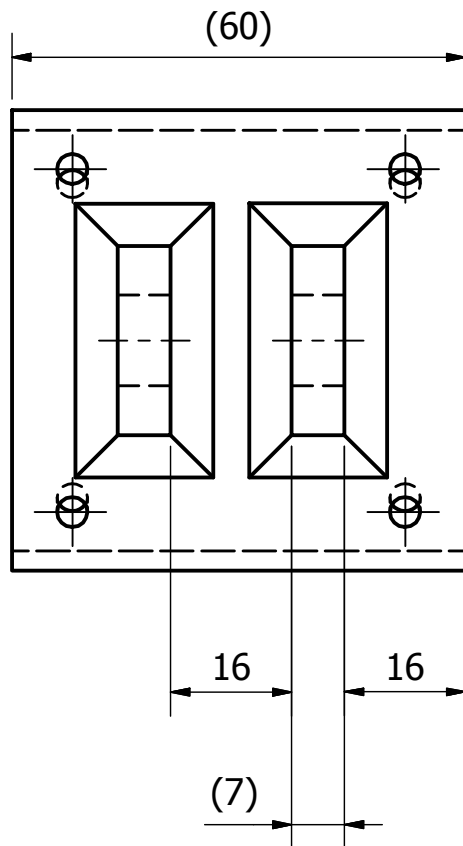
MATERIAL: 355JR x 7 mm Steel Plate



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Attachment Lug 01: QTY 12 OFF



SCALE: 1:1

NOTES:
 SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
 DIMENSIONS IN mm.
 DRILL & TAP CENTRE HOLE ONLY AFTER PLATE IS ROLLED

APPROVED:

CHECKED:

DATE: 08/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: ARB001

REV NO. 0



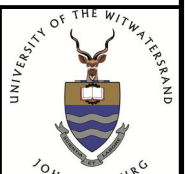
SHEET 4 OF 4

DESCRIPTION: 127 mm Roll Attachment Assembly Drawing

MATERIAL:



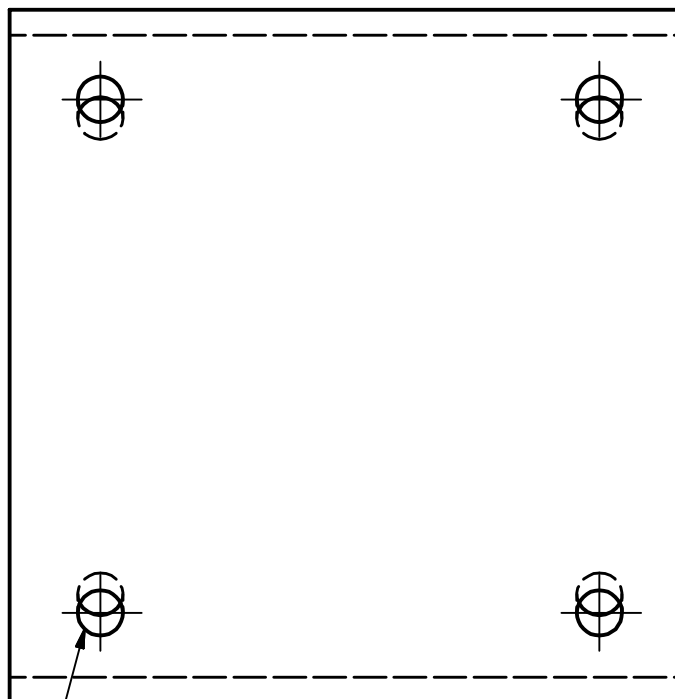
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Assembly: QTY 6 OFF



Ø4 THRU
(Guide Holes for Self Tapping Screws)
Cut BEFORE Rolling
4 PLCS



Rolled Plate 02: QTY 6 OFF

SCALE: 1.5:1

NOTES:
SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
DIMENSIONS IN mm.
DRILL & TAP CENTRE HOLE ONLY AFTER PLATE IS ROLLED

APPROVED:

CHECKED:

DATE: 08/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: ARB002

REV NO. 0



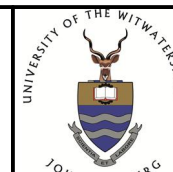
SHEET 1 OF 4

DESCRIPTION: 152 mm Roll Attachment Drawings

MATERIAL: 355JR x 6 mm Steel Plate



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Ø4 THRU
Guide Hole
(Cut Before Plate is Rolled)
4PLCS

DOWN 43,7° R76,0

SCALE: 1.5:1

NOTES:
SURFACE FINISH $\nabla^{3.2}$ UNLESS
OTHERWISE STATED.
DIMENSIONS IN mm.
DRILL & TAP CENTRE HOLE
ONLY AFTER PLATE IS
ROLLED

APPROVED:

CHECKED:

DATE: 08/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: **ARB002**

REV NO. 0



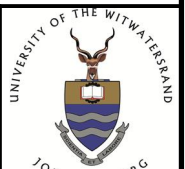
SHEET 2 OF 4

DESCRIPTION: Flat Pattern for the 152 mm Rolled Plate 02

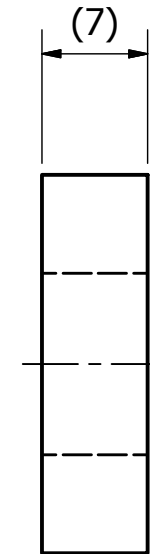
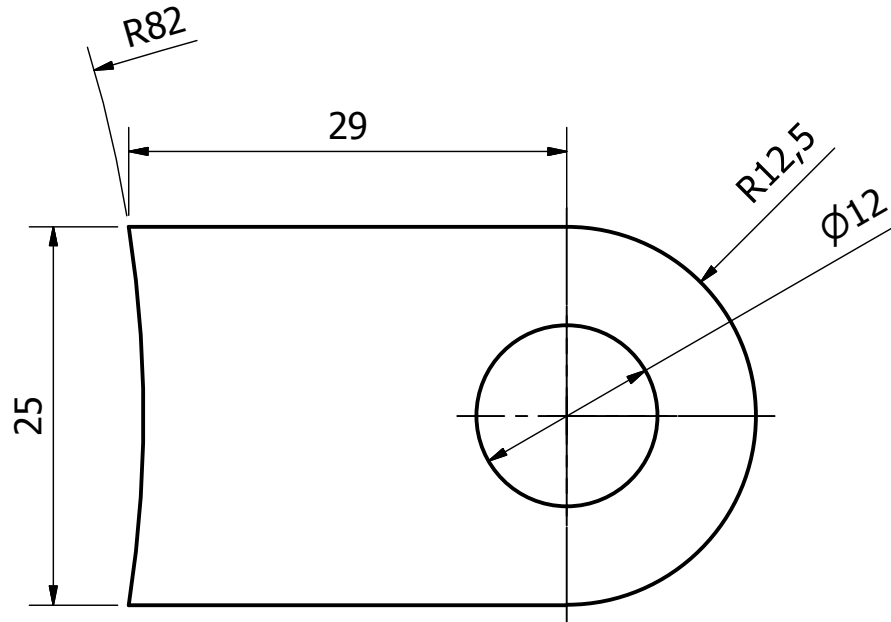
MATERIAL: 355JR x 6 mm Steel Plate



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Rolled Plate 02: QTY 6 OFF



SCALE: 2:1

NOTES: ^{3.2}✓
 SURFACE FINISH UNLESS
 OTHERWISE STATED.
 DIMENSIONS IN mm.
 DRILL & TAP CENTRE HOLE
 ONLY AFTER PLATE IS
 ROLLED

APPROVED:

CHECKED:

DATE: 08/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: ARB002

REV NO. 0



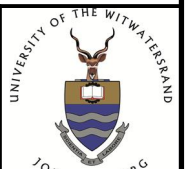
SHEET 3 OF 4

DESCRIPTION: Attachment Lug 02 Manufacturing Drawing

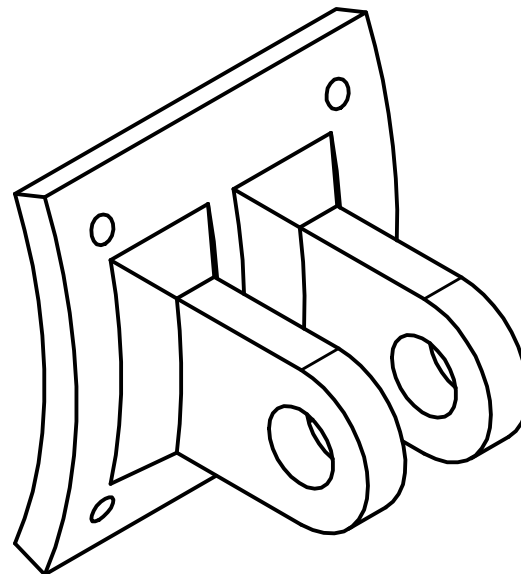
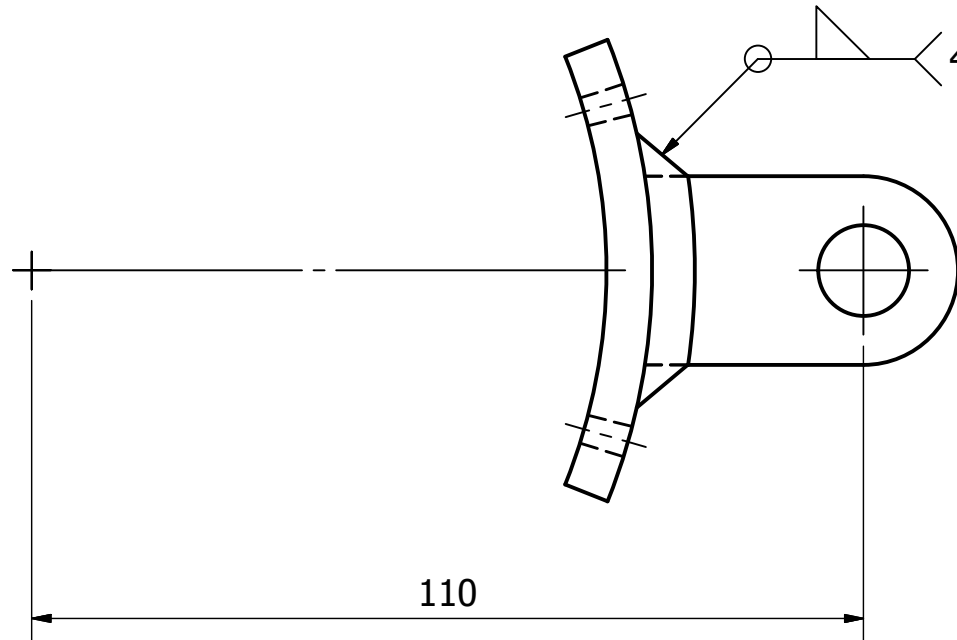
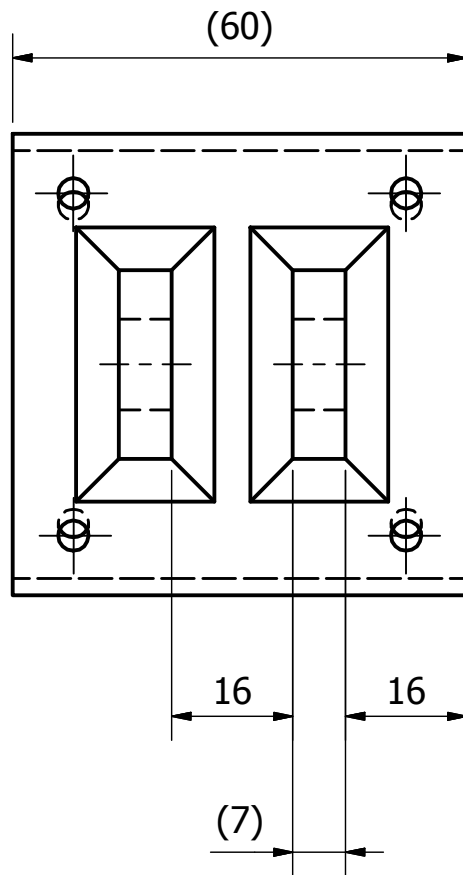
MATERIAL: 355JR x 7 mm Steel Plate



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Attachment Lug 02: QTY 12 OFF



SCALE: 1:1

NOTES:
 SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
 DIMENSIONS IN mm.
 DRILL & TAP CENTRE HOLE ONLY AFTER PLATE IS ROLLED

APPROVED:

CHECKED:

DATE: 08/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: ARB002

REV NO. 0



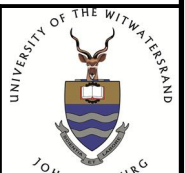
SHEET 4 OF 4

DESCRIPTION: 152 mm Roll Attachment Assembly Drawing

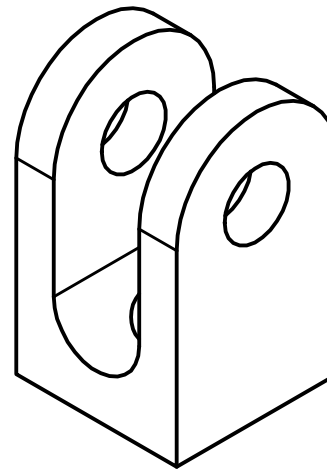
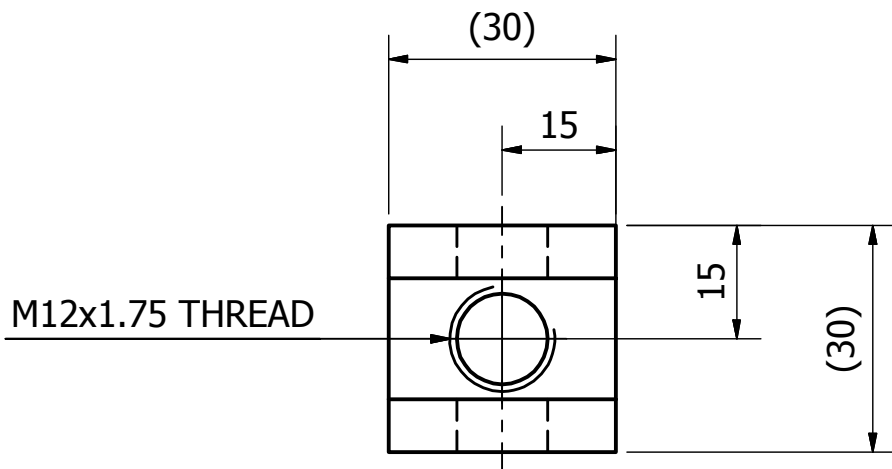
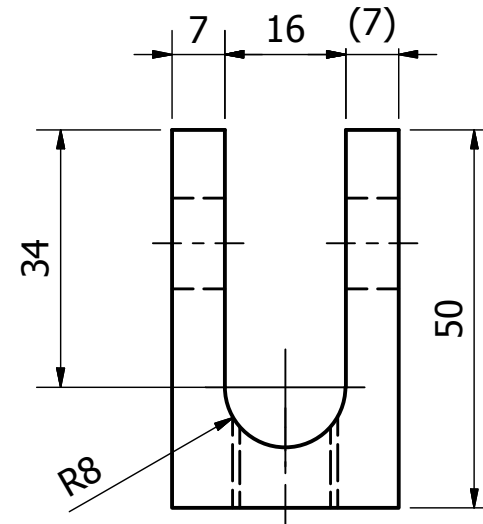
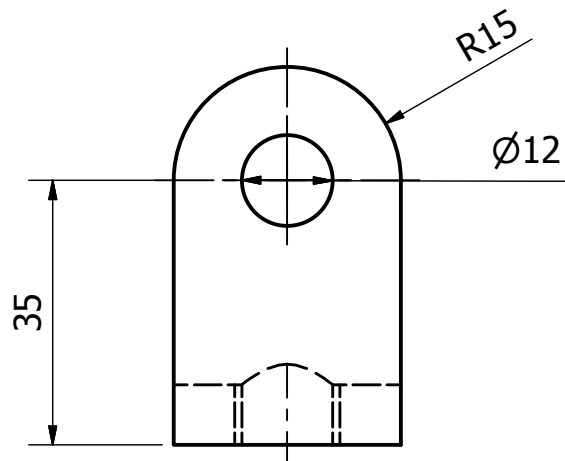
MATERIAL:



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Assembly: QTY 6 OFF



SCALE: 1:1

NOTES:
 SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
 DIMENSIONS IN mm.
 DRILL & TAP CENTRE HOLE ONLY AFTER PLATE IS ROLLED

APPROVED:

CHECKED:

DATE: 08/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: ARB003

REV NO. 0



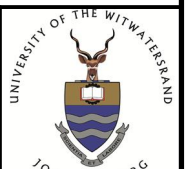
SHEET 1 OF 2

DESCRIPTION: Clevis Head Manufacturing Drawing

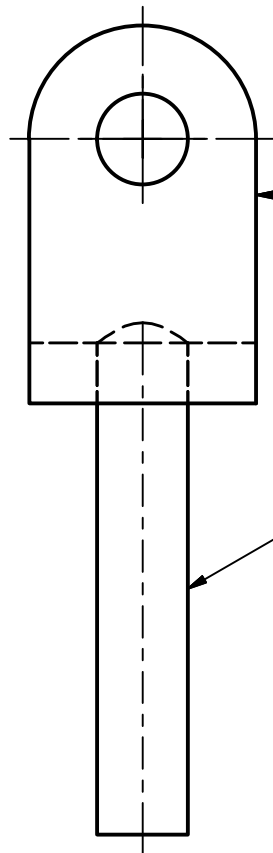
MATERIAL: 30 mm Square Bar Mild Steel



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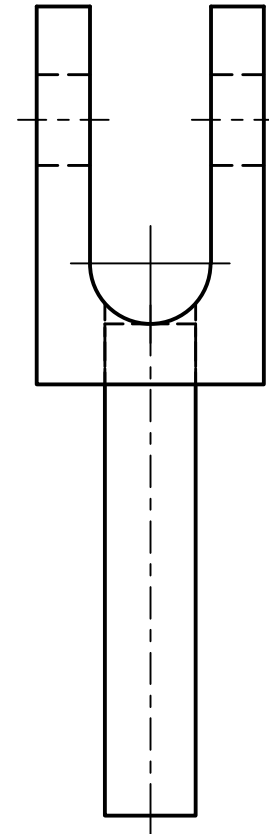
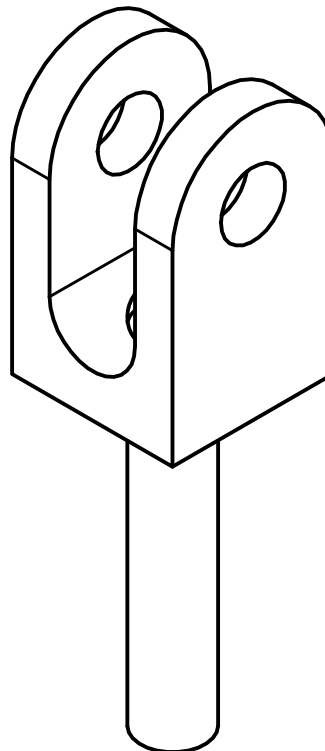


Clevis Head: QTY 6 OFF



Clevis Head

M12 x 65 mm Threaded Rod



SCALE: 1:1

NOTES:
SURFACE FINISH $\frac{3.2}{\checkmark}$ UNLESS OTHERWISE STATED.
DIMENSIONS IN mm.
DRILL & TAP CENTRE HOLE ONLY AFTER PLATE IS ROLLED

APPROVED:

CHECKED:

DATE: 08/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: ARB003

REV NO. 0



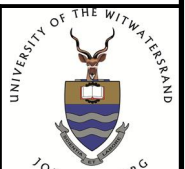
SHEET 2 OF 2

DESCRIPTION: Clevis Assembly Manufacturing Drawing

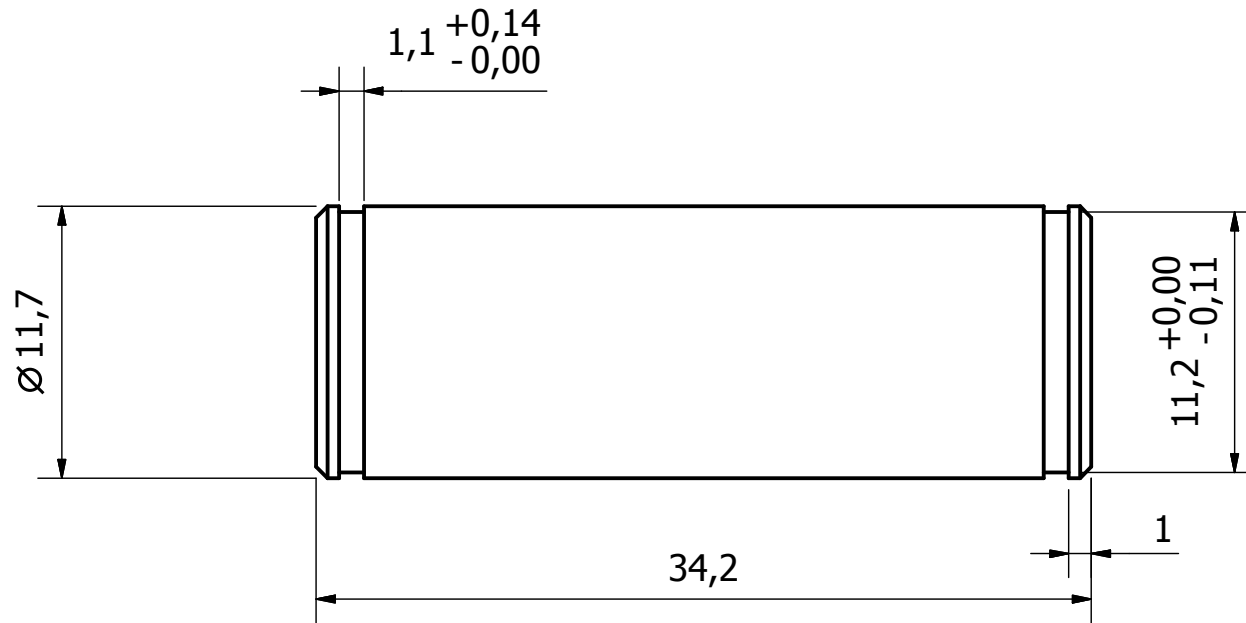
MATERIAL:



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Assembly: QTY 6 OFF



SCALE: 3:1

NOTES: 3.2/✓
 SURFACE FINISH UNLESS
 OTHERWISE STATED.
 DIMENSIONS IN mm.
 DRILL & TAP CENTRE HOLE
 ONLY AFTER PLATE IS
 ROLLED

APPROVED:

CHECKED:

DATE: 08/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: ARB004 REV NO. 0  SHEET 2 OF 2

DESCRIPTION: Clevis Pin Manufacturing Drawing

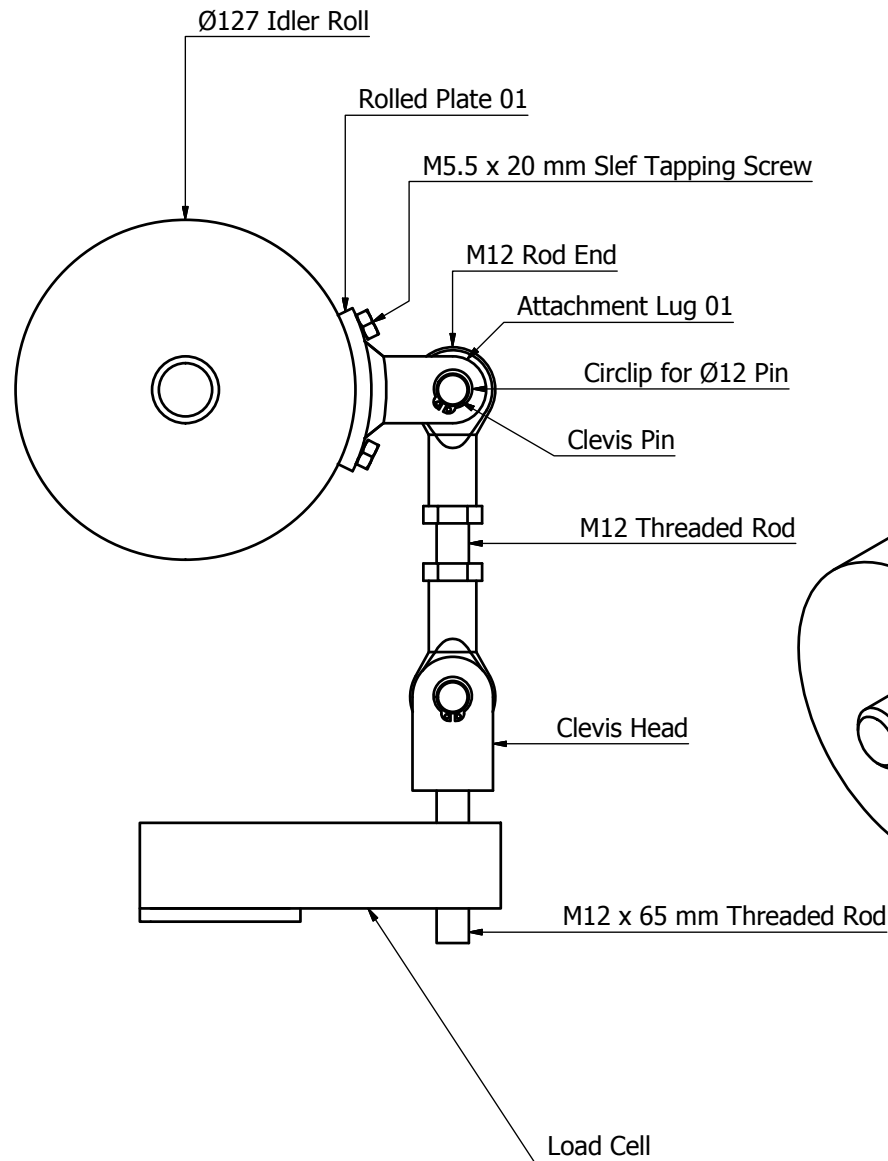
MATERIAL: 12 mm Diameter Round Bar

QTY 12 OFF



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QTY: 6 OFF

SCALE: 1:2

NOTES:
SURFACE FINISH $\sqrt{3.2}$ UNLESS
OTHERWISE STATED.
DIMENSIONS IN mm.

APPROVED:

CHECKED:

DATE: 08/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A3

DRG NO: ARB005

REV NO. 0



SHEET 1 OF 1

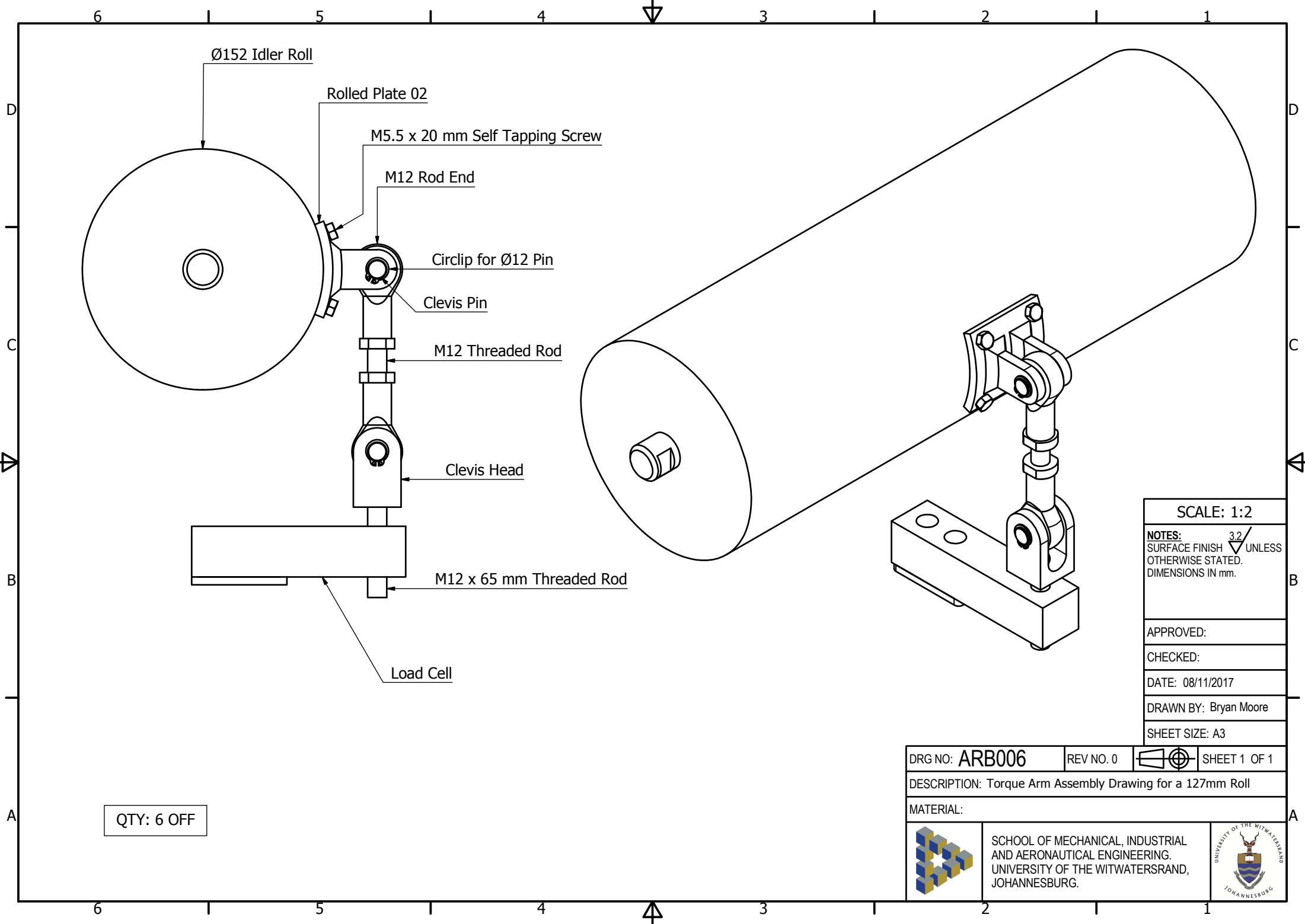
DESCRIPTION: Torque Arm Assembly Drawing for a 127mm Roll

MATERIAL:



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Ø152 Idler Roll

Rolled Plate 02

M5.5 x 20 mm Self Tapping Screw

M12 Rod End

Circlip for Ø12 Pin

Clevis Pin

M12 Threaded Rod

Clevis Head

M12 x 65 mm Threaded Rod

Load Cell

SCALE: 1:2

NOTES:
SURFACE FINISH $\sqrt{3.2}$ UNLESS
OTHERWISE STATED.
DIMENSIONS IN mm.

APPROVED:

CHECKED:

DATE: 08/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A3

DRG NO: ARB006

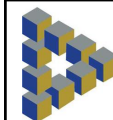
REV NO. 0



SHEET 1 OF 1

DESCRIPTION: Torque Arm Assembly Drawing for a 127mm Roll

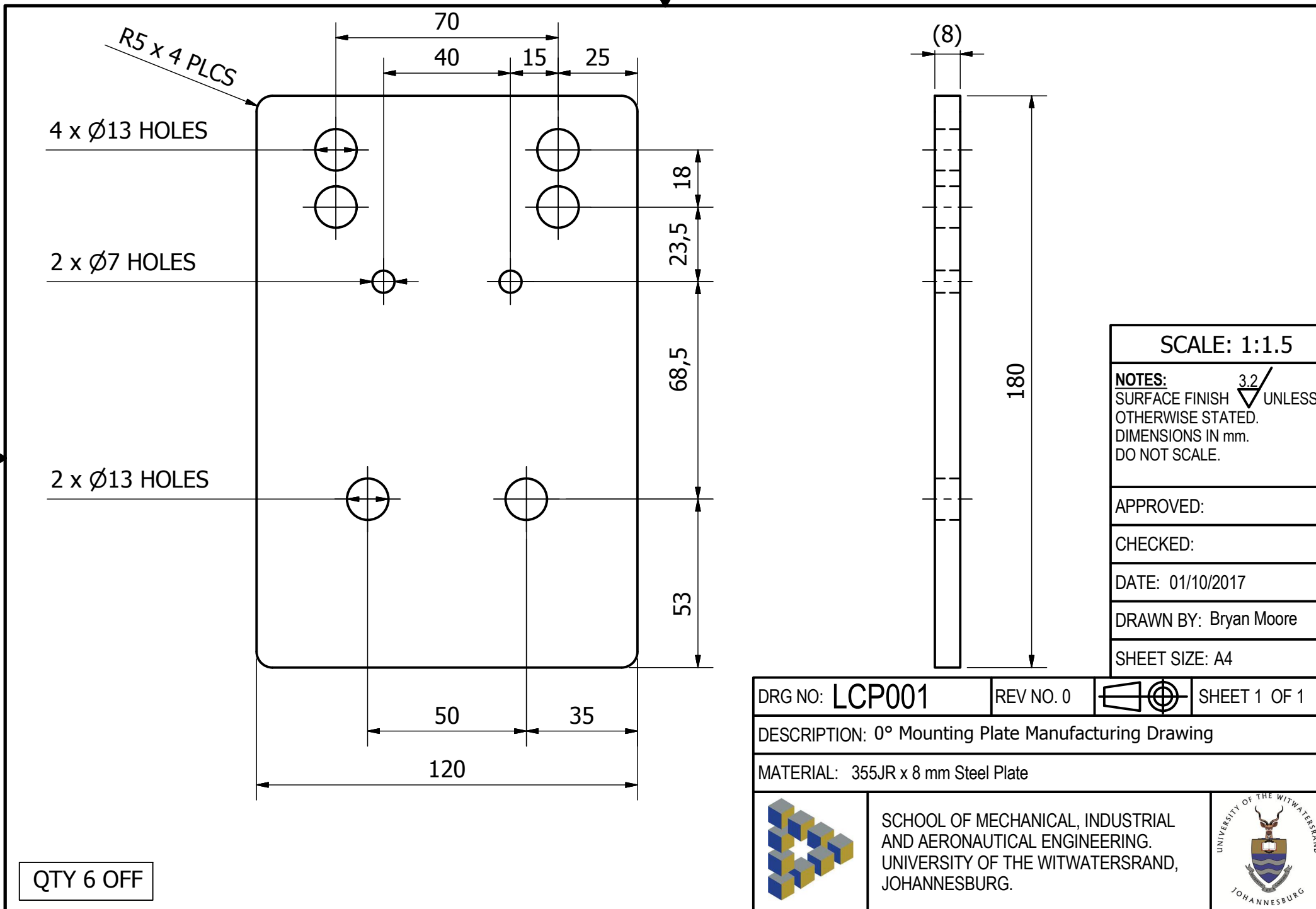
MATERIAL:



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QTY: 6 OFF



R5 x 4 PLCS

4 x Ø13 HOLES

2 x Ø7 HOLES

2 x Ø13 HOLES

70
40 15 25

18
23,5
68,5
53

50 35
120

(8)

180

SCALE: 1:1.5

NOTES: SURFACE FINISH 3.2 UNLESS OTHERWISE STATED. DIMENSIONS IN mm. DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 01/10/2017

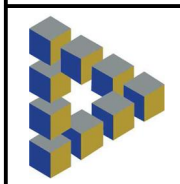
DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: LCP001 REV NO. 0 SHEET 1 OF 1

DESCRIPTION: 0° Mounting Plate Manufacturing Drawing

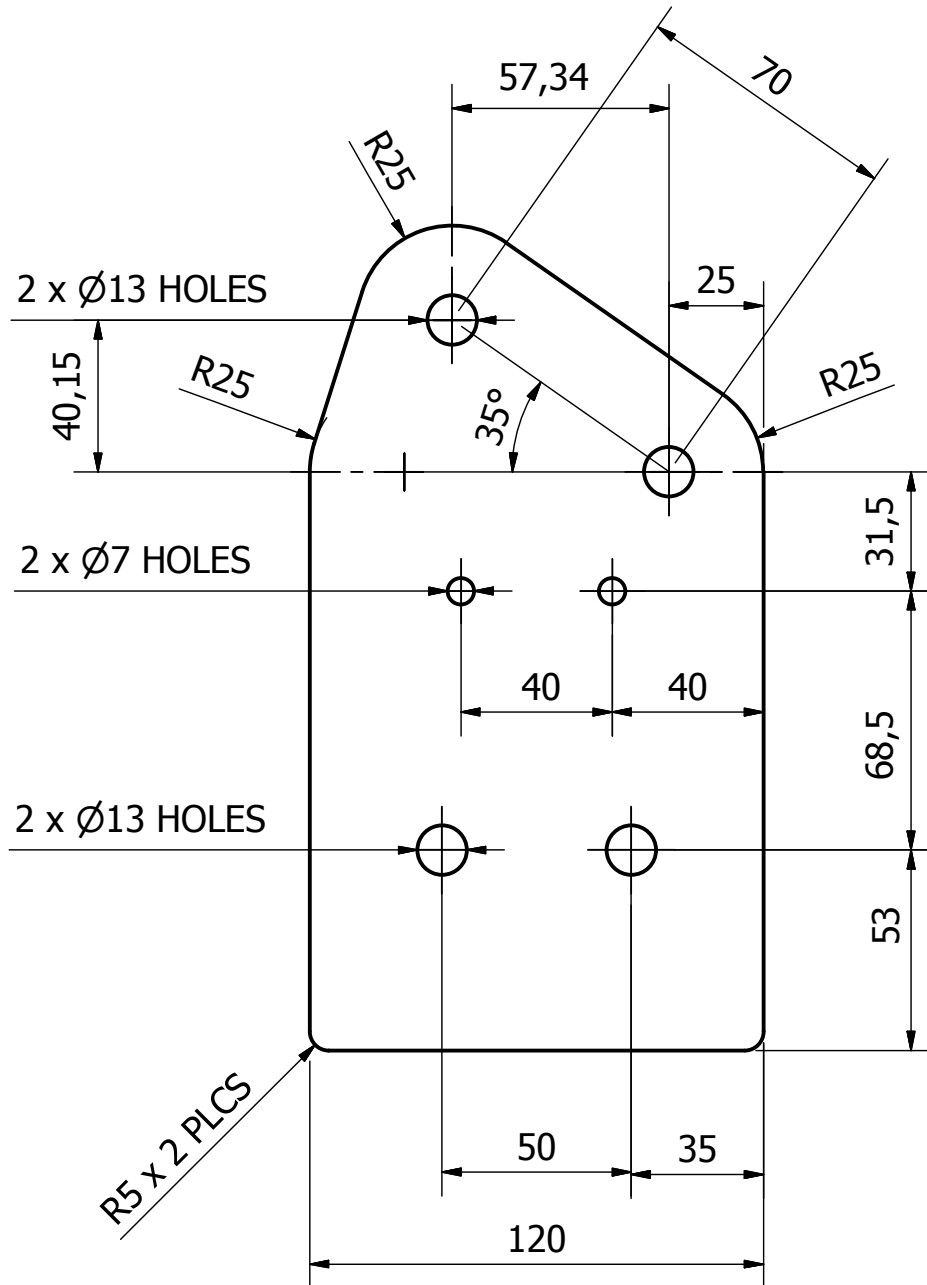
MATERIAL: 355JR x 8 mm Steel Plate



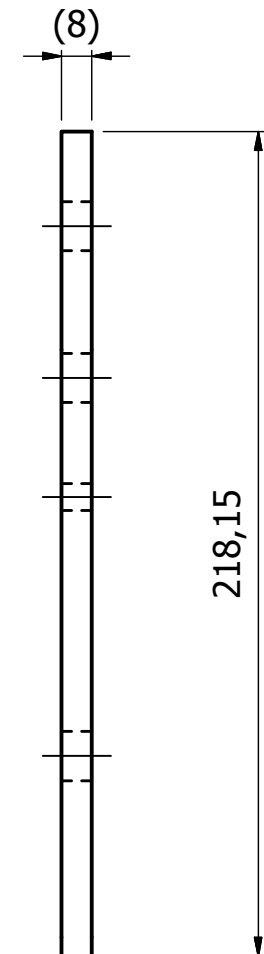
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QTY 6 OFF



QTY 4 OFF



SCALE: 1:2

NOTES: ^{3.2}✓
SURFACE FINISH UNLESS
OTHERWISE STATED.
DIMENSIONS IN mm.
DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 01/10/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: LCP002

REV NO. 0



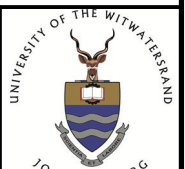
SHEET 1 OF 1

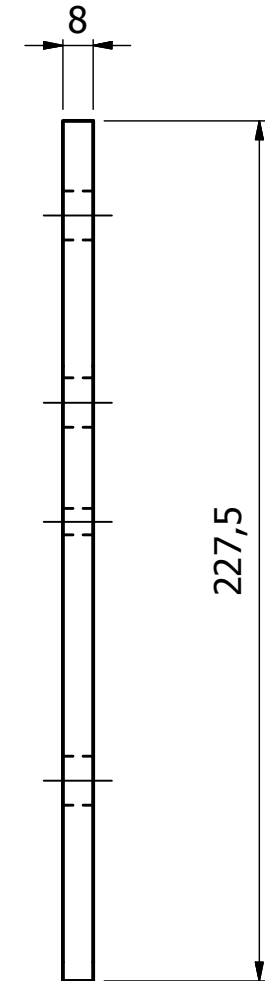
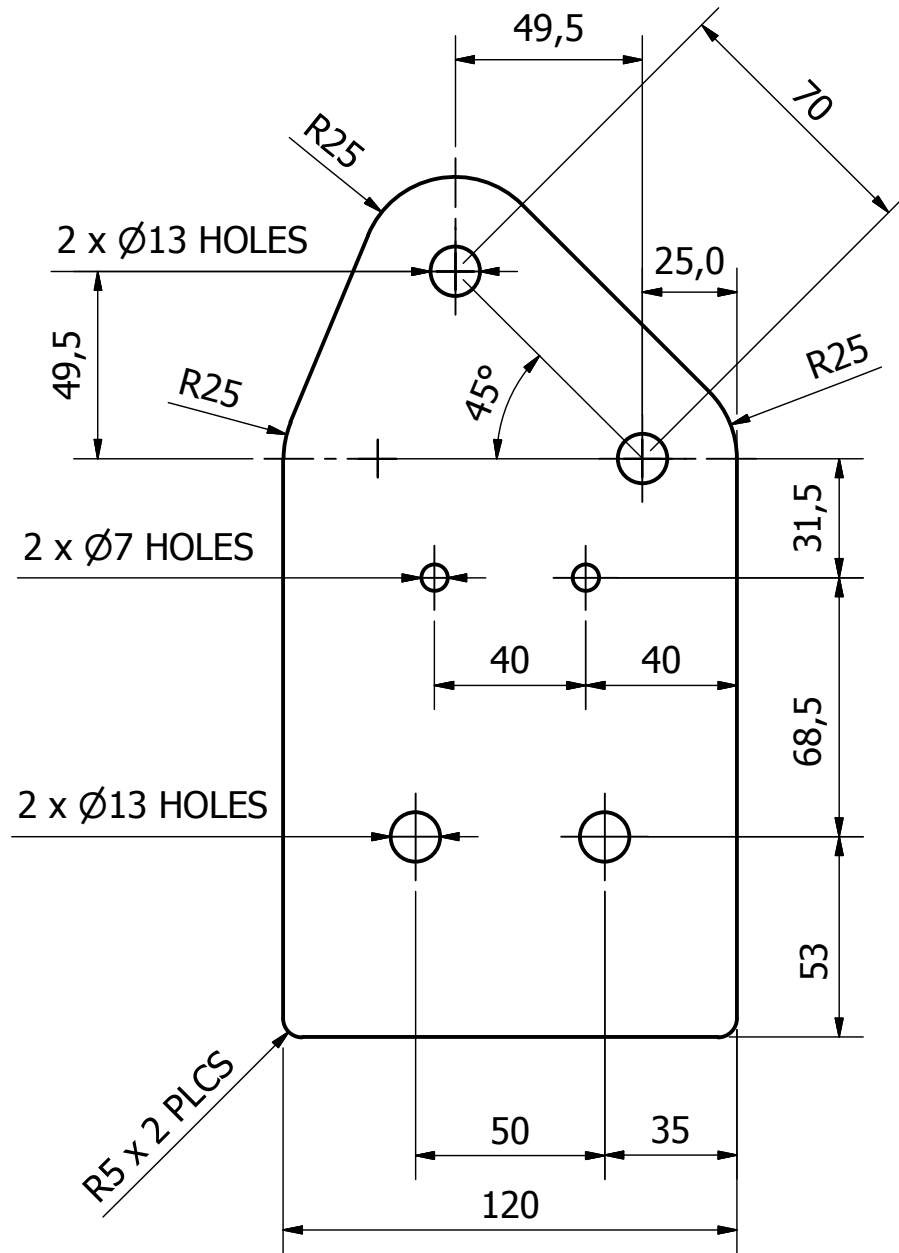
DESCRIPTION: 35° Mounting Plate Manufacturing Drawing

MATERIAL: 355JR x 8 mm Steel Plate



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SCALE: 1:2

NOTES:
 SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
 DIMENSIONS IN mm.
 DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 01/10/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: LCP003

REV NO. 0



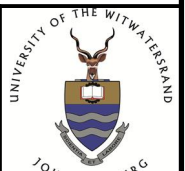
SHEET 1 OF 1

DESCRIPTION: 45° Mounting Plate Manufacturing Drawing

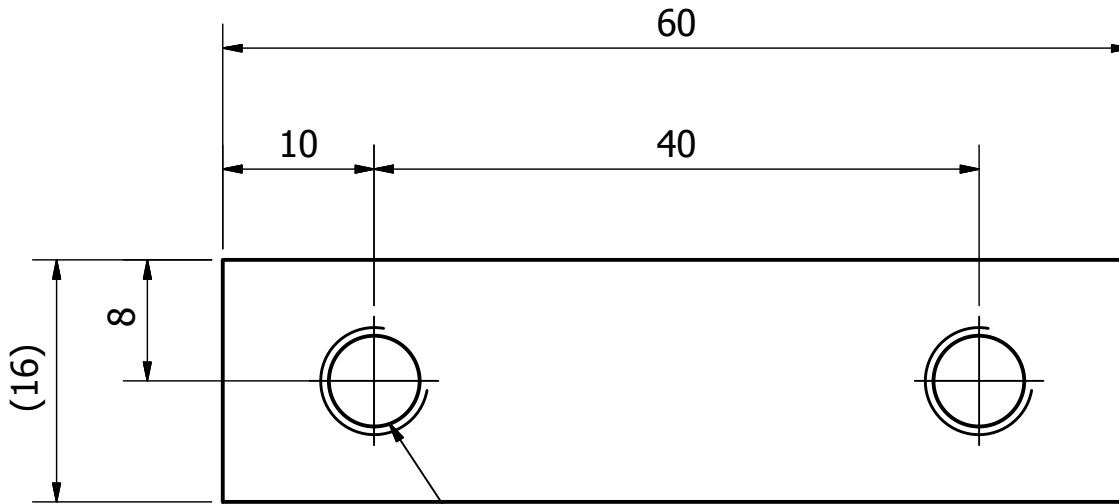
MATERIAL: 355JR x 8 mm Steel Plate



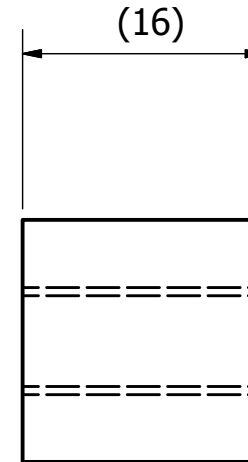
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 JOHANNESBURG.



QTY 4 OFF



Ø5 THRU
M6x1 THREAD
2 PLCS



SCALE: 2:1

NOTES: SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
DIMENSIONS IN mm.
DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 01/10/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: LCP004

REV NO. 0



SHEET 1 OF 1

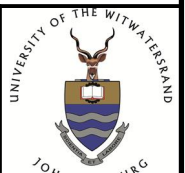
DESCRIPTION: Positioning Bar Manufacturing Drawing

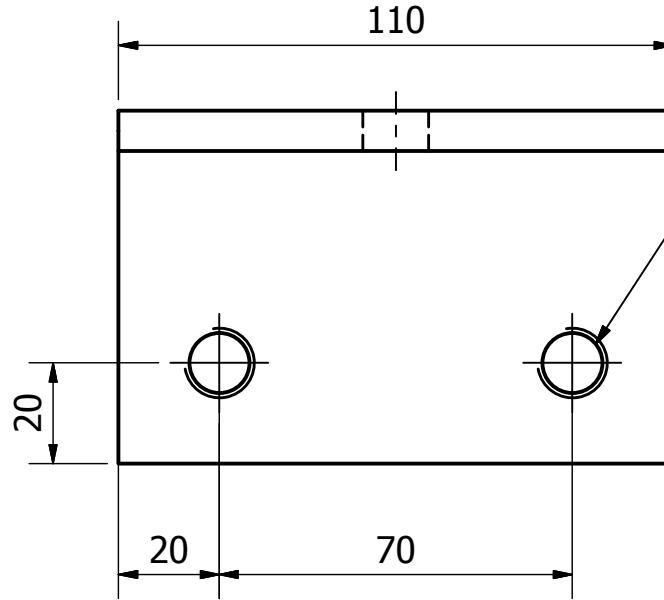
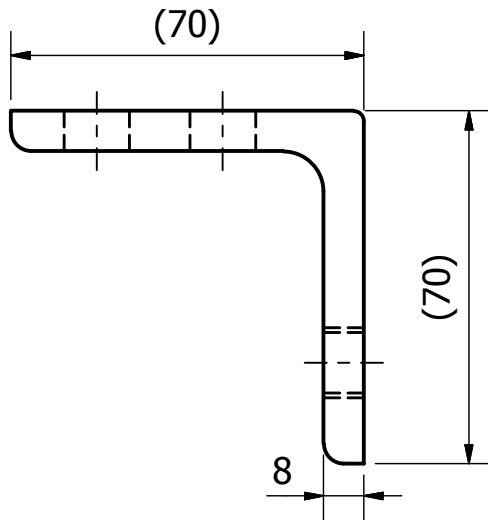
MATERIAL: 16 x 16 mm Square Bar Mild Steel

QTY 6 OFF

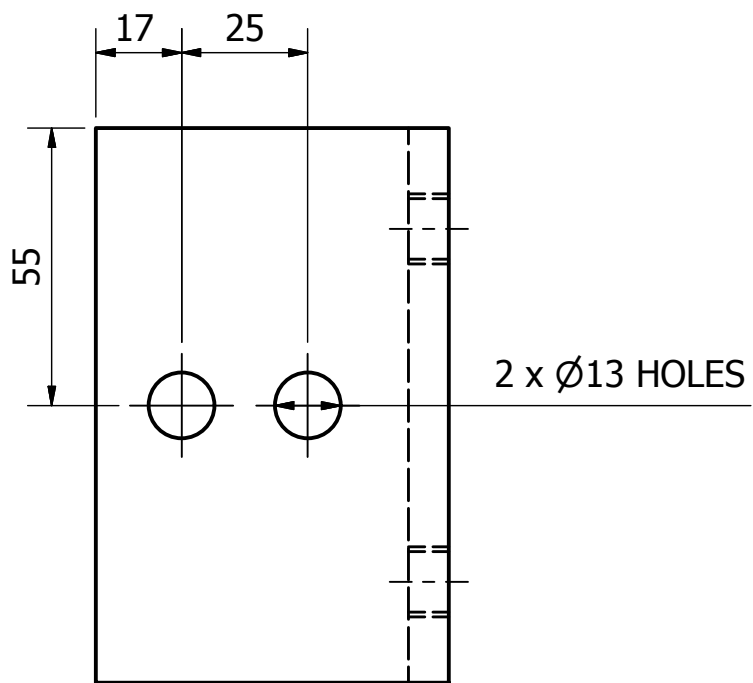


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Ø10.2 THRU
M12x1.75 THREAD
2 PLCS



QTY 6 OFF

SCALE: 1:1.5

NOTES:
SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
DIMENSIONS IN mm.
DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 01/10/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: LCP005	REV NO. 0		SHEET 1 OF 1
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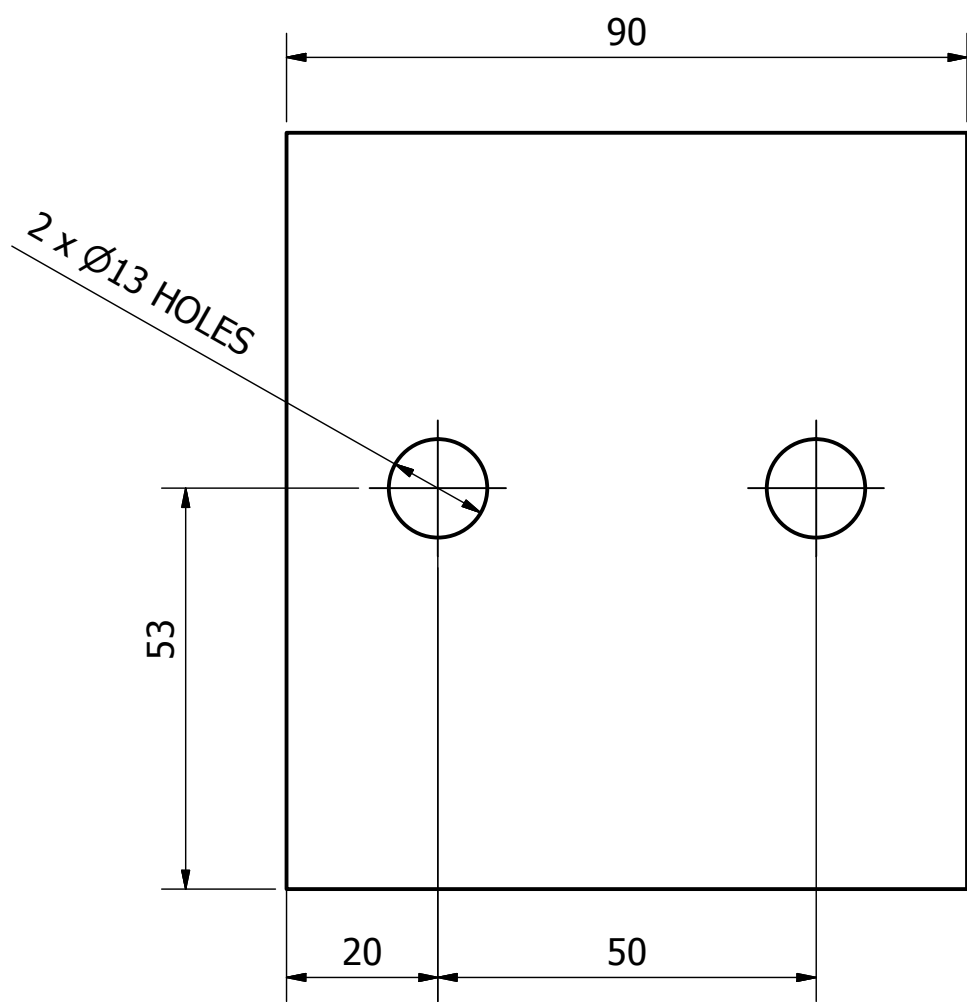
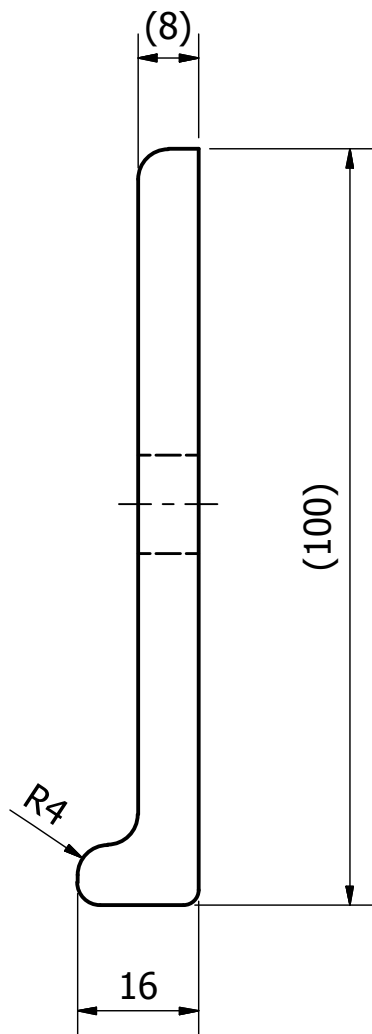
DESCRIPTION: Attachment Angle Iron Manufacturing Drawing

MATERIAL: 70 x 70 x 8 mm Equal Angle SANS 1431 Grade 350WA Steel



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SCALE: 1:1	
NOTES: SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED. DIMENSIONS IN mm. MADE BY CUTTING 100 X 100 X 8 mm ANGLE IRON TO LENGTH.	
APPROVED:	
CHECKED:	
DATE: 01/10/2017	
DRAWN BY: Bryan Moore	
SHEET SIZE: A4	

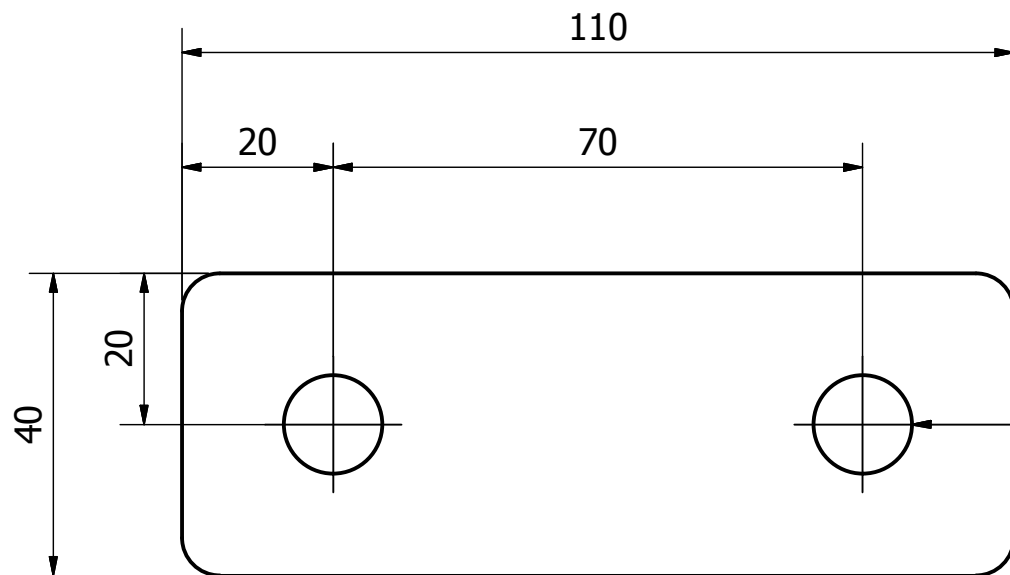
DRG NO: LCP006	REV NO. 0		SHEET 1 OF 1
DESCRIPTION: Clamping Angle Iron Manufacturing Drawing			
MATERIAL: 100 x 100 x 8 mm Equal Angle SANS 1431 Grade 350WA Steel			

QTY 6 OFF

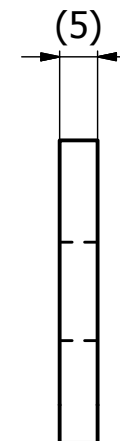


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Ø13 THRU
2 PLCS



SCALE: 1:1

NOTES:
SURFACE FINISH $\sqrt{3.2}$ UNLESS
OTHERWISE STATED.
DIMENSIONS IN mm.

APPROVED:

CHECKED:

DATE: 10/10/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: LCP007

REV NO. 0



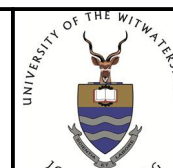
SHEET 1 OF 1

DESCRIPTION: Spacer Plate

MATERIAL: 355JR x 5 mm Steel Plate



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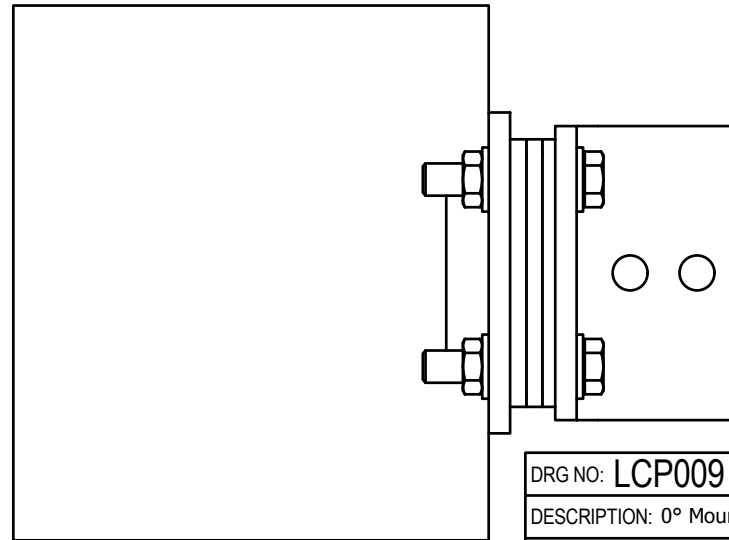
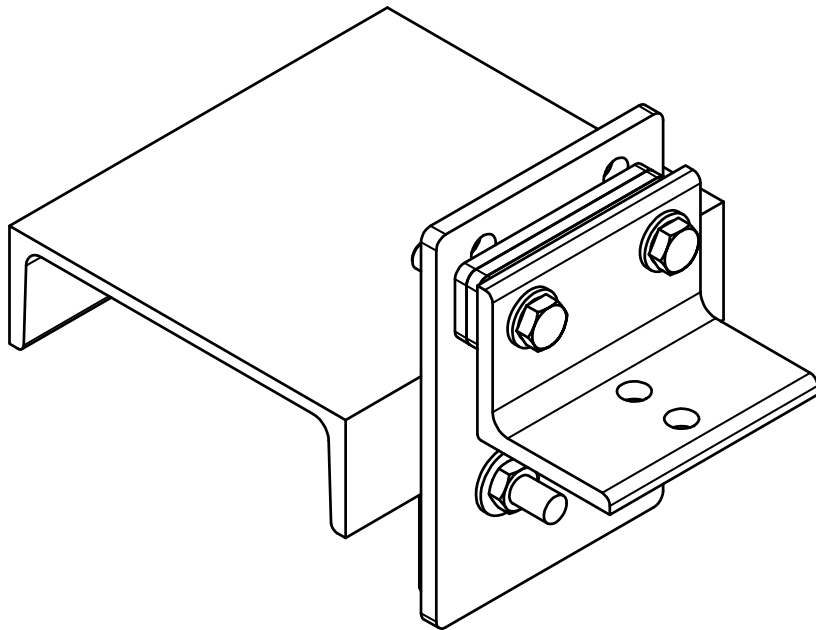
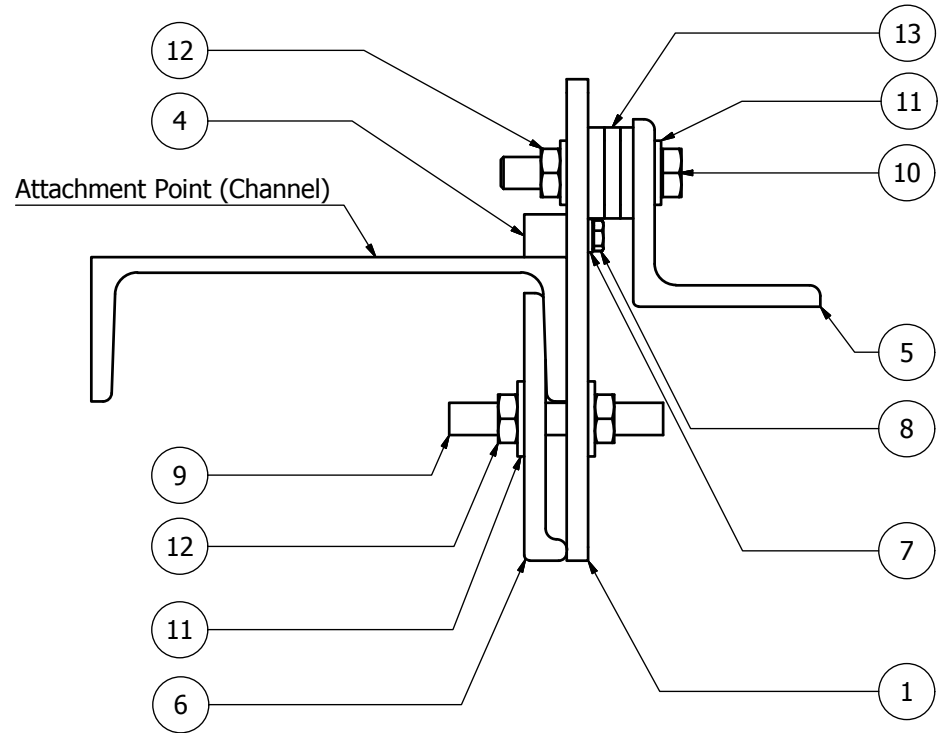
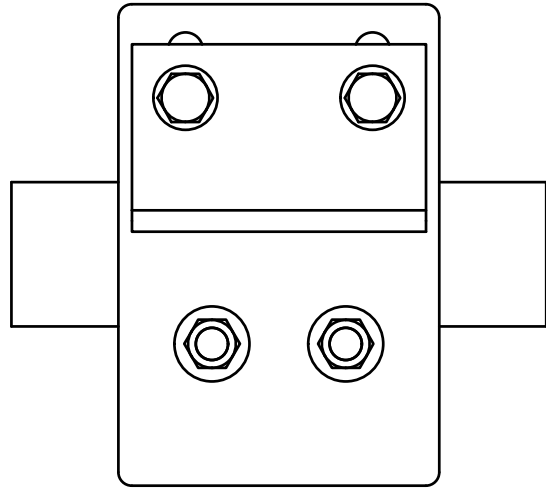


QTY 60 OFF

PARTS LIST	
ITEM	PART
1	0 Degree Mounting Plate
2	35 Degree Mounting Plate
3	45 Degree Mounting Plate
4	Positioning Bar
5	Attachment Angle Iron
6	Clamping Angle Iron
7	M6 Washers
8	M6 x 20 mm Bolt
9	M12 x 80 mm Threaded Rod
10	M12 x 80 mm Bolts
11	M12 Washers
12	M12 Nuts
13	Spacer Plate

APPROVED:
CHECKED:
DATE: 10/10/2017
DRAWN BY: Bryan Moore
SHEET SIZE: A4

DRG NO: LCP008	REV NO. 0		SHEET 1 OF 1
DESCRIPTION: Parts List for Assembly Drawings			
MATERIAL:			
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SCALE: 1:2

NOTES:
SURFACE FINISH $\sqrt{3.2}$ UNLESS
OTHERWISE STATED.
DIMENSIONS IN mm.

APPROVED:

CHECKED:

DATE: 01/10/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A3

DRG NO: LCP009

REV NO. 0



SHEET 1 OF 1

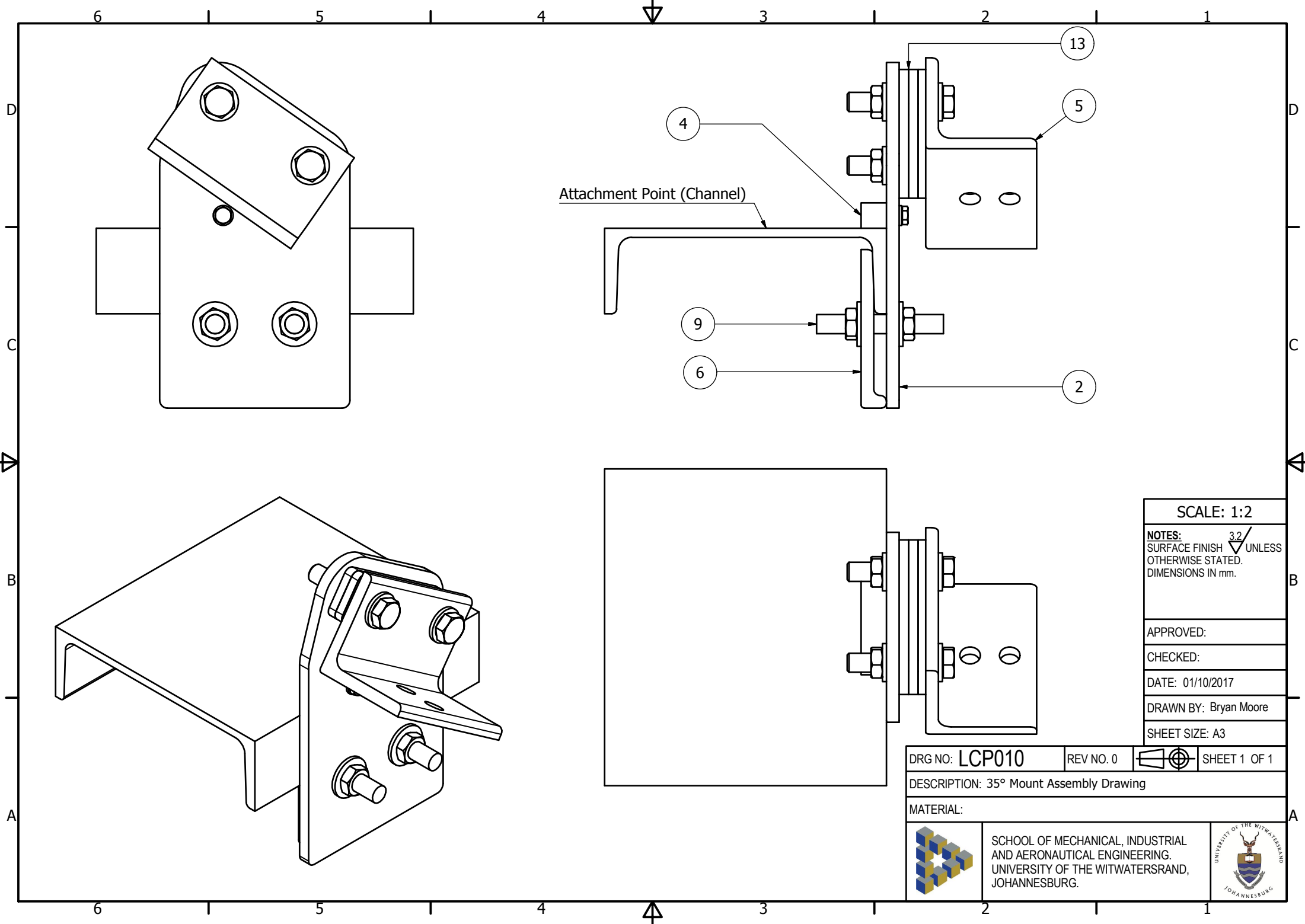
DESCRIPTION: 0° Mount Assembly Drawing

MATERIAL:



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6

5

4

3

2

1

D

D

C

C

B

B

A

A

Attachment Point (Channel)

4

13

5

9

6

2

SCALE: 1:2

NOTES:
SURFACE FINISH $\sqrt{3.2}$ UNLESS
OTHERWISE STATED.
DIMENSIONS IN mm.

APPROVED:

CHECKED:

DATE: 01/10/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A3

DRG NO: LCP010

REV NO. 0



SHEET 1 OF 1

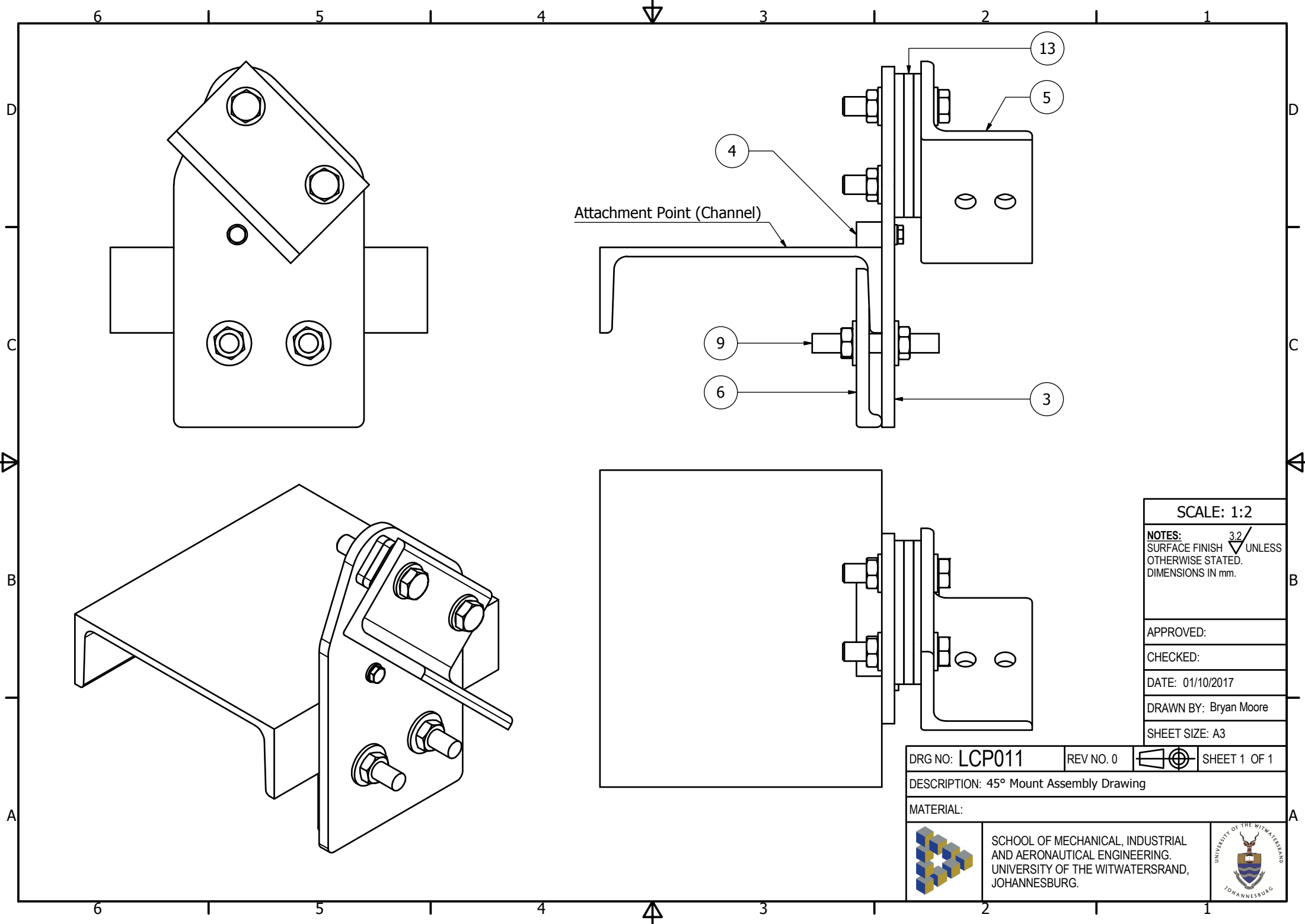
DESCRIPTION: 35° Mount Assembly Drawing

MATERIAL:



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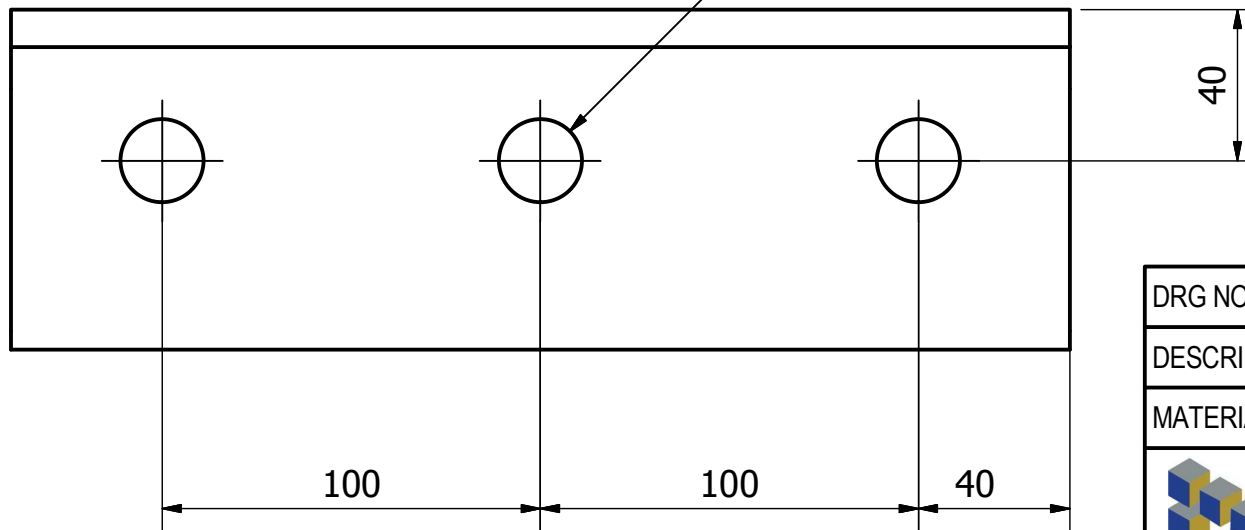
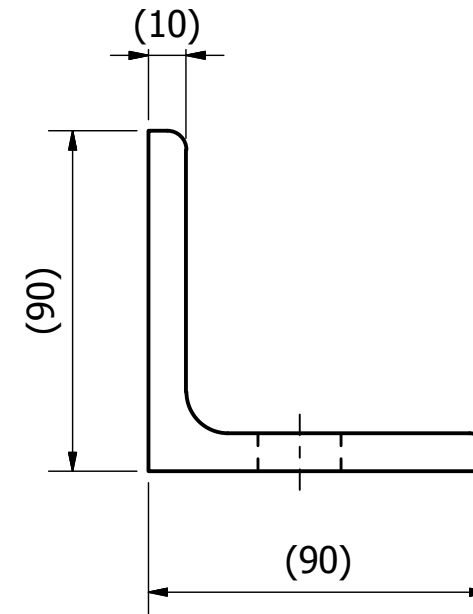
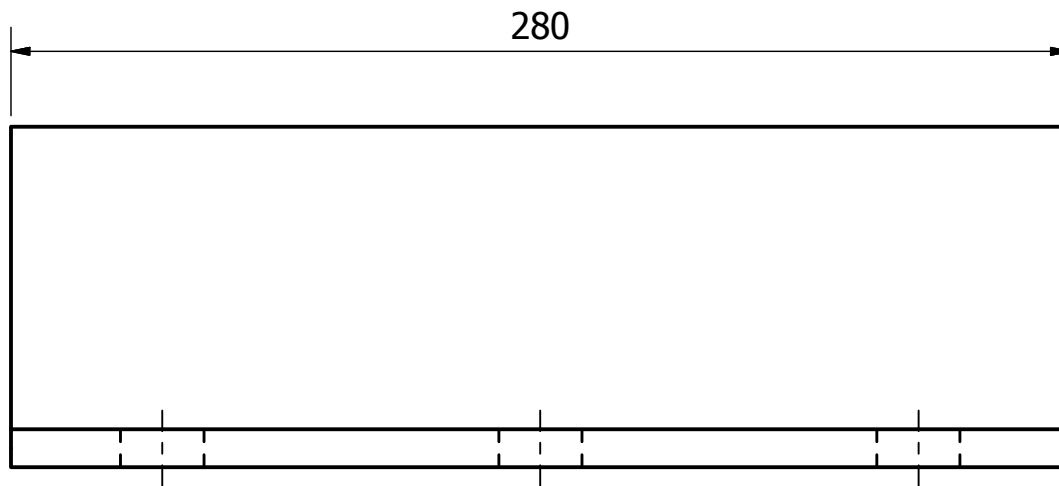
SCALE: 1:2	
NOTES: SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED. DIMENSIONS IN mm.	
APPROVED:	
CHECKED:	
DATE: 01/10/2017	
DRAWN BY: Bryan Moore	
SHEET SIZE: A3	

DRG NO: LCP011	REV NO. 0		SHEET 1 OF 1
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DESCRIPTION: 45° Mount Assembly Drawing

MATERIAL:

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Ø22 THRU
3 PLCS

SCALE: 1:2

NOTES:
SURFACE FINISH $\sqrt{3.2}$ UNLESS
OTHERWISE STATED.
DIMENSIONS IN mm.
DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 09/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL001

REV NO. 3



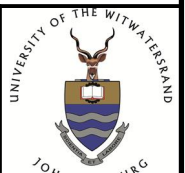
SHEET 1 OF 1

DESCRIPTION: Clamping Angle Iron Manufacturing Drawing

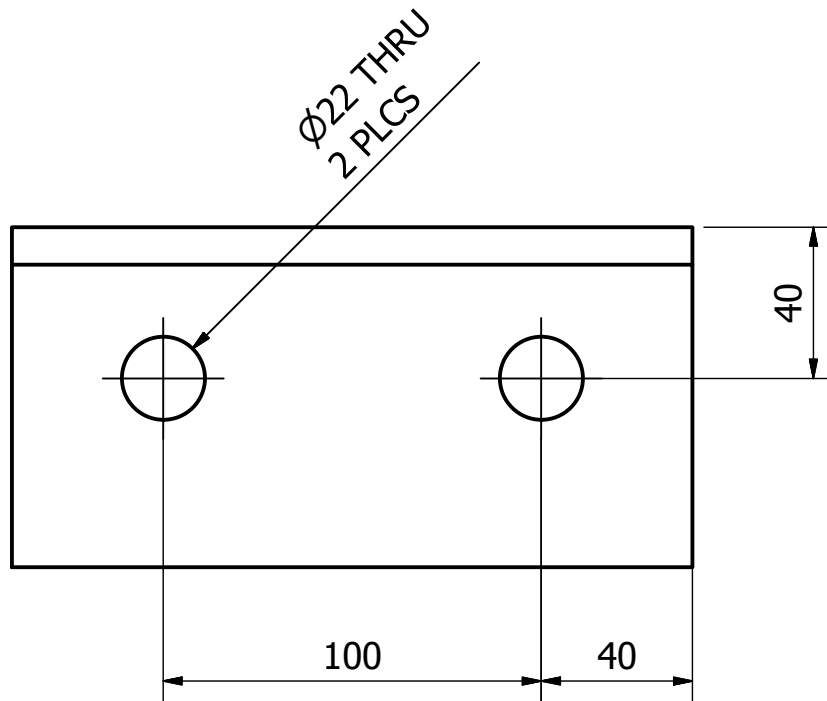
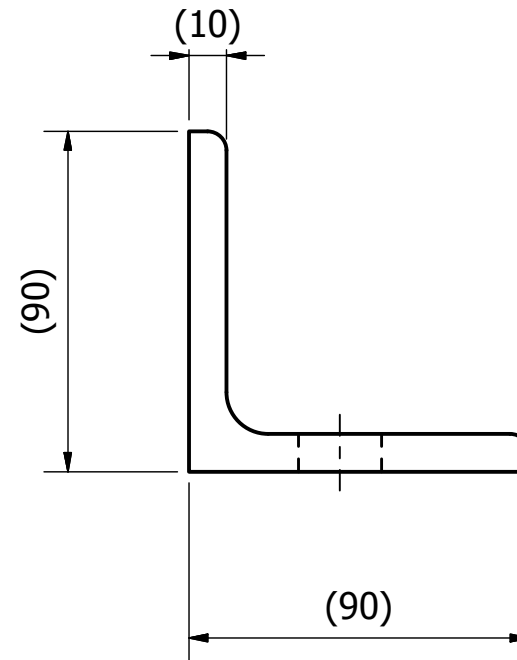
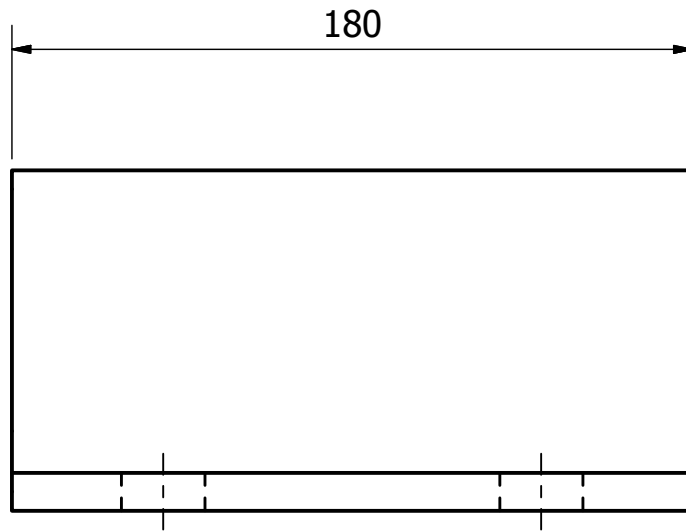
MATERIAL: 90 x 90 x 10 mm Equal Angle SANS 50025 Grade S355JR Steel



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QTY 6 OFF



SCALE: 1:2

NOTES: 3.2/✓
SURFACE FINISH UNLESS
OTHERWISE STATED.
DIMENSIONS IN mm.
DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 09/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL002

REV NO. 3



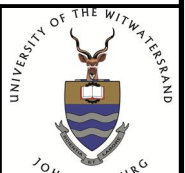
SHEET 1 OF 1

DESCRIPTION: Short Clamping Angle Iron Manufacturing Drawing

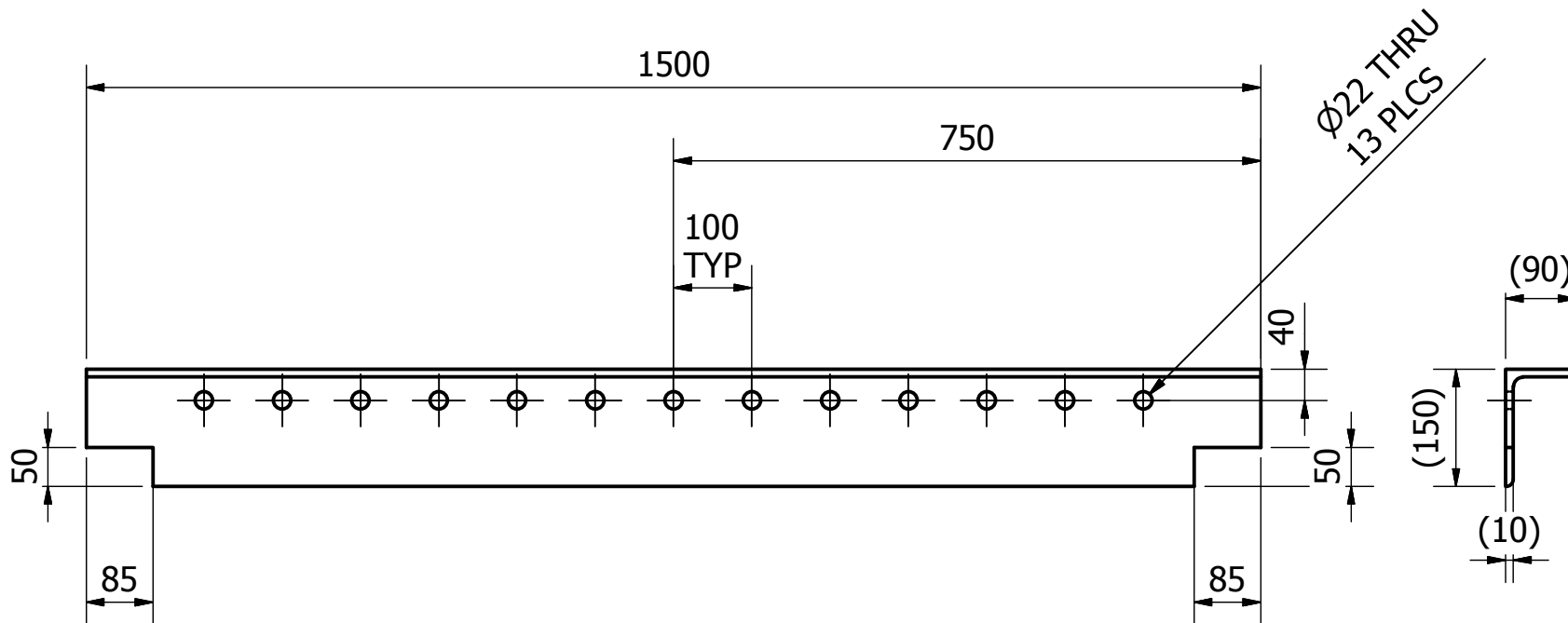
MATERIAL: 90 x 90 x 10 mm Equal Angle SANS 50025 Grade S355JR Steel



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QTY 4 OFF



SCALE: 1:9

NOTES:
 SURFACE FINISH $\checkmark$ 3.2 UNLESS OTHERWISE STATED.
 DIMENSIONS IN mm.
 DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 09/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL003

REV NO. 3



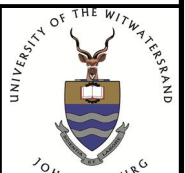
SHEET 1 OF 1

DESCRIPTION: 0° Belt Support Angle Iron Manufacturing Drawing

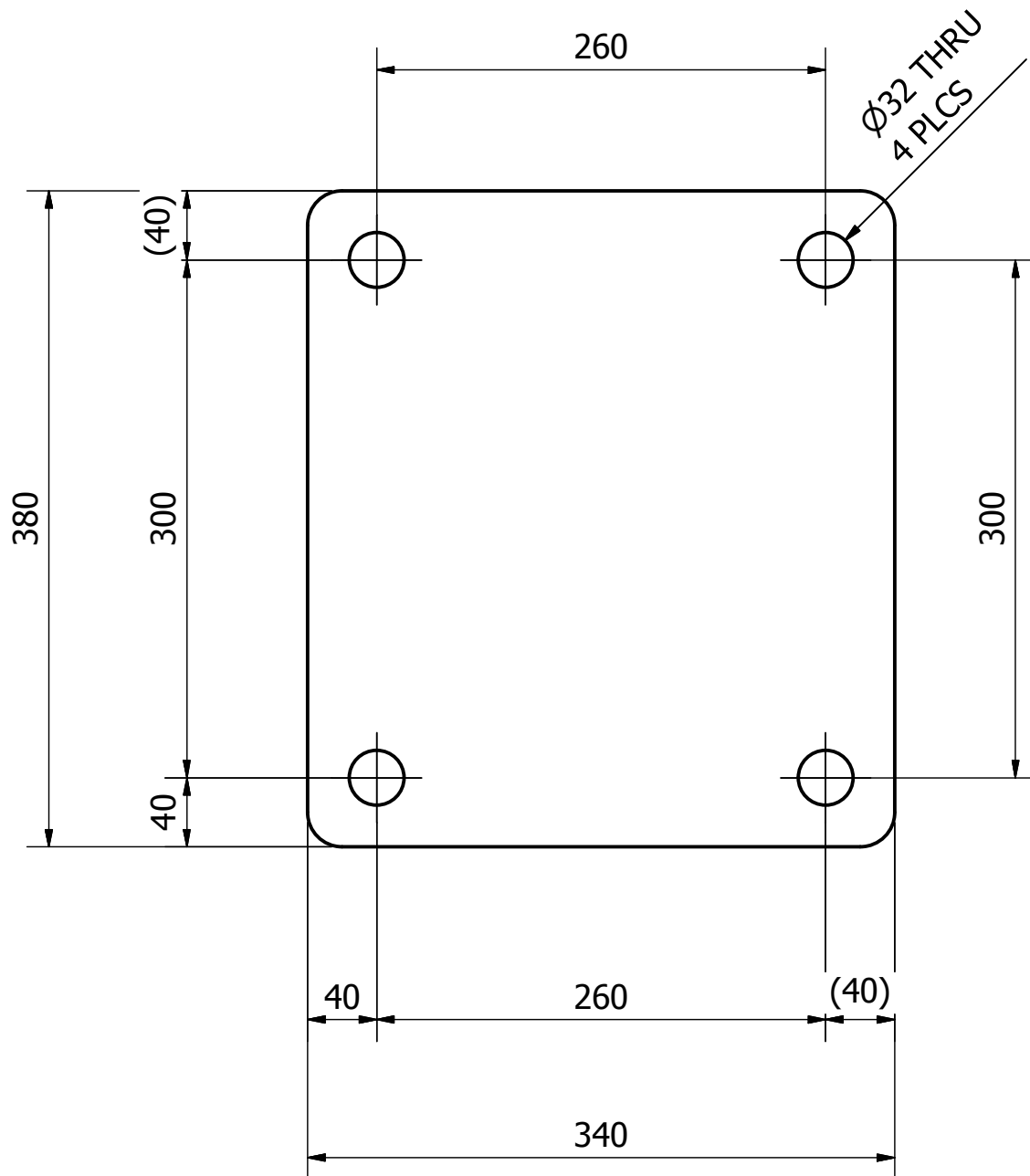
MATERIAL: 150 x 90 x 10 mm Unequal Angle SANS 50025 Grade S355JR Steel



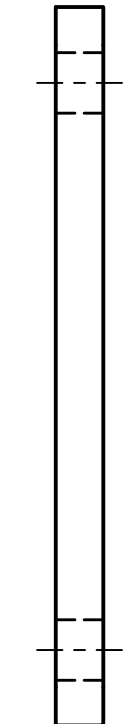
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QTY 2 OFF



(25)



SCALE: 1:4

NOTES:
SURFACE FINISH $\sqrt{3.2}$ UNLESS
OTHERWISE STATED.
DIMENSIONS IN mm.
DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 09/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL004

REV NO. 3



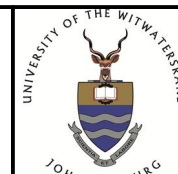
SHEET 1 OF 1

DESCRIPTION: Mounting Plate Base Plate Manufacturing Drawing

MATERIAL: 340 x 25 mm PLT x 380 LG



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5 X 45°



R20
2 PLCS

(10)

SCALE: 1:4

NOTES:
SURFACE FINISH $\checkmark$ 3.2 UNLESS
OTHERWISE STATED.
DIMENSIONS IN mm.
DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 09/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL005

REV NO. 3



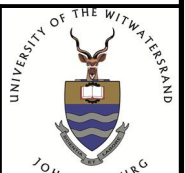
SHEET 1 OF 2

DESCRIPTION: 0° Mounting Plate Drawings

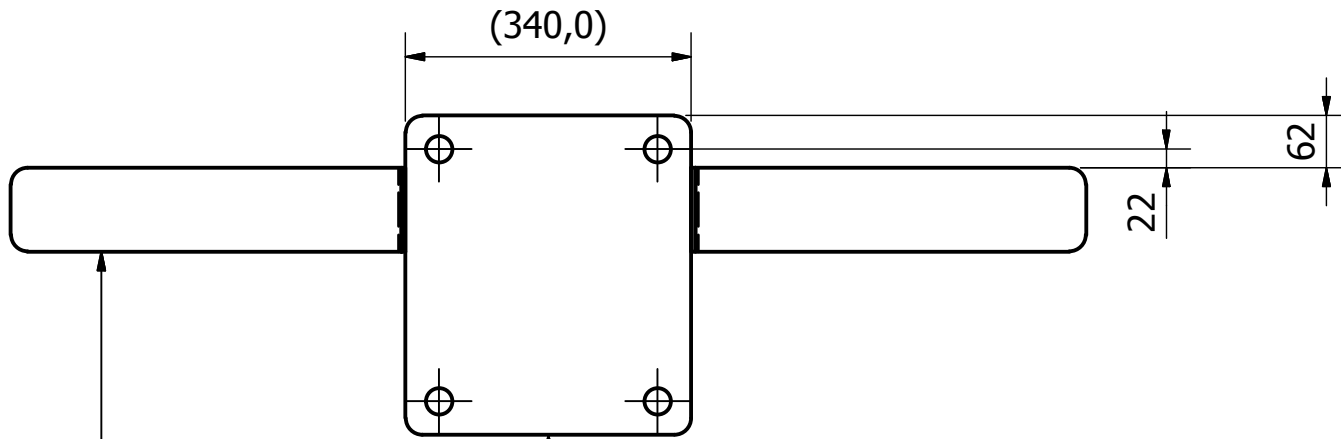
MATERIAL: 100 x 10 mm PLT x 470 LG



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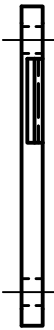
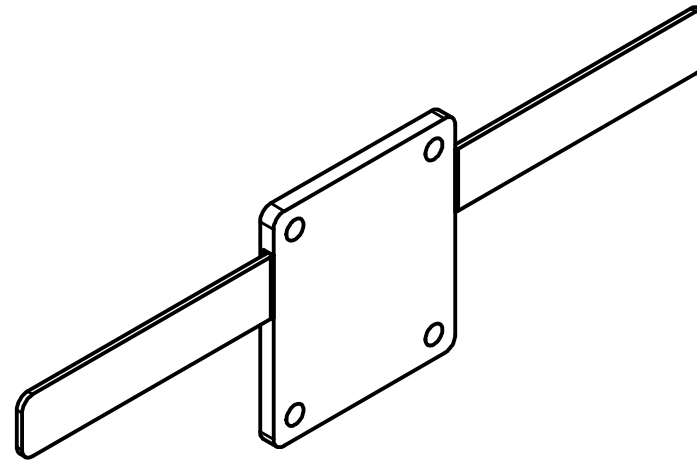


BCL005-1 : QTY 4 OFF



BCL005-1

BCL004



SCALE: 1:9

NOTES:
 SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
 DIMENSIONS IN mm.
 DO NOT SCALE.

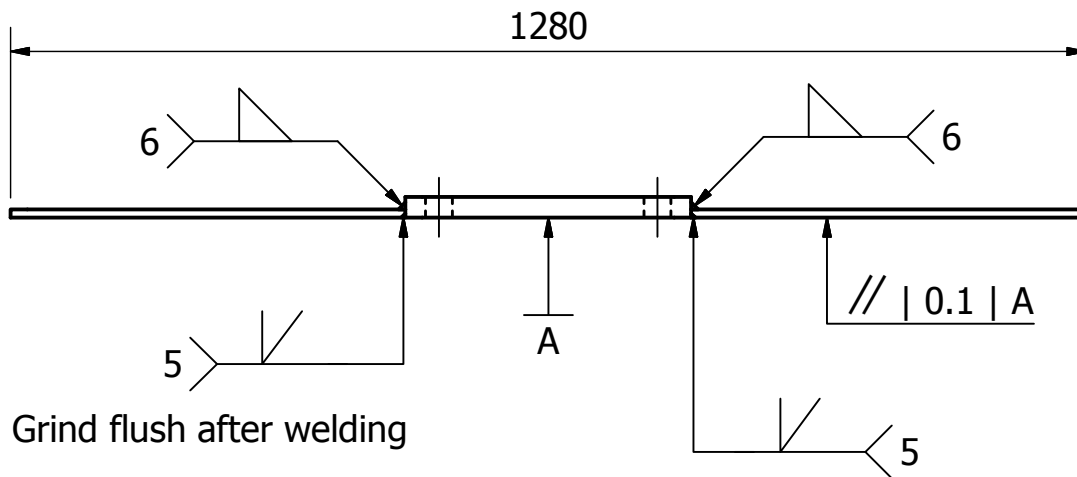
APPROVED:

CHECKED:

DATE: 09/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4



Grind flush after welding

Grind flush after welding

ASSEMBLY : QTY 2 OFF

DRG NO: BCL005

REV NO. 3



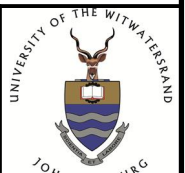
SHEET 2 OF 2

DESCRIPTION: 0° Mounting Plate Drawings

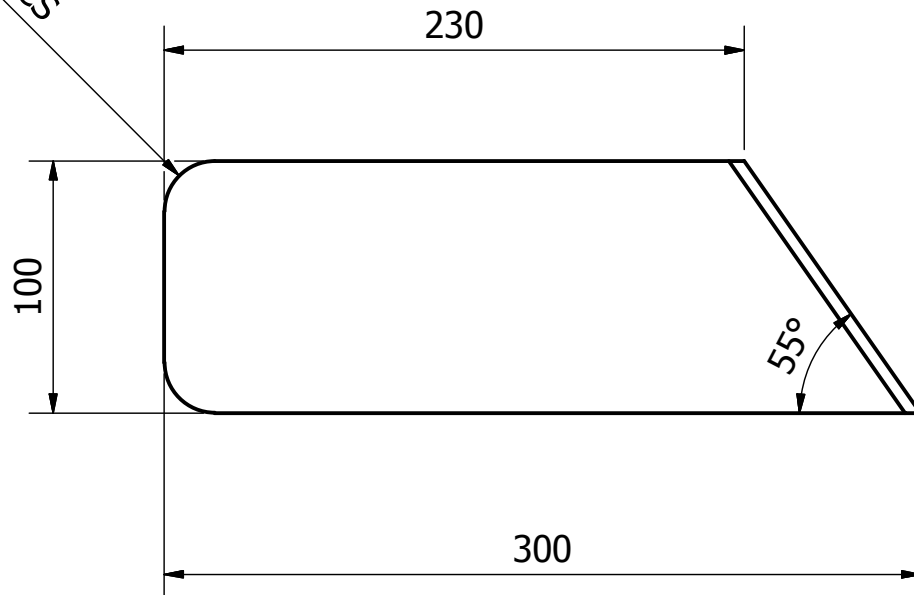
MATERIAL: As per components



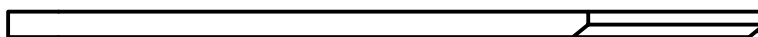
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R20
2 PLCS



(10)



5 X 45°

SCALE: 1:3

NOTES:
SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
DIMENSIONS IN mm.
DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 09/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL006

REV NO. 3



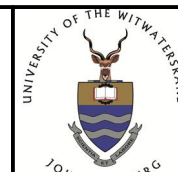
SHEET 1 OF 2

DESCRIPTION: 900, 35° Mounting Plate Drawings

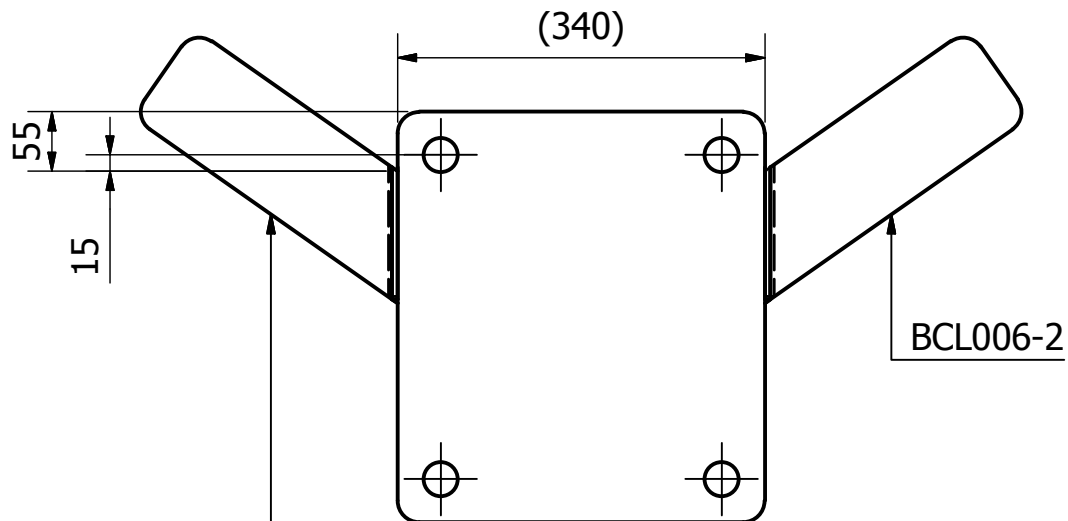
MATERIAL: 355JR x 10 mm Steel Plate



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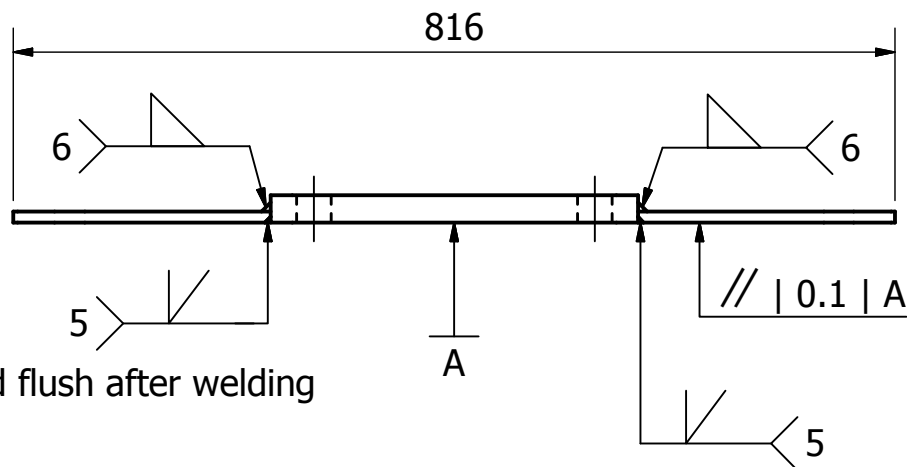
BCL006-1 : QTY 2 AS DRAW
BCL006-2 : QTY 2 TO OPP HAND



BCL006-1

BCL004

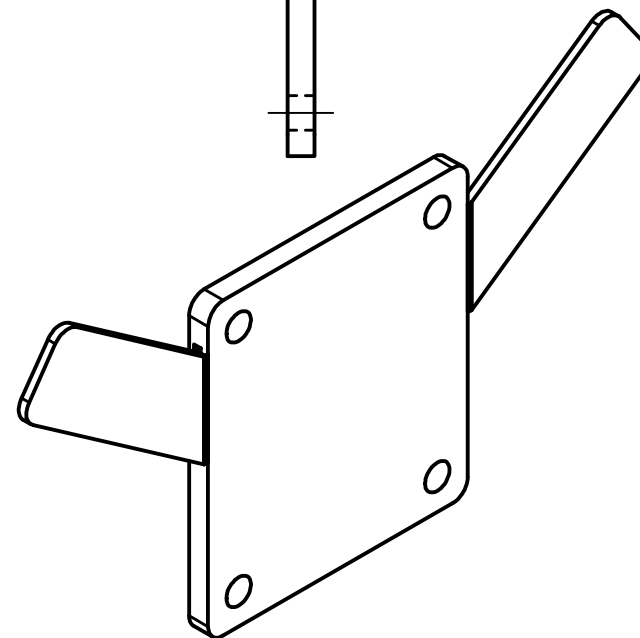
BCL006-2



Grind flush after welding

Grind flush after welding

ASSEMBLY : QTY 2 OFF



SCALE: 1:7

NOTES:
SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
DIMENSIONS IN mm.
DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 09/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL006

REV NO. 3



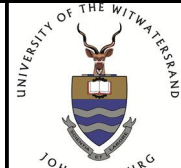
SHEET 2 OF 2

DESCRIPTION: 900, 35° Mounting Plate Drawings

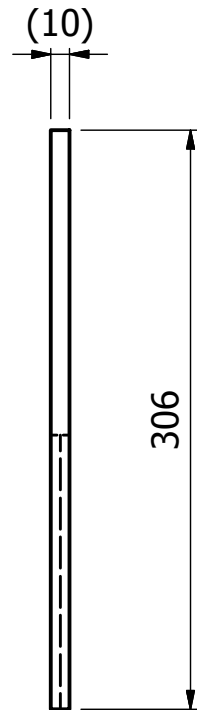
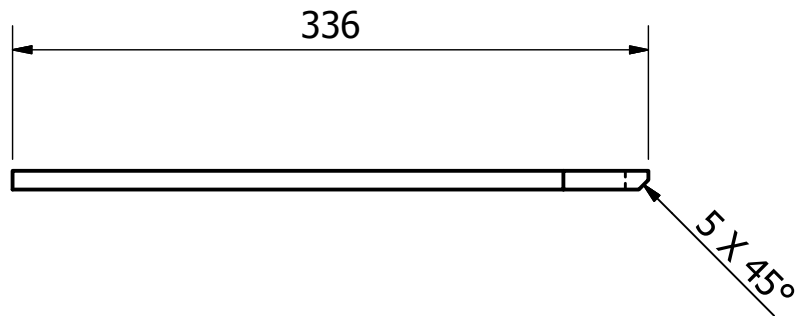
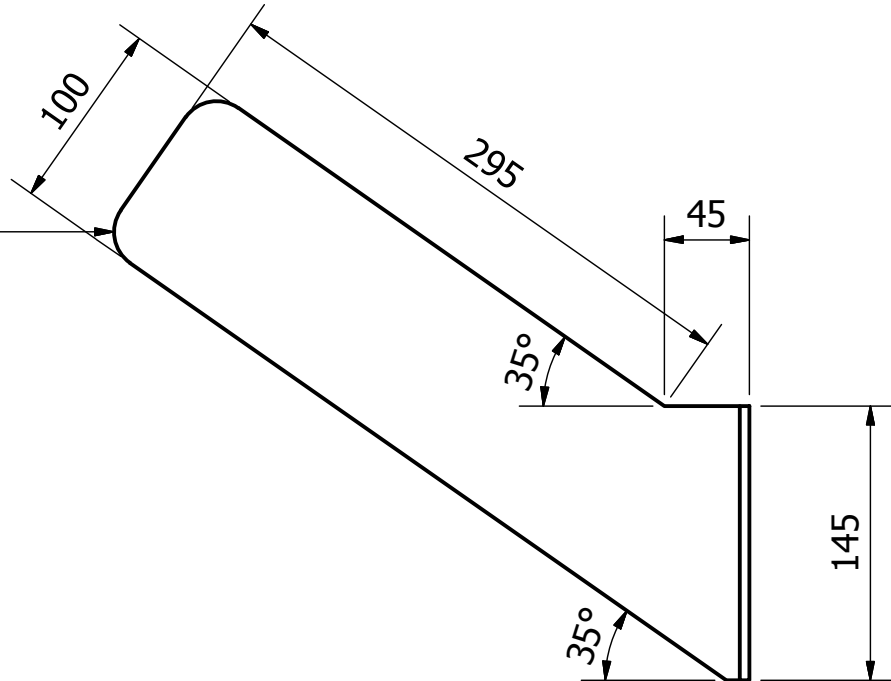
MATERIAL: As per components



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R20
2 PLCS



SCALE: 1:4

NOTES:
SURFACE FINISH $\checkmark$ 3.2 UNLESS OTHERWISE STATED.
DIMENSIONS IN mm.
DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 09/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL007

REV NO. 3



SHEET 1 OF 2

DESCRIPTION: 1200, 35° Mounting Plate Drawings

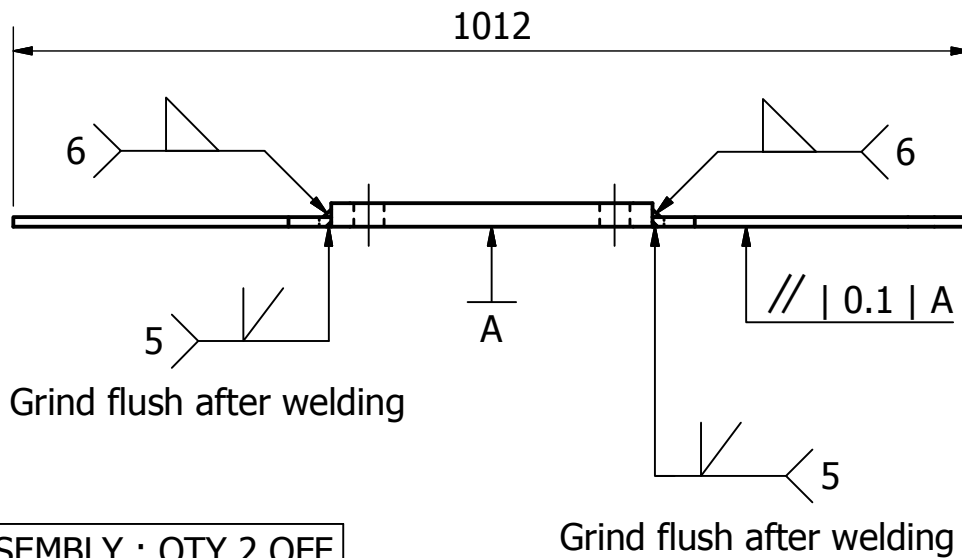
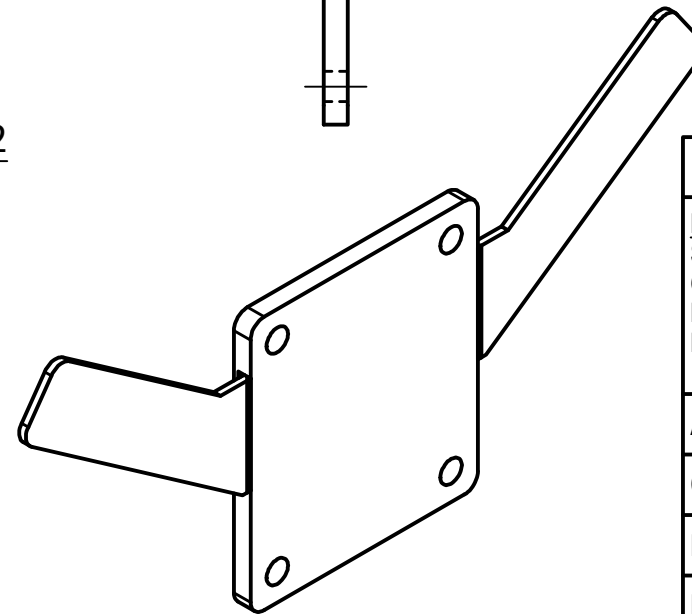
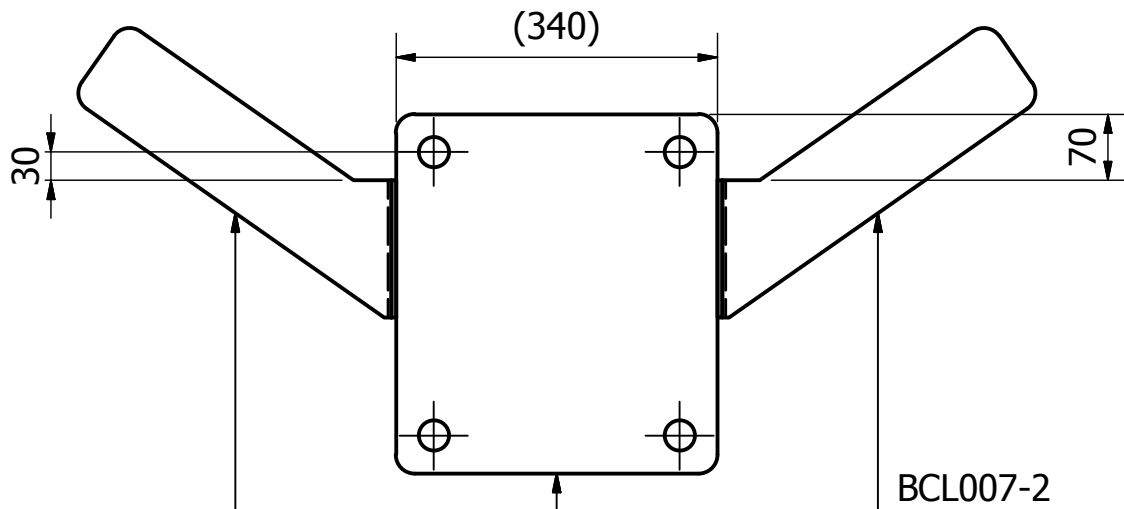
MATERIAL: 355JR x 10 mm Steel Plate



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BCL007-1 : QTY 2 AS DRAW
BCL007-2 : QTY 2 TO OPP HAND



ASSEMBLY : QTY 2 OFF

SCALE: 1:8

NOTES:
 SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
 DIMENSIONS IN mm.
 DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 10/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL007

REV NO. 3



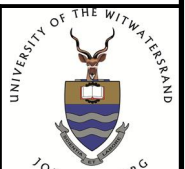
SHEET 2 OF 2

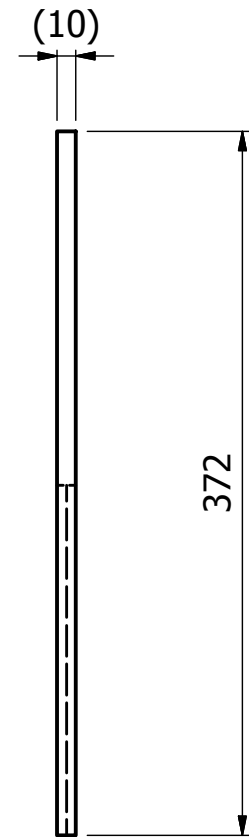
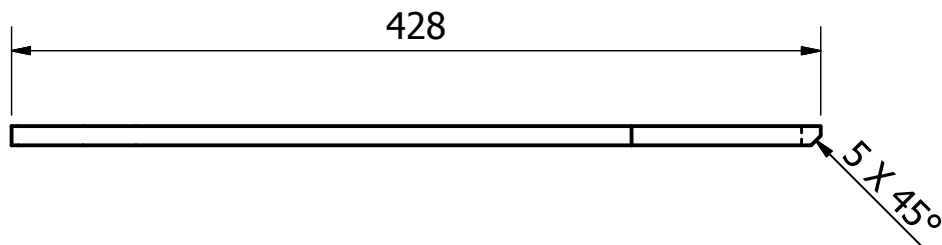
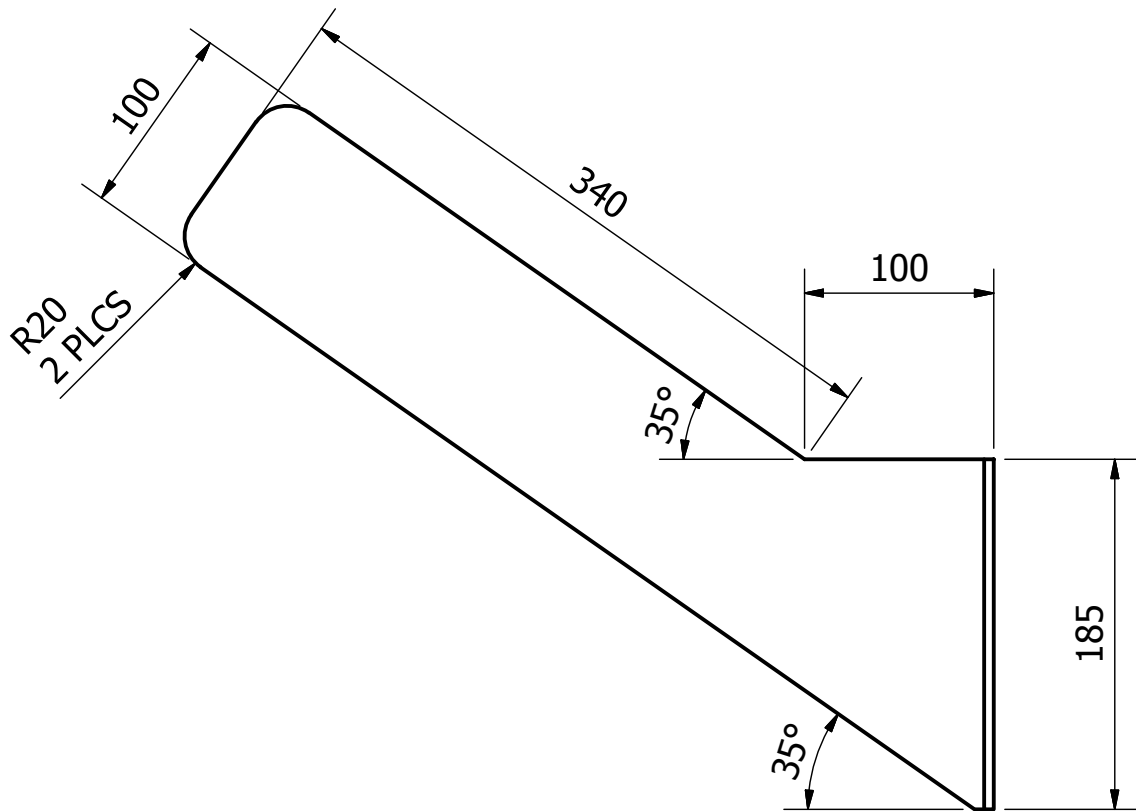
DESCRIPTION: 1200, 35° Mounting Plate Drawings

MATERIAL: As per components



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SCALE: 1:4

NOTES:
 SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
 DIMENSIONS IN mm.
 DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 10/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL008

REV NO. 3



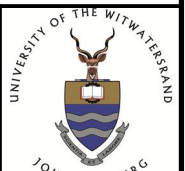
SHEET 1 OF 2

DESCRIPTION: 1500, 35° Mounting Plate Drawings

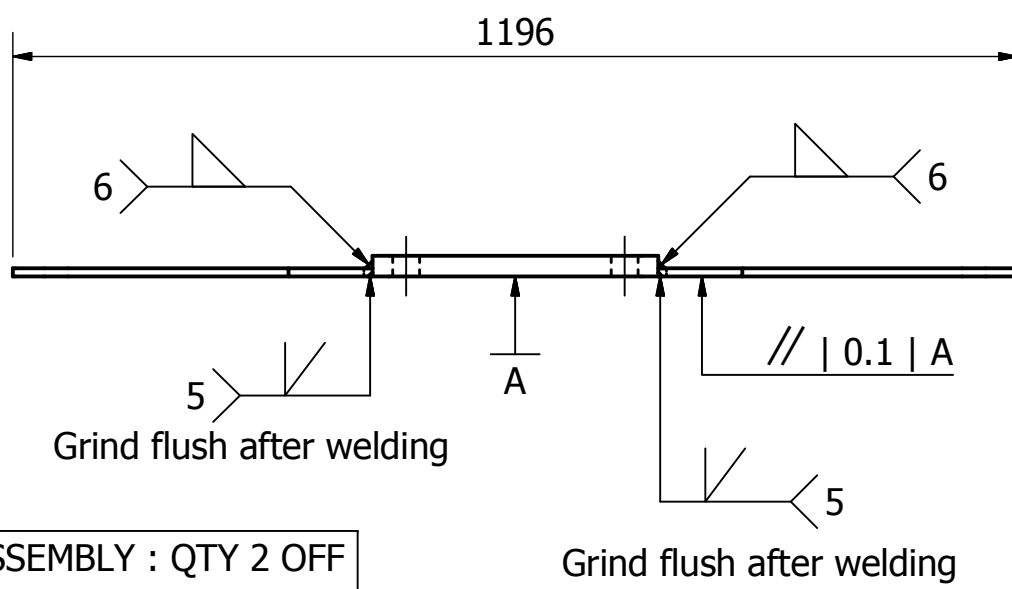
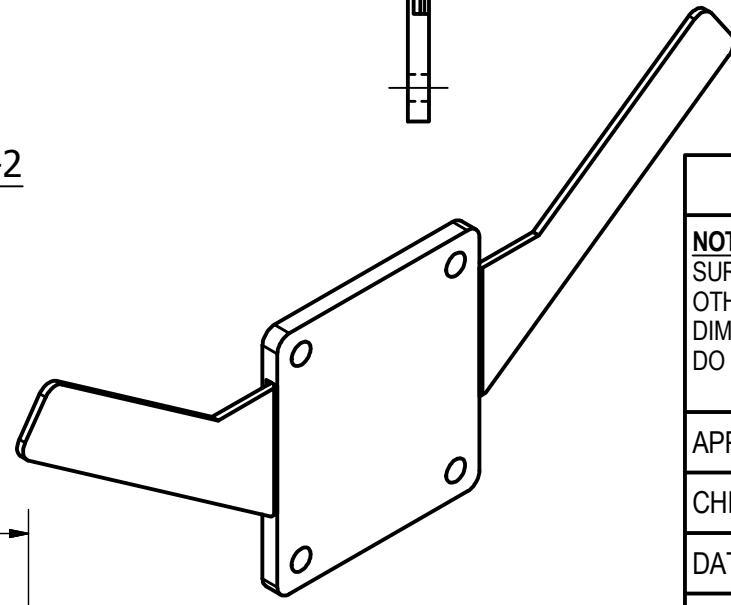
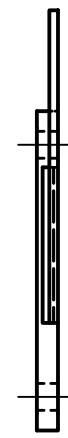
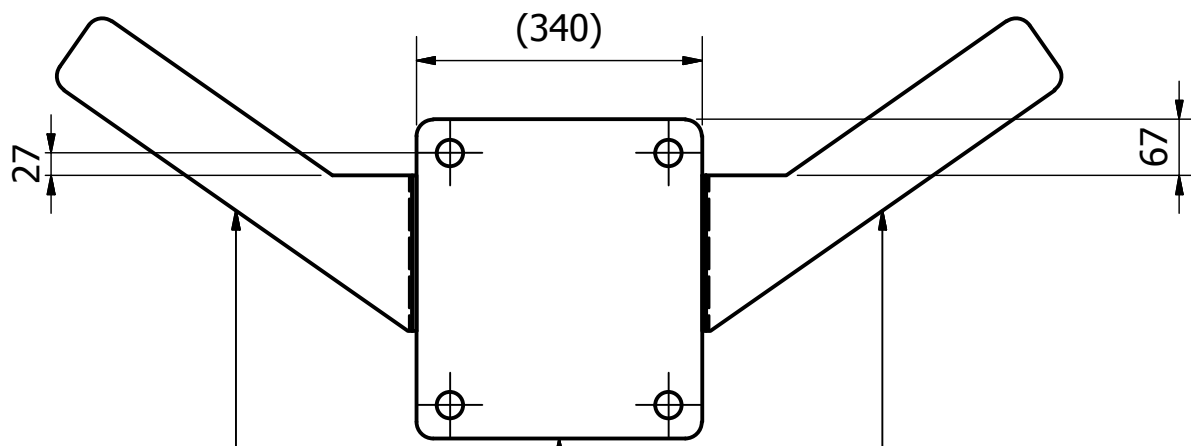
MATERIAL: 355JR x 10 mm Steel Plate



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BCL008-1 : QTY 2 AS DRAW
 BCL008-2 : QTY TO OPP HAND



SCALE: 1:9

NOTES:
 SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
 DIMENSIONS IN mm.
 DO NOT SCALE.

APPROVED:
 CHECKED:
 DATE: 10/11/2017
 DRAWN BY: Bryan Moore
 SHEET SIZE: A4

DRG NO: BCL008	REV NO. 3		SHEET 2 OF 2
DESCRIPTION: 1500, 35° Mounting Plate Drawings			
MATERIAL: As per components			
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ASSEMBLY : QTY 2 OFF

R20
2 PLCS

230

100

330

45°

(10)

SCALE: 1:3

NOTES:
SURFACE FINISH $\checkmark$ 3.2 UNLESS
OTHERWISE STATED.
DIMENSIONS IN mm.
DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 10/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL009

REV NO. 3



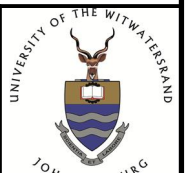
SHEET 1 OF 2

DESCRIPTION: 900, 45° Mounting Plate Drawings

MATERIAL: 355JR x 10 mm Steel Plate

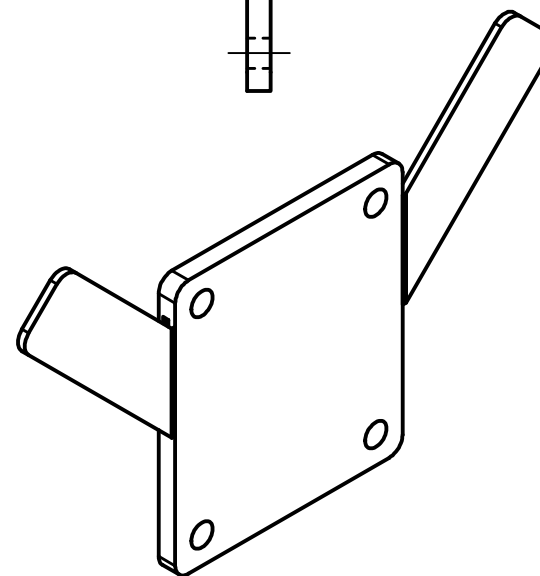
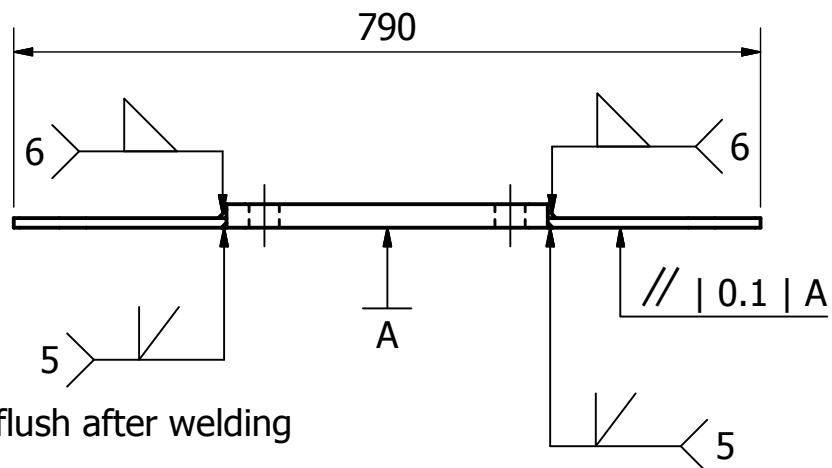
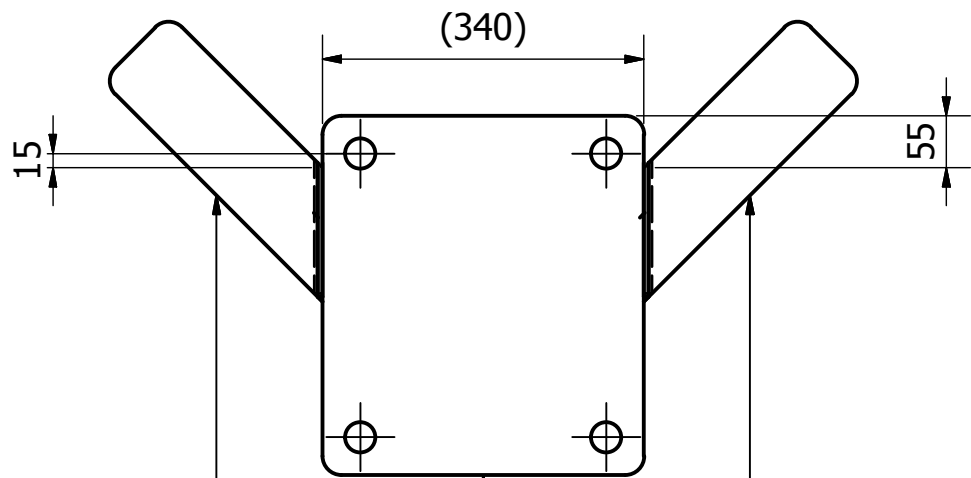


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BCL009-1 : QTY 2 AS DRAW
BCL009-2 : QTY 2 TO OPP HAND

5 X 45°



SCALE: 1:8

NOTES:
 SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
 DIMENSIONS IN mm.
 DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 10/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL009

REV NO. 3



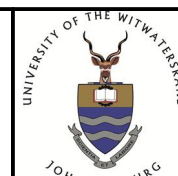
SHEET 2 OF 2

DESCRIPTION: 900, 45° Mounting Plate Drawings

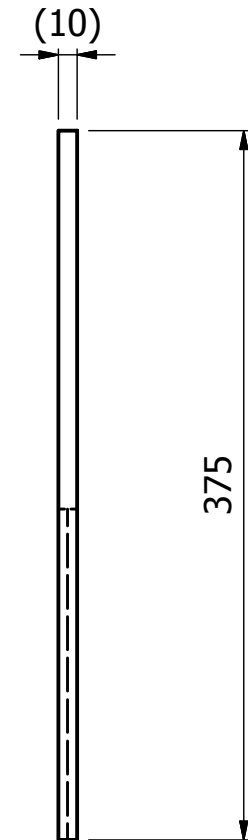
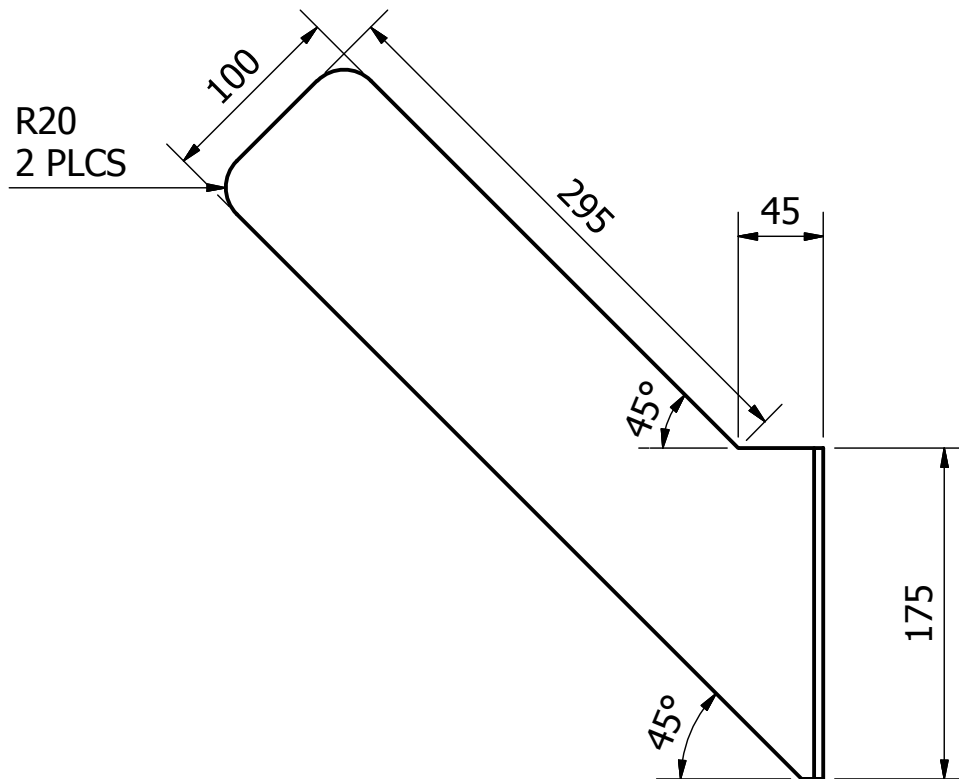
MATERIAL: As per components



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ASSEMBLY : QTY 2 OFF



SCALE: 1:4

NOTES:
SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
DIMENSIONS IN mm.
DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 10/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL010

REV NO. 3



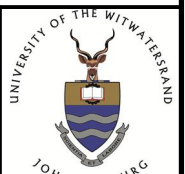
SHEET 1 OF 2

DESCRIPTION: 1200, 45° Mounting Plate Drawings

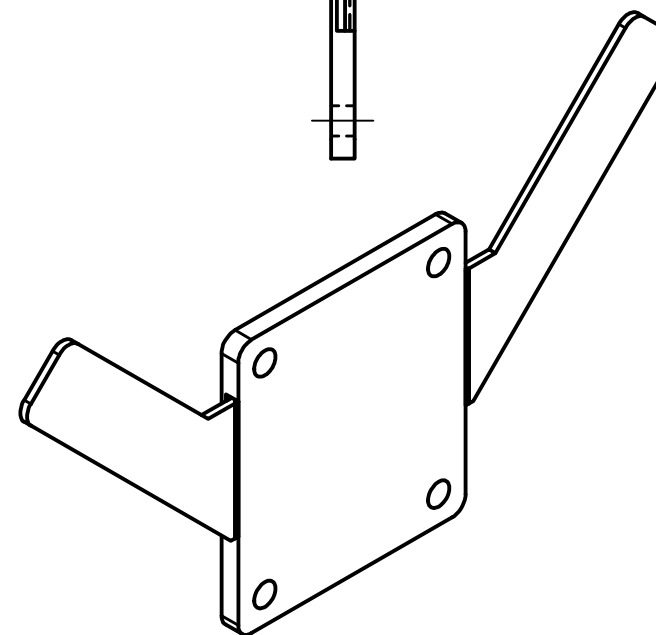
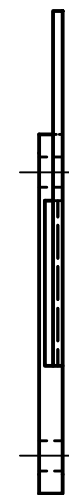
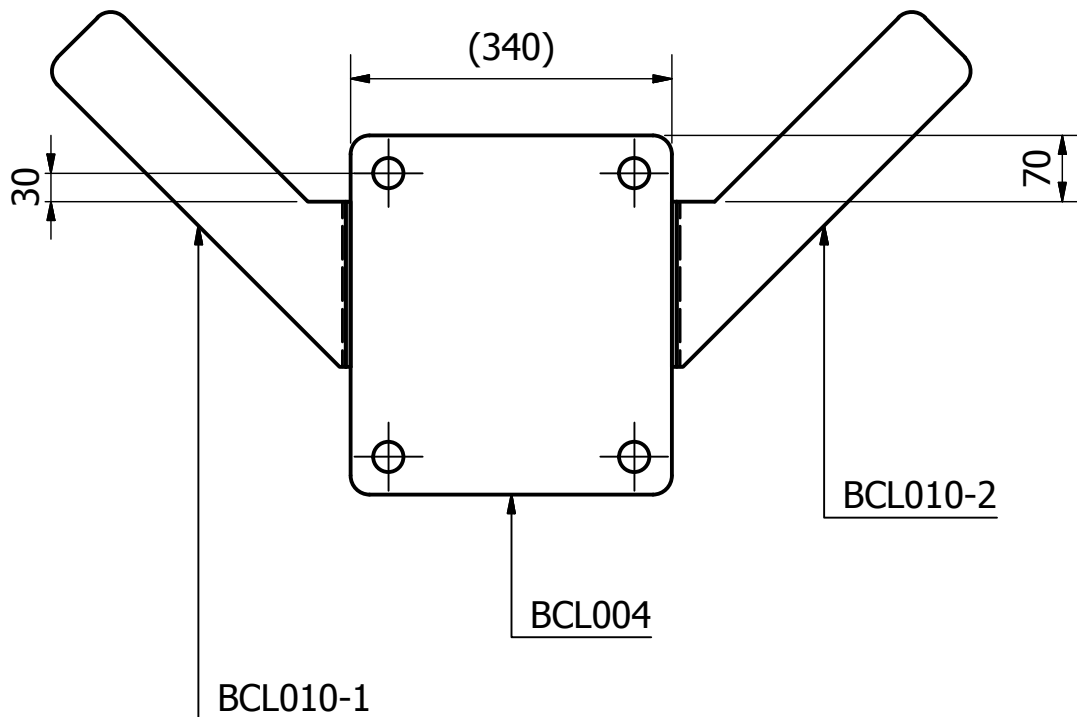
MATERIAL: 355JR x 10 mm Steel Plate



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BCL010-1 : QTY 2 AS DRAW
BCL010-2 : QTY 2 TO OPP HAND



SCALE: 1:8

NOTES:
 SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
 DIMENSIONS IN mm.
 DO NOT SCALE.

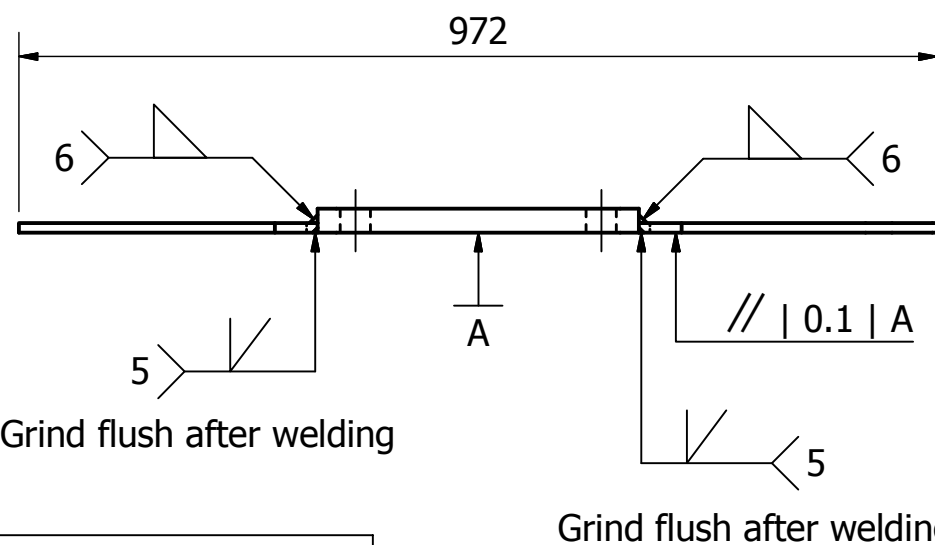
APPROVED:

CHECKED:

DATE: 10/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4



ASSEMBLY : QTY 2 OFF

DRG NO: BCL010

REV NO. 3



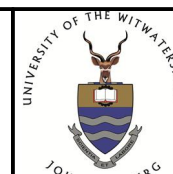
SHEET 2 OF 2

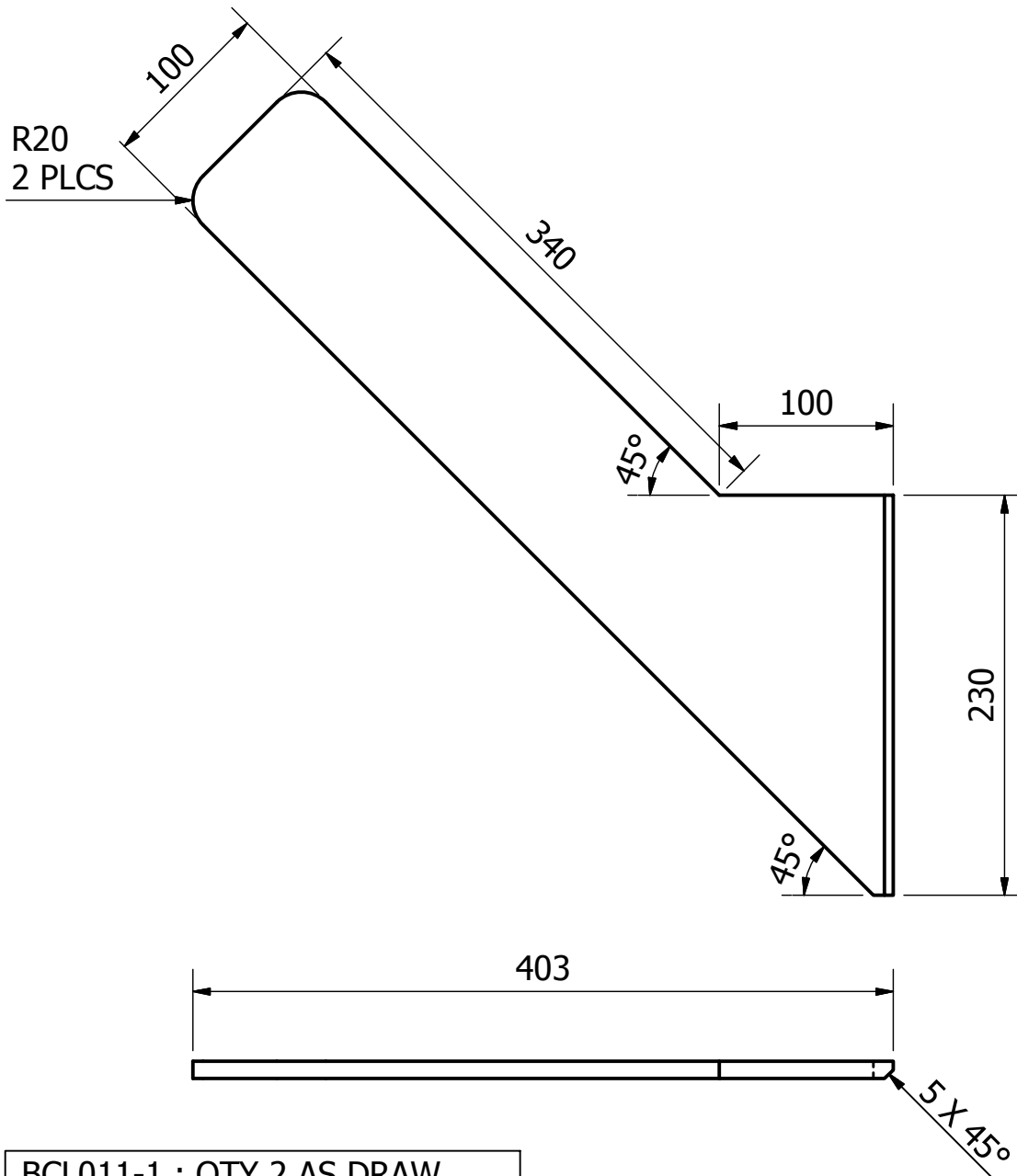
DESCRIPTION: 1200, 45° Mounting Plate Drawings

MATERIAL: As per components

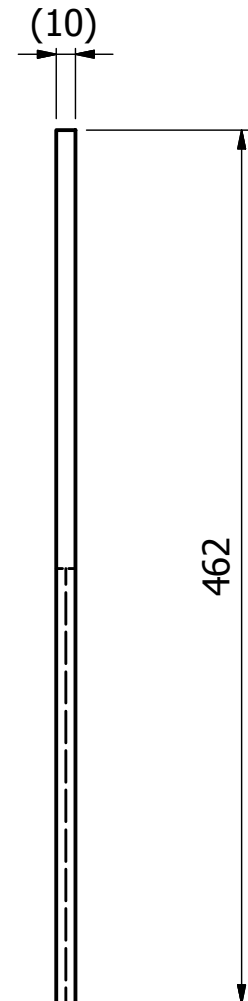


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BCL011-1 : QTY 2 AS DRAW
BCL011-2 : QTY 2 TO OPP HAND



SCALE: 1:4

NOTES:
SURFACE FINISH $\checkmark$ 3.2 UNLESS
OTHERWISE STATED.
DIMENSIONS IN mm.
DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 13/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL011

REV NO. 3



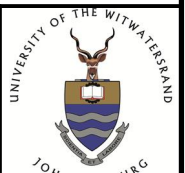
SHEET 1 OF 2

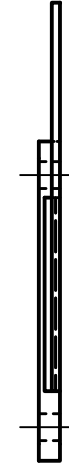
DESCRIPTION: 1500, 45° Mounting Plate Drawings

MATERIAL: 355JR x 10 mm Steel Plate



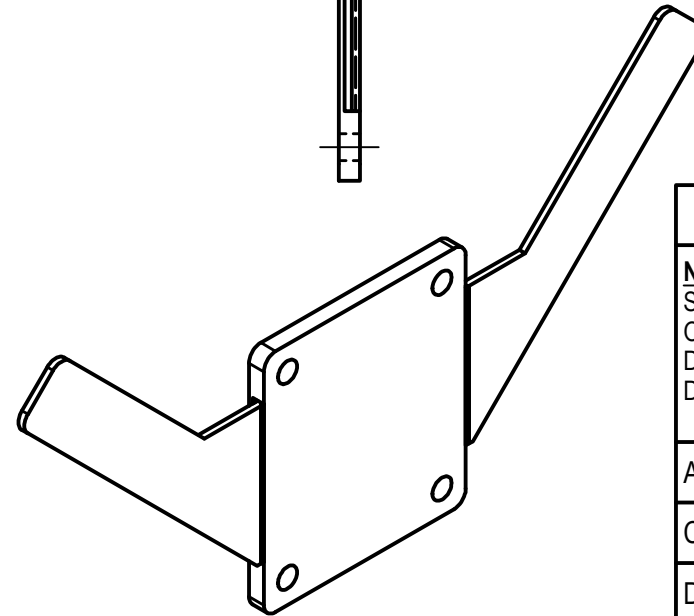
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BCL004

BCL011-1



NOTES:
SURFACE FINISH  UNLESS
OTHERWISE STATED.
DIMENSIONS IN mm.
DO NOT SCALE.

CHECKED:

DRAWN BY: Bryan Moore

SHEET SIZE: A4

REV NO. 3

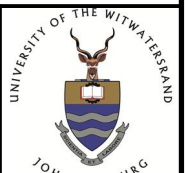


DESCRIPTION: 1500, 45° Mounting Plate Drawings

MATERIAL: As per components

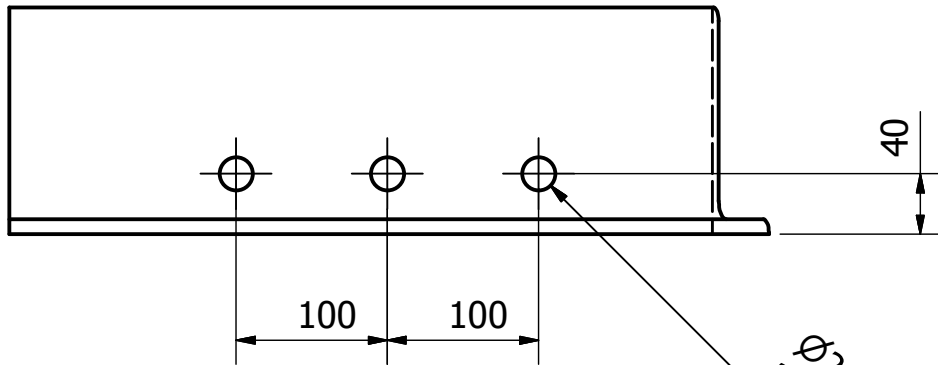


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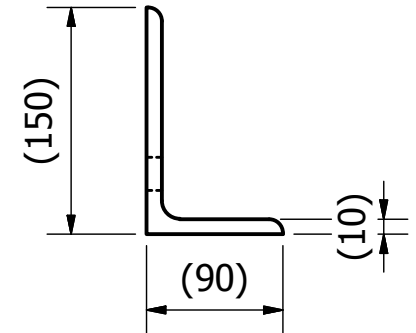


ASSEMBLY : QTY 2 OFF

Grind flush after welding



Ø22 THRU
3 PLCS



SCALE: 1:5

NOTES:
SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
DIMENSIONS IN mm.
DO NOT SCALE.

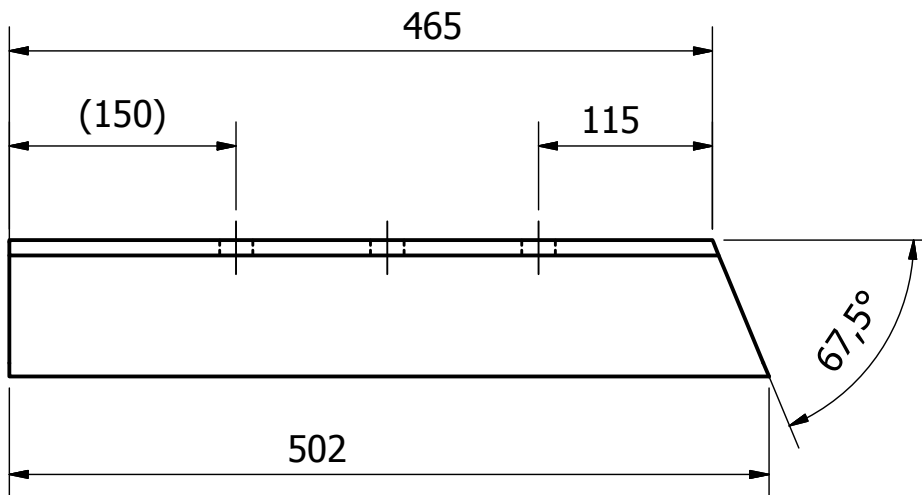
APPROVED:

CHECKED:

DATE: 13/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4



BCL012-1 : QTY 2 AS DRAW
BCL012-2 : QTY 2 TO OPP HAND

DRG NO: BCL012

REV NO. 3



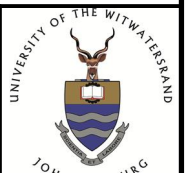
SHEET 1 OF 3

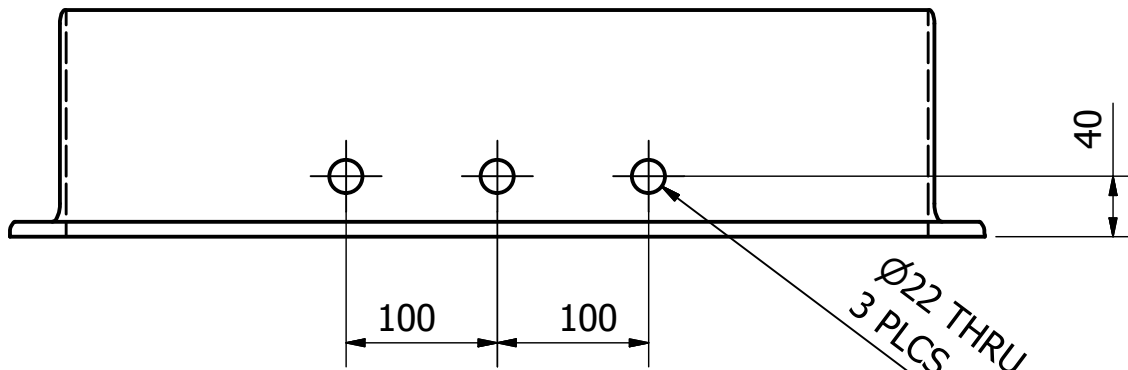
DESCRIPTION: 1500, 45° Support Angle Parts and Assembly Drawings

MATERIAL: 150 x 90 x 10 mm Unequal Angle SANS 50025 Grade S355JR Steel

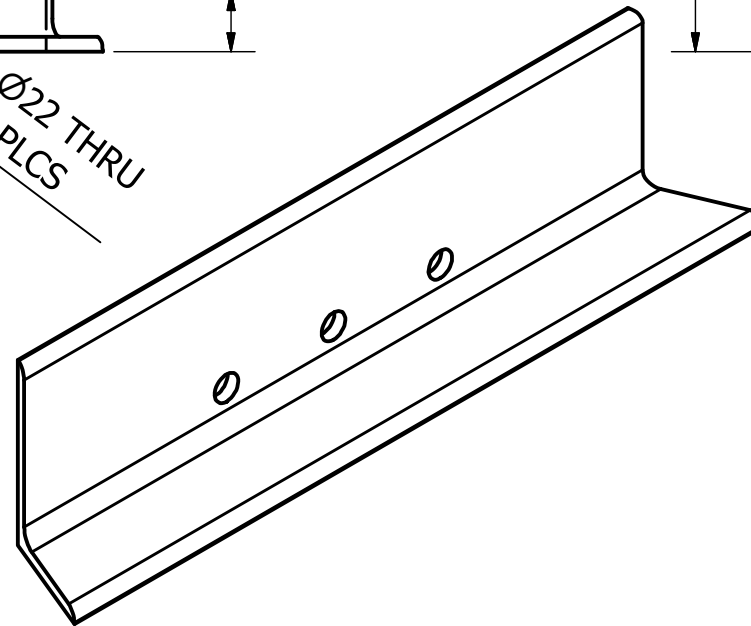
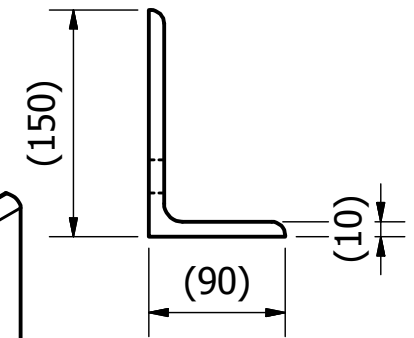


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Ø22 THRU
3 PLCS



SCALE: 1:5

NOTES:
SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
DIMENSIONS IN mm.
DO NOT SCALE.

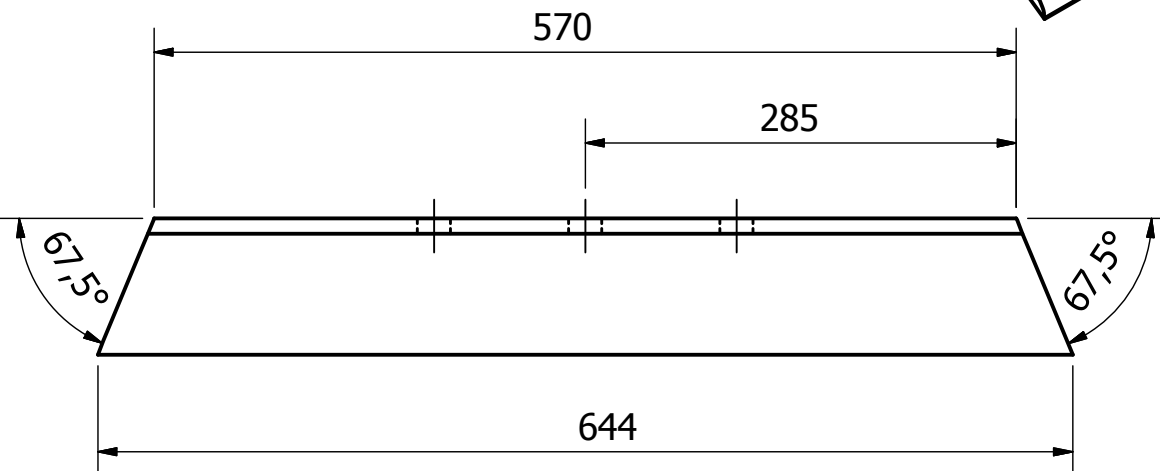
APPROVED:

CHECKED:

DATE: 13/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4



DRG NO: BCL012

REV NO. 3



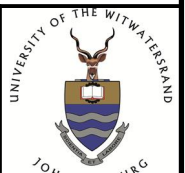
SHEET 2 OF 3

DESCRIPTION: 1500, 45° Support Angle Parts and Assembly Drawings

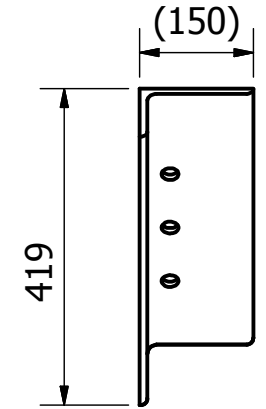
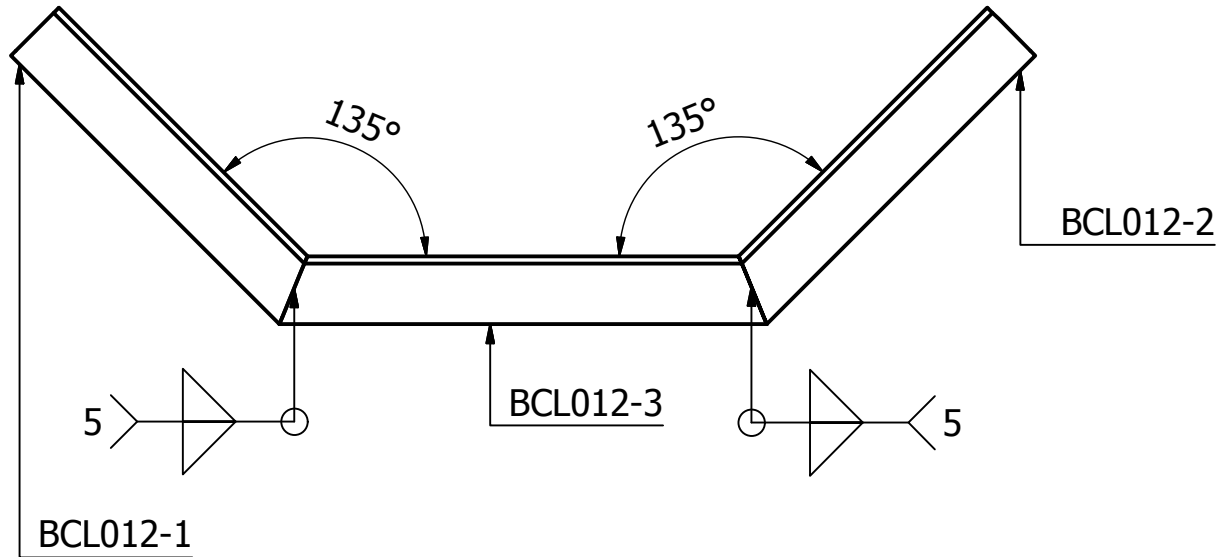
MATERIAL: 150 x 90 x 10 mm Unequal Angle SANS 50025 Grade S355JR Steel



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BCL012-3 : QTY 2 OFF



SCALE: 1:10

NOTES:
 SURFACE FINISH $\checkmark$ 3.2 UNLESS OTHERWISE STATED.
 DIMENSIONS IN mm.
 DO NOT SCALE.

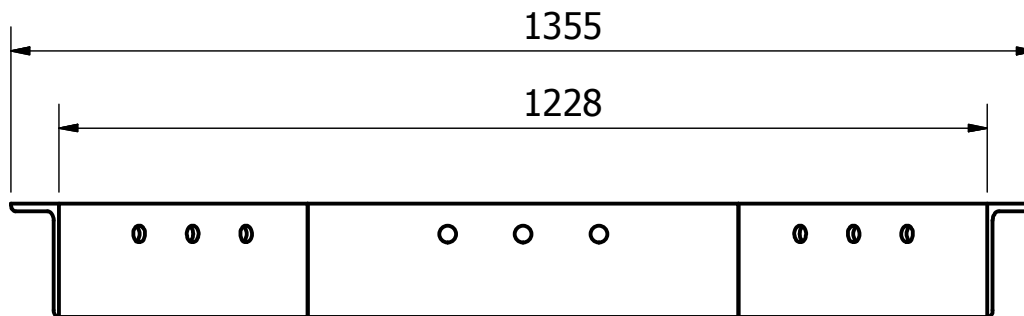
APPROVED:

CHECKED:

DATE: 13/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4



ASSEMBLY : QTY 2 OFF

DRG NO: BCL012

REV NO. 3



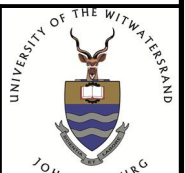
SHEET 3 OF 3

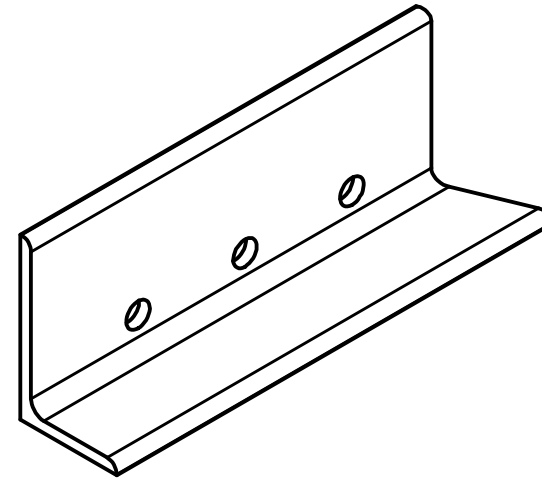
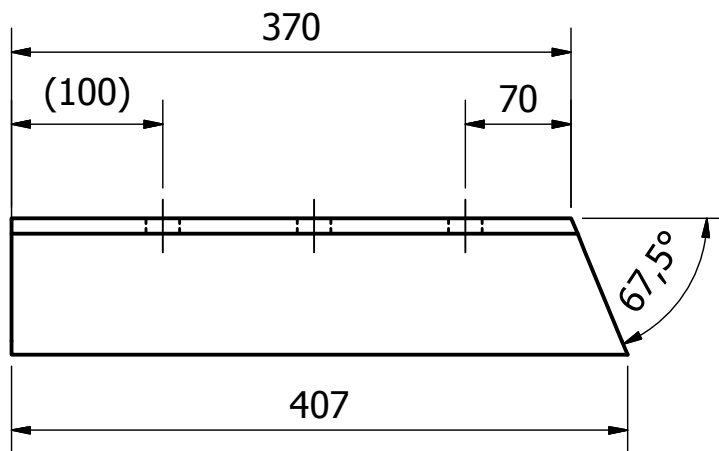
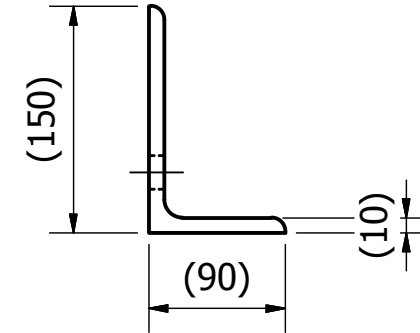
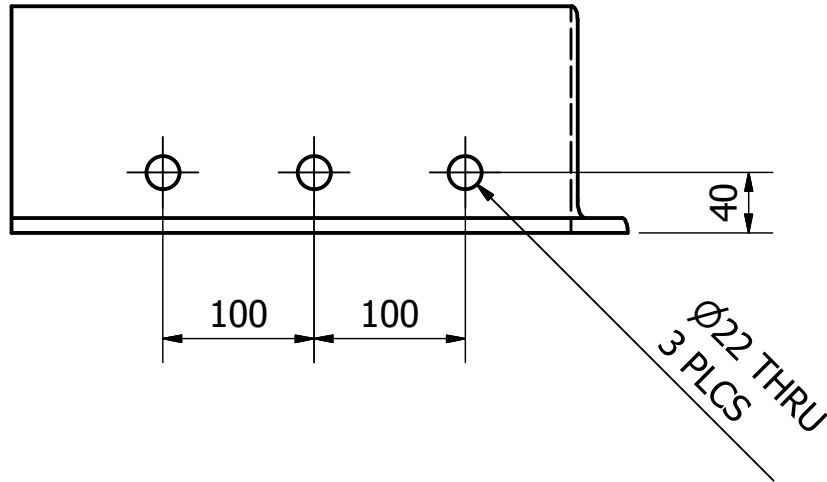
DESCRIPTION: 1500, 45° Support Angle Parts and Assembly Drawings

MATERIAL: 150 x 90 x 10 mm Unequal Angle SANS 50025 Grade S355JR Steel



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SCALE: 1:5

NOTES:
SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
DIMENSIONS IN mm.
DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 13/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL013

REV NO. 3



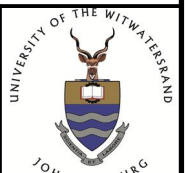
SHEET 1 OF 3

DESCRIPTION: 1200, 45° Support Angle Parts and Assembly Drawings

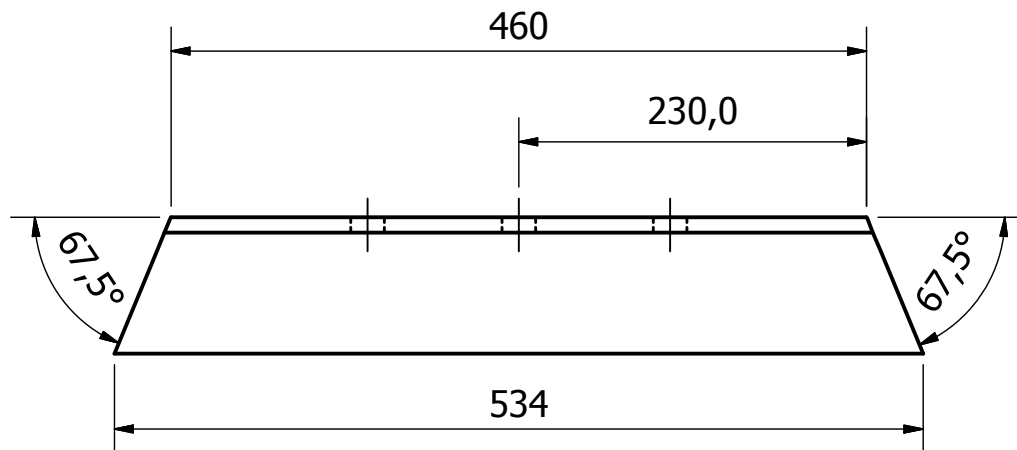
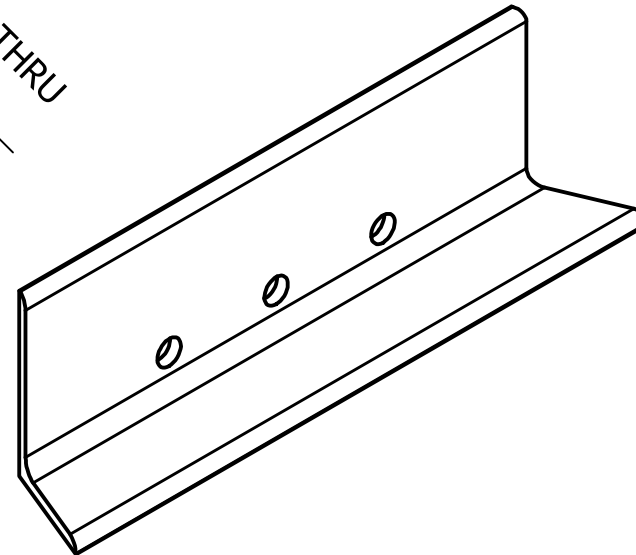
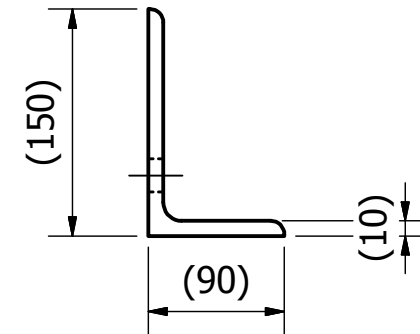
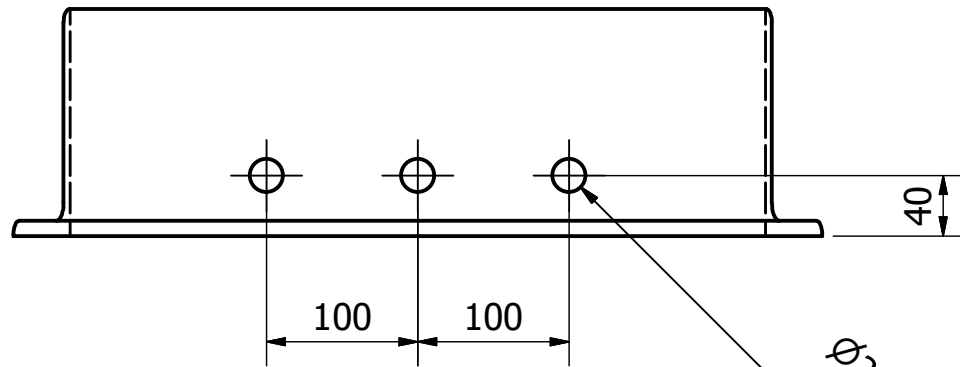
MATERIAL: 150 x 90 x 10 mm Unequal Angle SANS 50025 Grade S355JR Steel



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BCL013-1 : QTY 2 AS DRAW
BCL013-2 : QTY 2 TO OPP HAND



SCALE: 1:5

NOTES:
 SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
 DIMENSIONS IN mm.
 DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 13/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL013

REV NO. 3



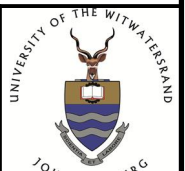
SHEET 2 OF 3

DESCRIPTION: 1200, 45° Support Angle Parts and Assembly Drawings

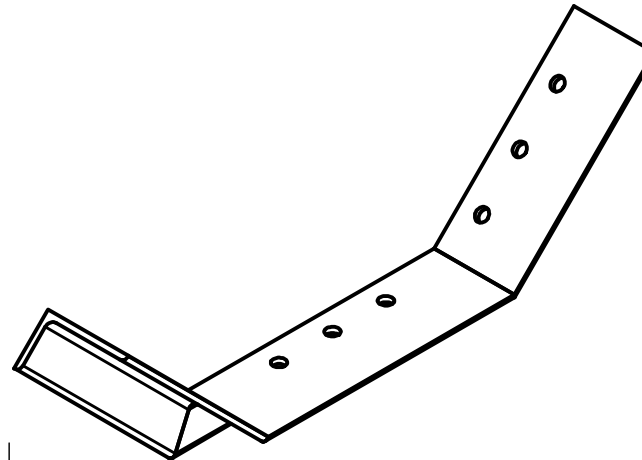
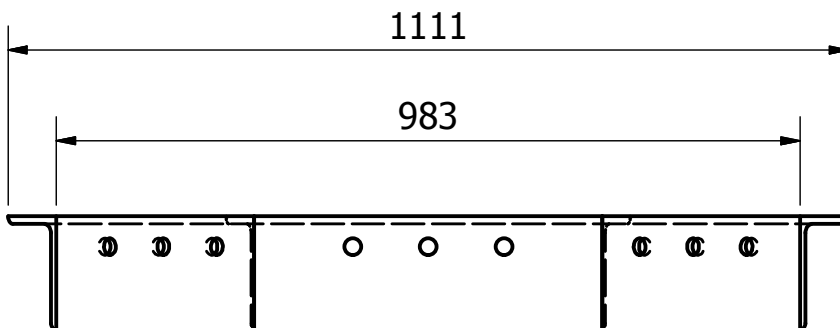
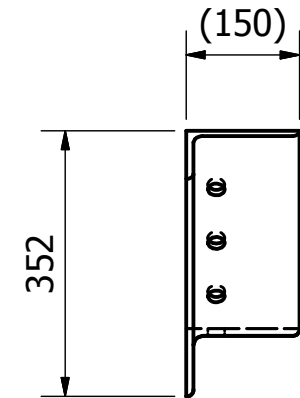
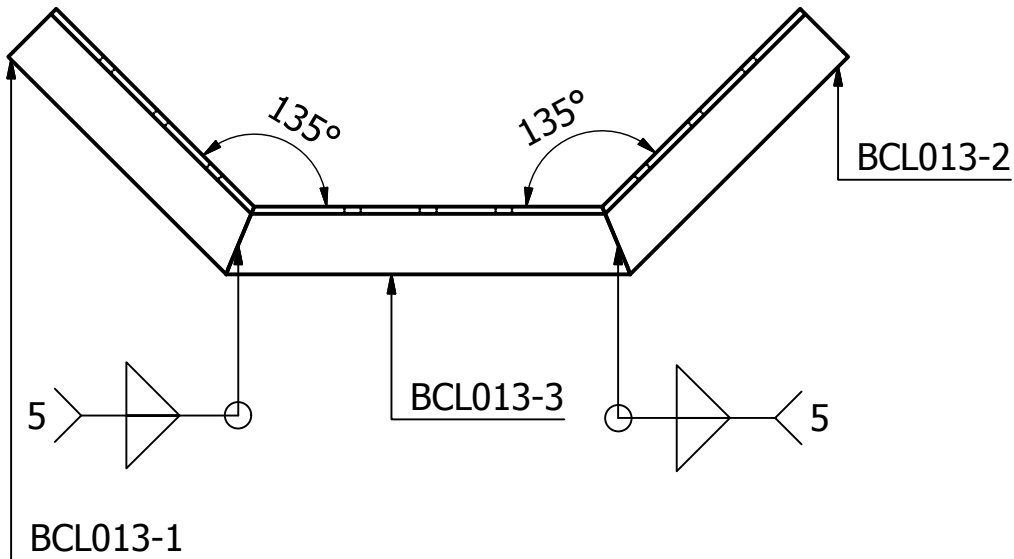
MATERIAL: 150 x 90 x 10 mm Unequal Angle SANS 50025 Grade S355JR Steel



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BCL013-3 : QTY 2 OFF



SCALE: 1:10

NOTES:
 SURFACE FINISH $\frac{3.2}{\sqrt{\quad}}$ UNLESS OTHERWISE STATED.
 DIMENSIONS IN mm.
 DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 13/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL013

REV NO. 3



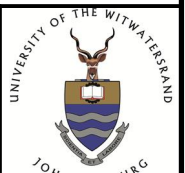
SHEET 3 OF 3

DESCRIPTION: 1200, 45° Support Angle Parts and Assembly Drawings

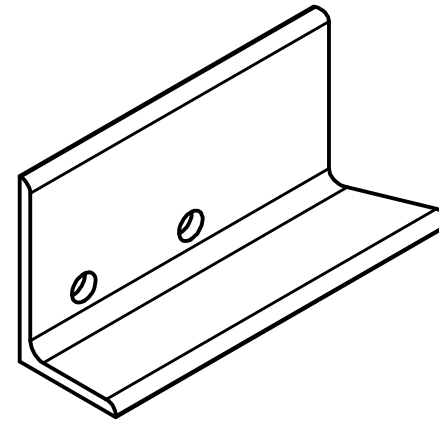
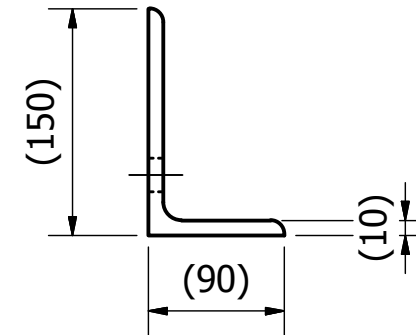
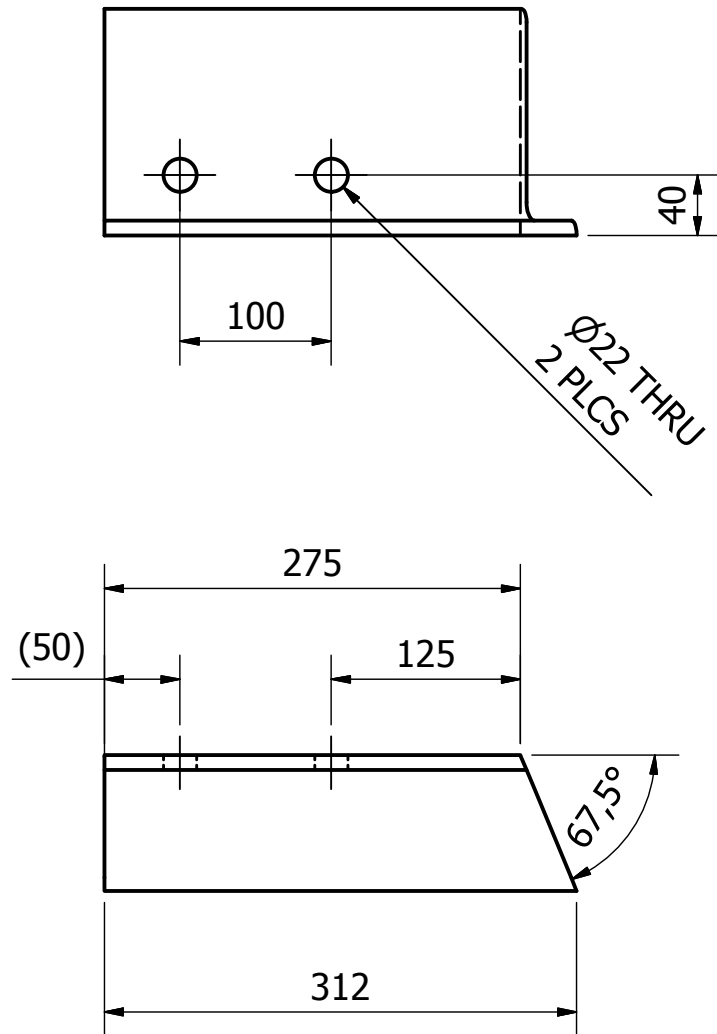
MATERIAL: 150 x 90 x 10 mm Unequal Angle SANS 50025 Grade S355JR Steel



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ASSEMBLY : QTY 2 OFF



SCALE: 1:5

NOTES:
 SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
 DIMENSIONS IN mm.
 DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 13/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL014

REV NO. 3



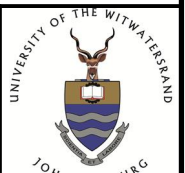
SHEET 1 OF 3

DESCRIPTION: 900, 45° Support Angle Parts and Assembly Drawings

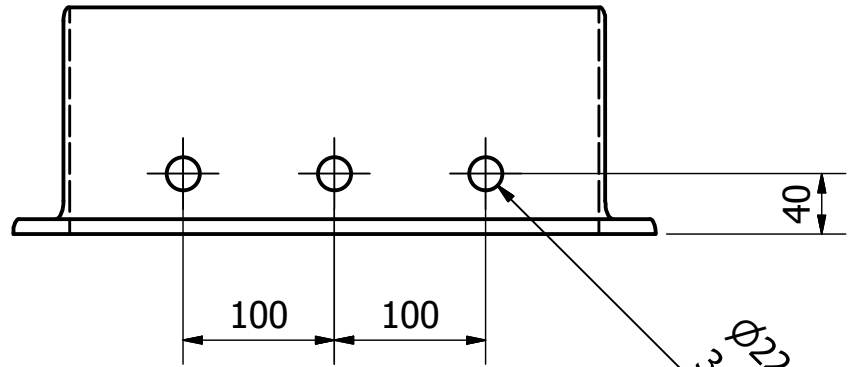
MATERIAL: 150 x 90 x 10 mm Unequal Angle SANS 50025 Grade S355JR Steel



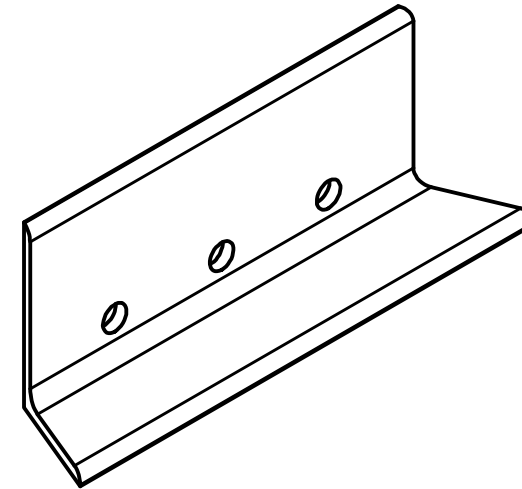
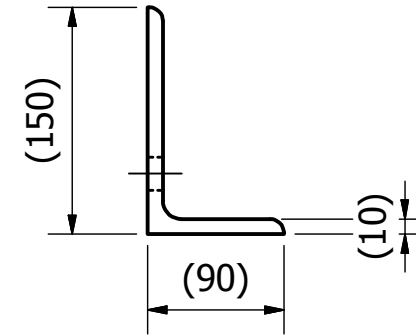
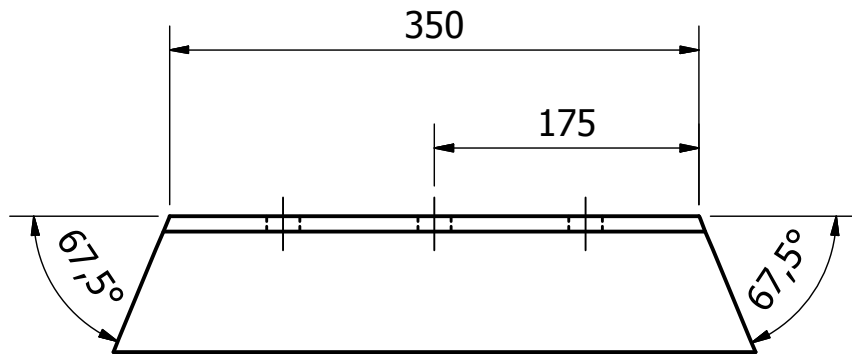
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 JOHANNESBURG.



BCL014-1 : QTY 2 AS DRAW
 BCL014-2 : QTY 2 TO OPP HAND



Ø22 THRU
3 PLCS



SCALE: 1:5

NOTES:
SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
DIMENSIONS IN mm.
DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 13/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL014

REV NO. 3



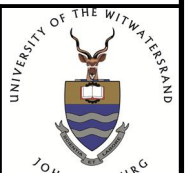
SHEET 2 OF 3

DESCRIPTION: 900, 45° Support Angle Parts and Assembly Drawings

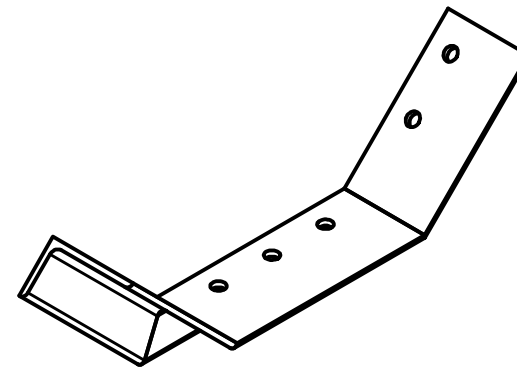
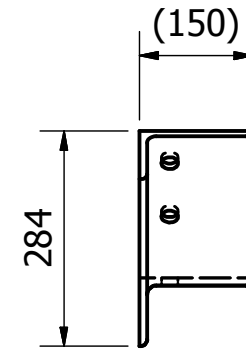
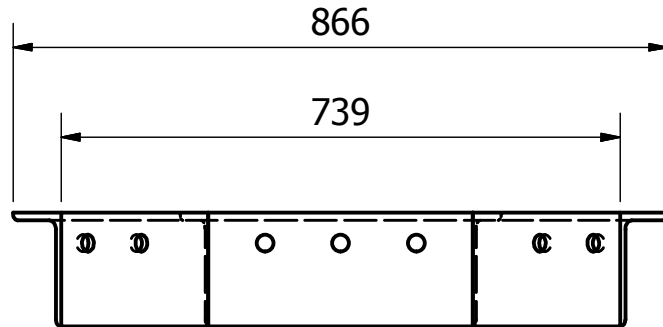
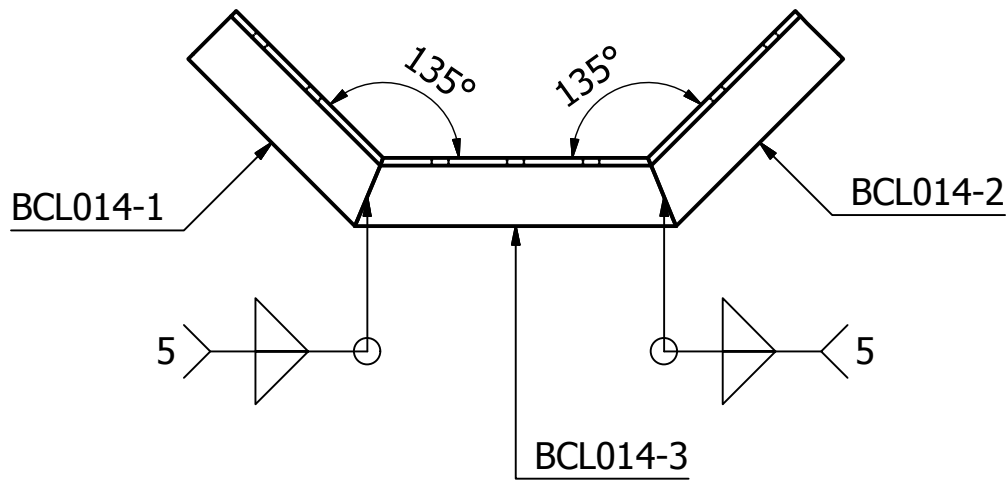
MATERIAL: 150 x 90 x 10 mm Unequal Angle SANS 50025 Grade S355JR Steel



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BCL014-3 : QTY 2 OFF



SCALE: 1:10

NOTES:
 SURFACE FINISH $\frac{3.2}{\sqrt{\text{mm}}}$ UNLESS OTHERWISE STATED.
 DIMENSIONS IN mm.
 DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 13/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL014

REV NO. 3



SHEET 3 OF 3

DESCRIPTION: 900, 45° Support Angle Parts and Assembly Drawings

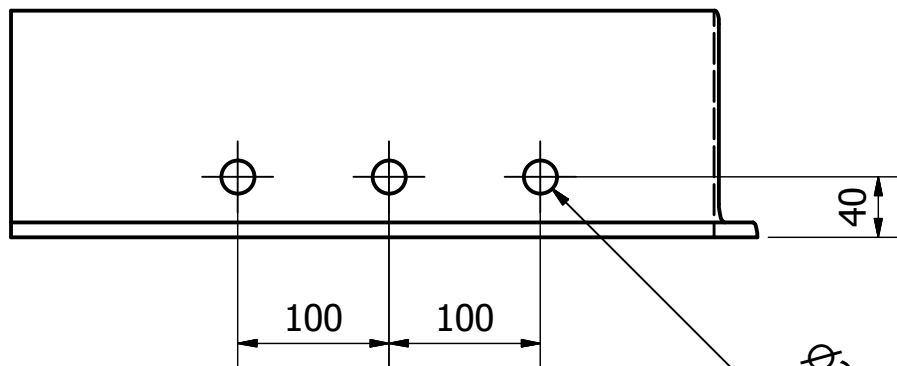
MATERIAL: 150 x 90 x 10 mm Unequal Angle SANS 50025 Grade S355JR Steel



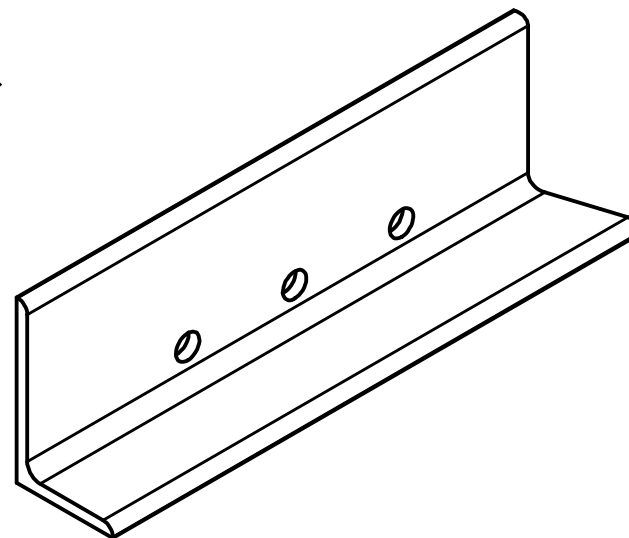
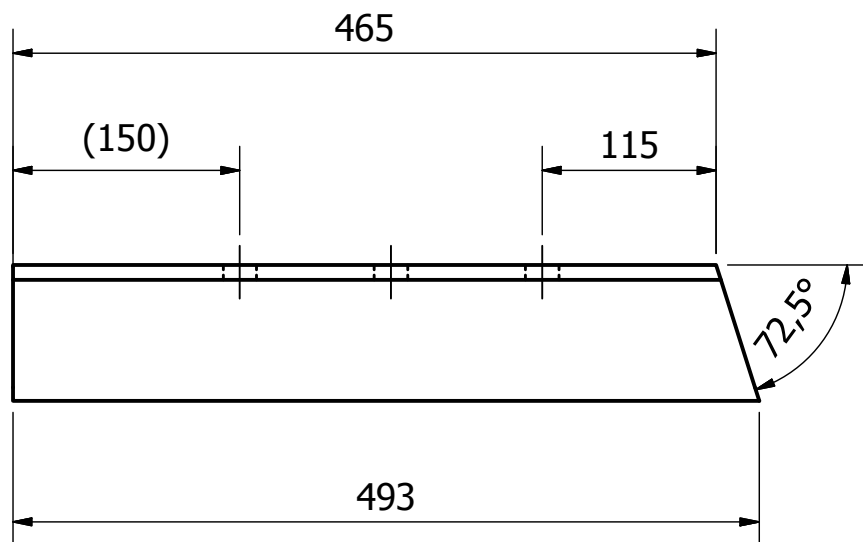
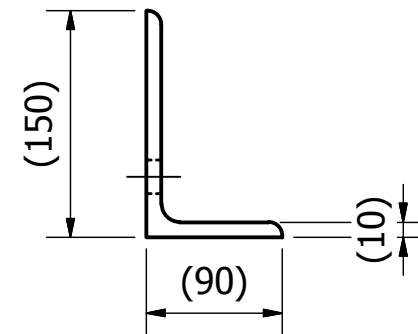
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ASSEMBLY : QTY 2 OFF



Ø22 THRU
3 PLCS



SCALE: 1:5

NOTES:
SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
DIMENSIONS IN mm.
DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 13/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL015

REV NO. 3



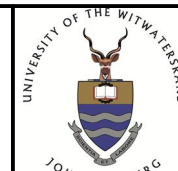
SHEET 1 OF 3

DESCRIPTION: 1500, 35° Support Angle Parts and Assembly Drawings

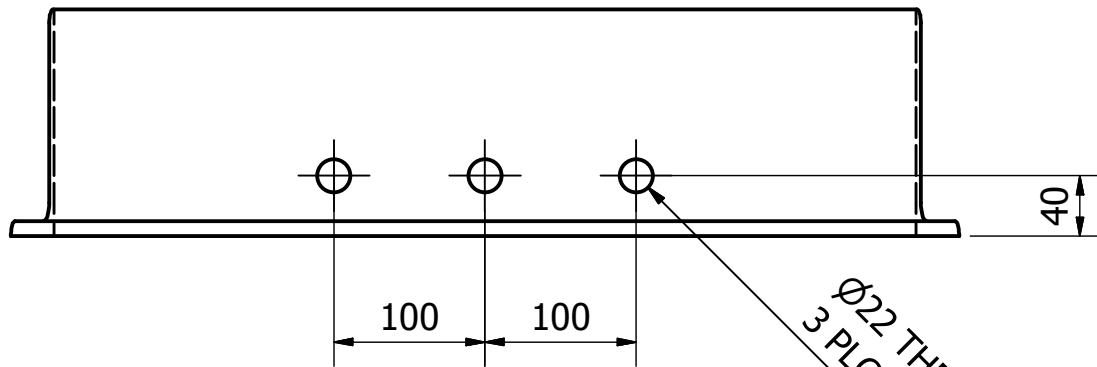
MATERIAL: 150 x 90 x 10 mm Unequal Angle SANS 50025 Grade S355JR Steel



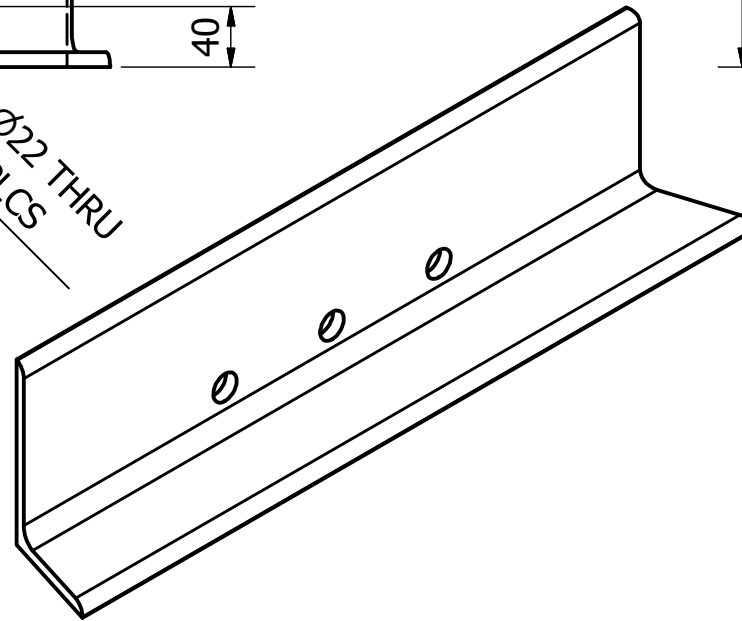
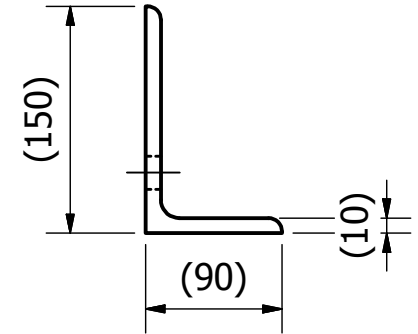
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BCL015-1 : QTY 2 AS DRAW
BCL015-2 : QTY 2 TO OPP HAND



Ø22 THRU
3 PLCS



SCALE: 1:5

NOTES:
SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
DIMENSIONS IN mm.
DO NOT SCALE.

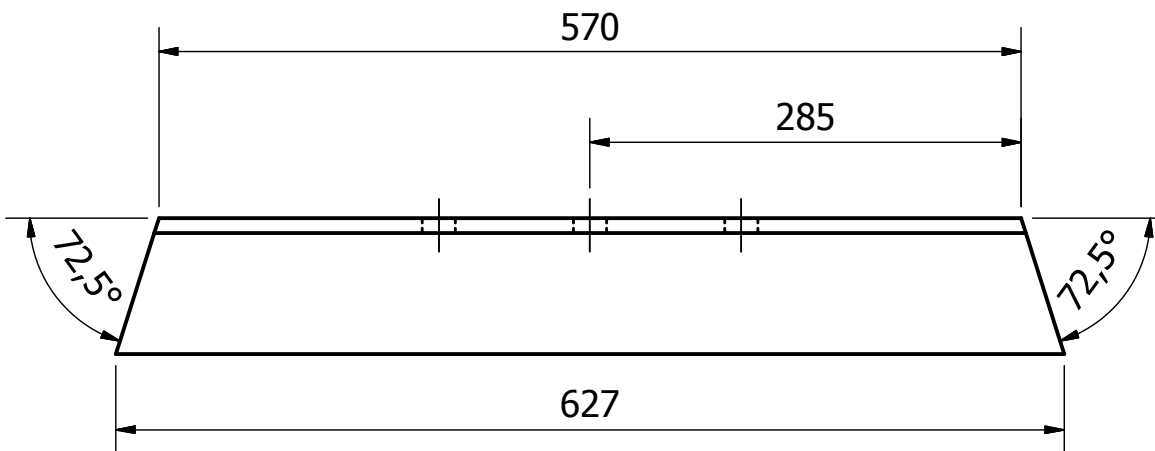
APPROVED:

CHECKED:

DATE: 13/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4



BCL015-3 : QTY 2 OFF

DRG NO: BCL015

REV NO. 3



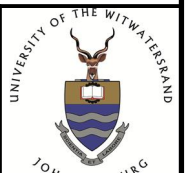
SHEET 2 OF 3

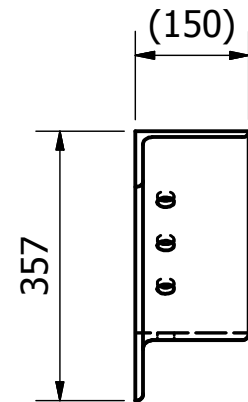
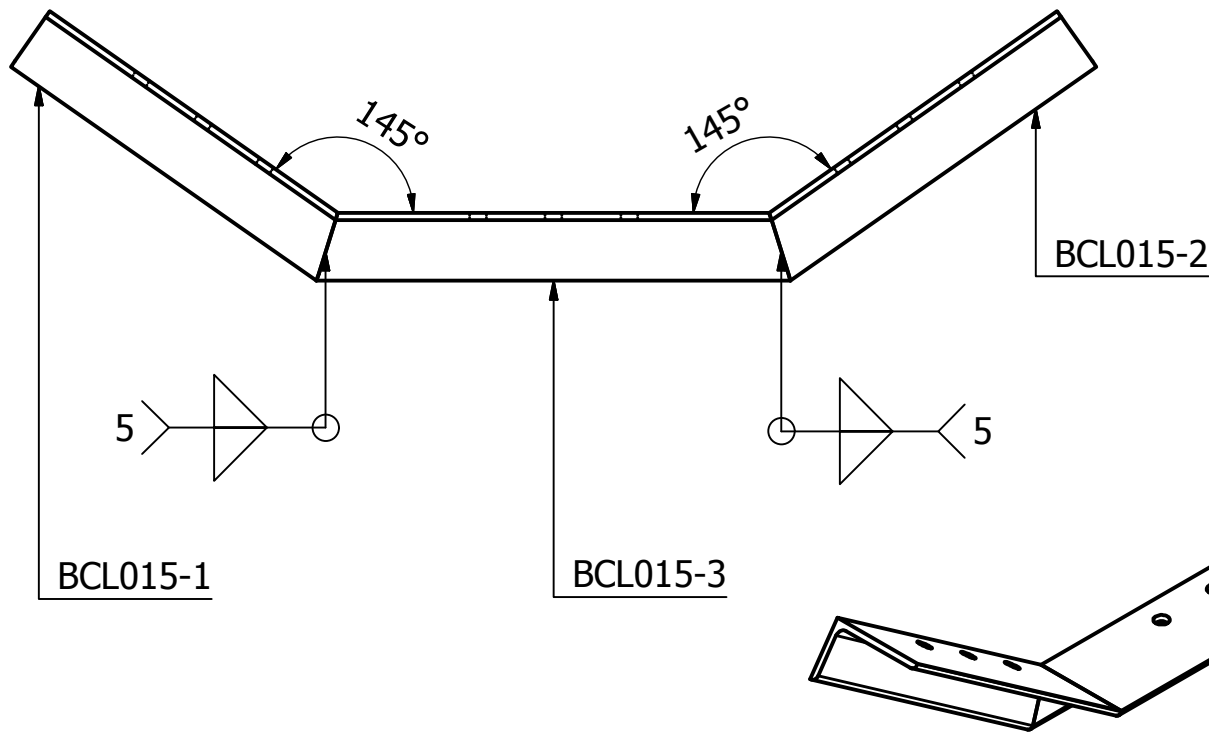
DESCRIPTION: 1500, 35° Support Angle Parts and Assembly Drawings

MATERIAL: 150 x 90 x 10 mm Unequal Angle SANS 50025 Grade S355JR Steel



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SCALE: 1:10

NOTES:
 SURFACE FINISH $\frac{3.2}{\sqrt{\quad}}$ UNLESS OTHERWISE STATED.
 DIMENSIONS IN mm.
 DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 13/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL015

REV NO. 3



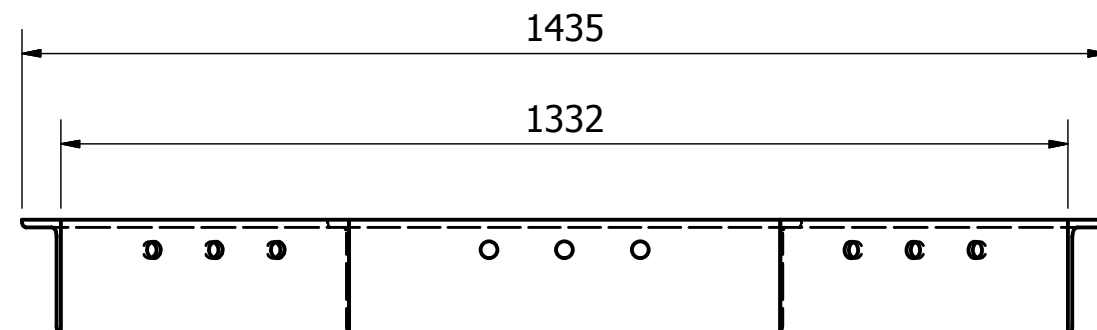
SHEET 3 OF 3

DESCRIPTION: 1500, 35° Support Angle Parts and Assembly Drawings

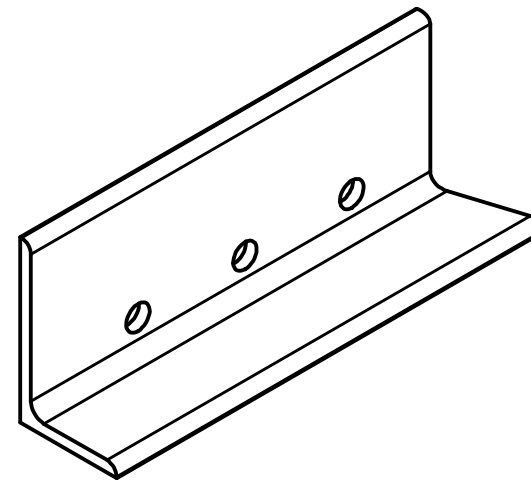
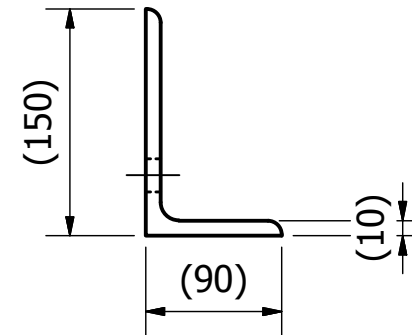
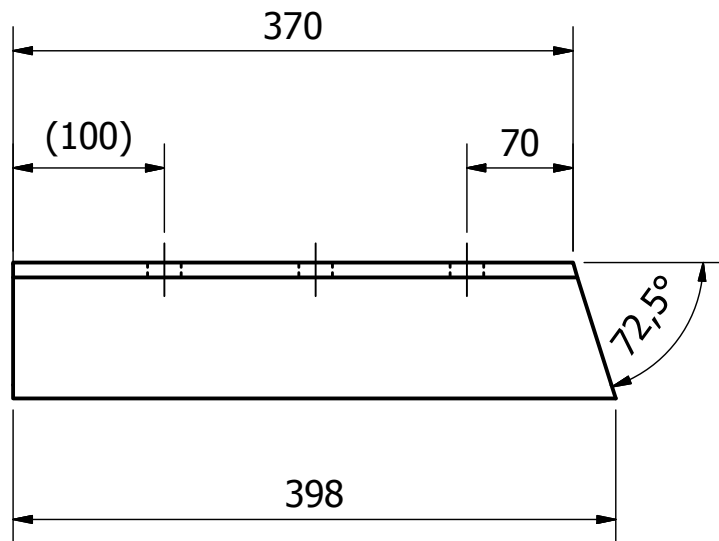
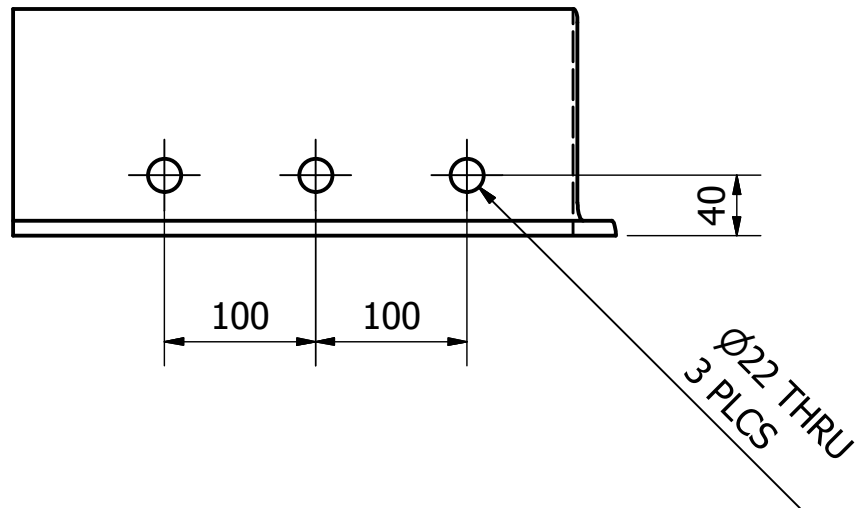
MATERIAL: 150 x 90 x 10 mm Unequal Angle SANS 50025 Grade S355JR Steel



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ASSEMBLY : QTY 2 OFF



SCALE: 1:5

NOTES: SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED. DIMENSIONS IN mm. DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 13/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL016

REV NO. 3



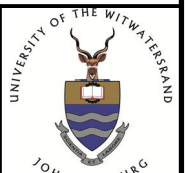
SHEET 1 OF 3

DESCRIPTION: 1200, 35° Support Angle Parts and Assembly Drawings

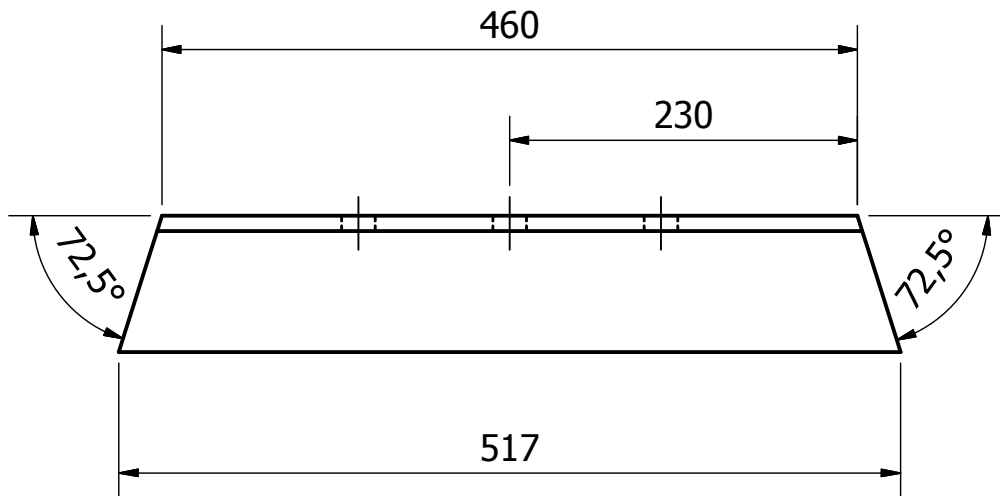
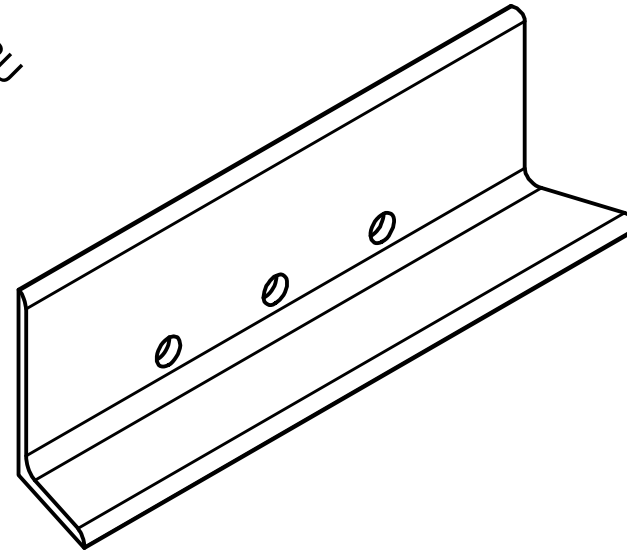
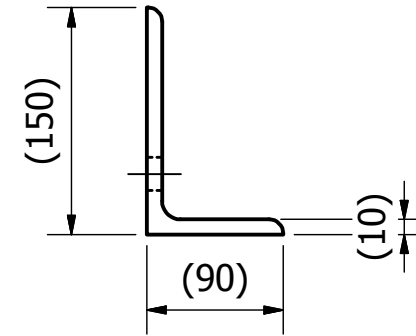
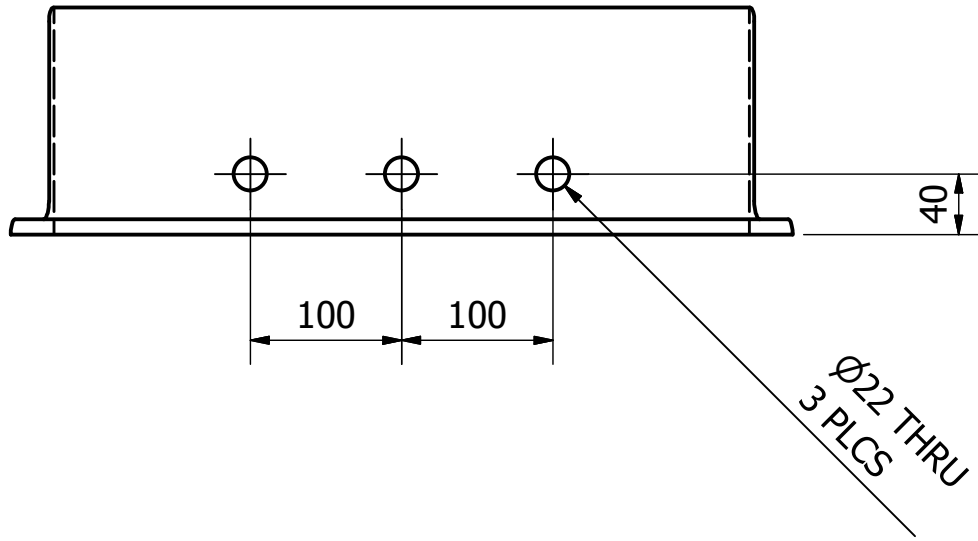
MATERIAL: 150 x 90 x 10 mm Unequal Angle SANS 50025 Grade S355JR Steel



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BCL016-1 : QTY 2 AS DRAW
BCL016-2 : QTY 2 TO OPP HAND



BCL016-3 : QTY 2 OFF

SCALE: 1:5

NOTES:
SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
DIMENSIONS IN mm.
DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 13/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL016

REV NO. 3



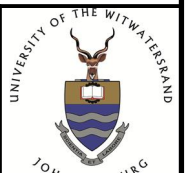
SHEET 2 OF 3

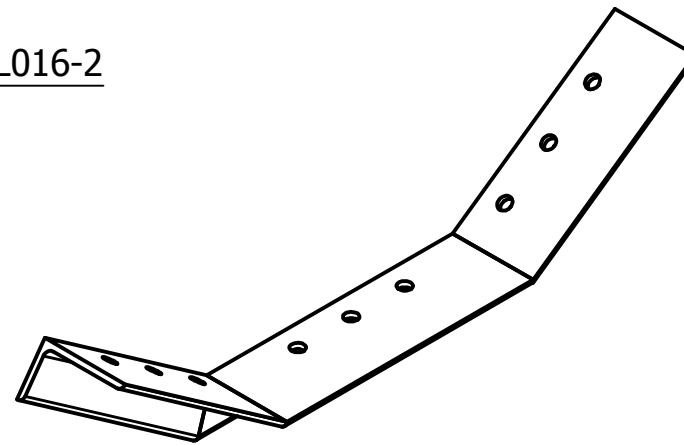
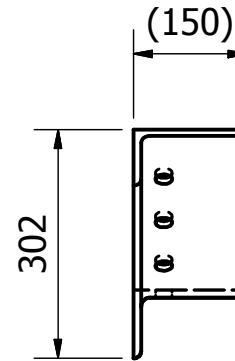
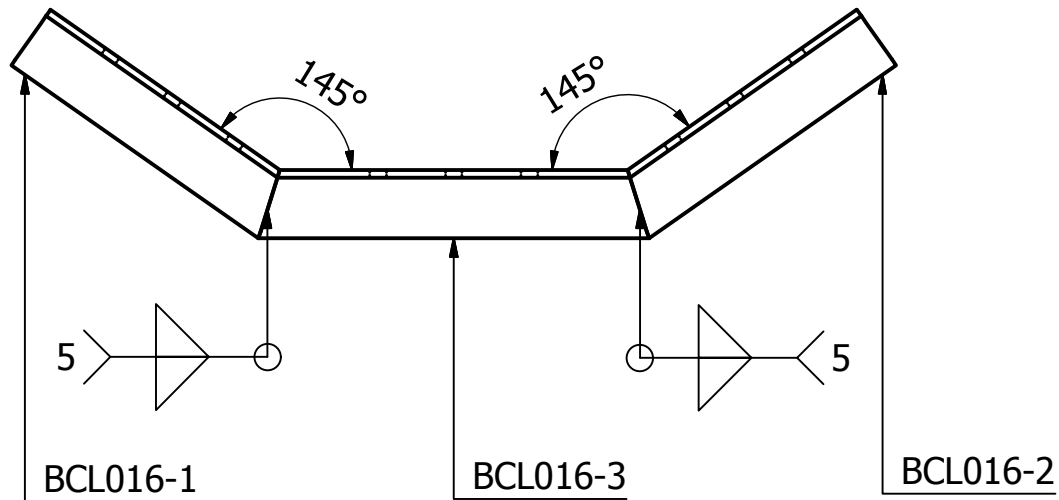
DESCRIPTION: 1200, 35° Support Angle Parts and Assembly Drawings

MATERIAL: 150 x 90 x 10 mm Unequal Angle SANS 50025 Grade S355JR Steel



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SCALE: 1:10

NOTES:
 SURFACE FINISH $\checkmark$ 3.2 UNLESS OTHERWISE STATED.
 DIMENSIONS IN mm.
 DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 13/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL016

REV NO. 3



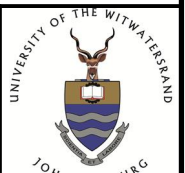
SHEET 3 OF 3

DESCRIPTION: 1200, 35° Support Angle Parts and Assembly Drawings

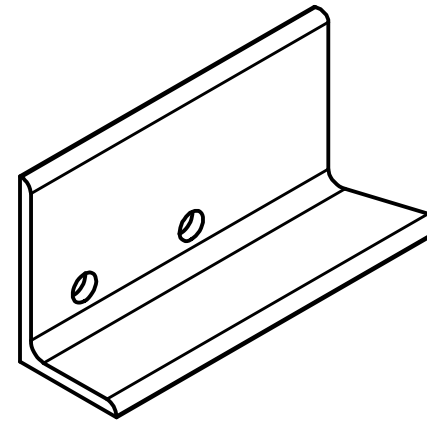
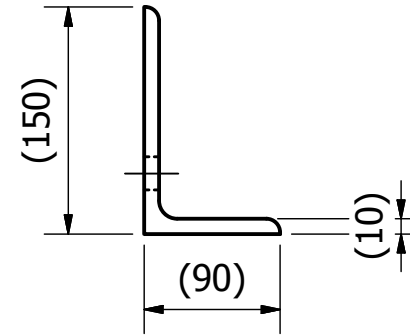
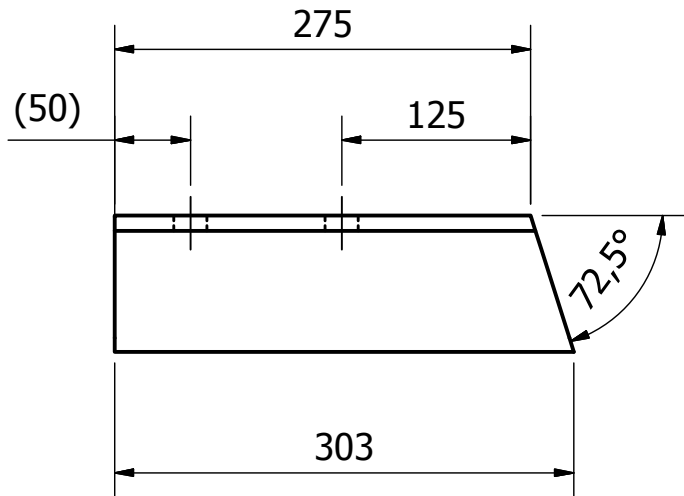
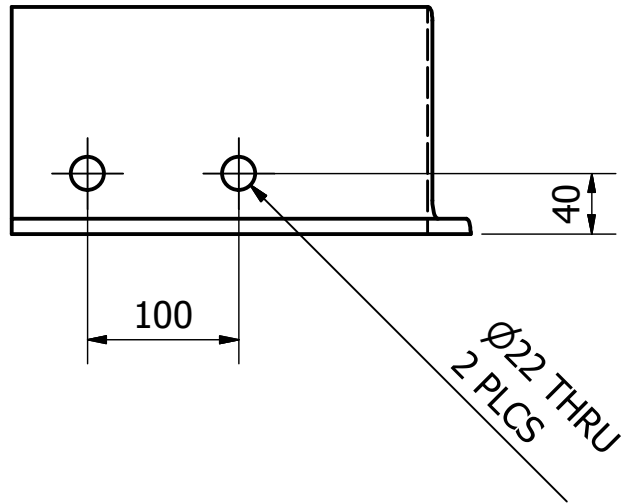
MATERIAL: 150 x 90 x 10 mm Unequal Angle SANS 50025 Grade S355JR Steel



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ASSEMBLY : QTY 2 OFF



SCALE: 1:5

NOTES:
 SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
 DIMENSIONS IN mm.
 DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 13/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL017

REV NO. 3



SHEET 1 OF 3

DESCRIPTION: 900, 35° Support Angle Parts and Assembly Drawings

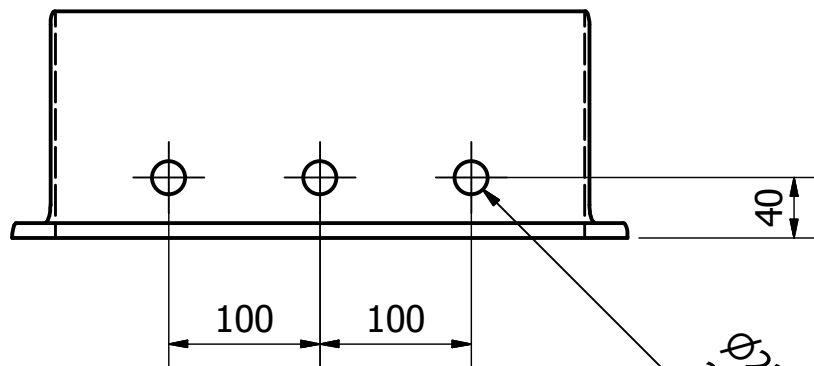
MATERIAL: 150 x 90 x 10 mm Unequal Angle SANS 50025 Grade S355JR Steel



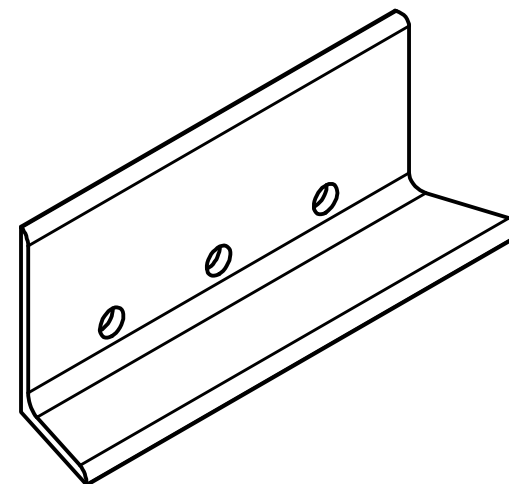
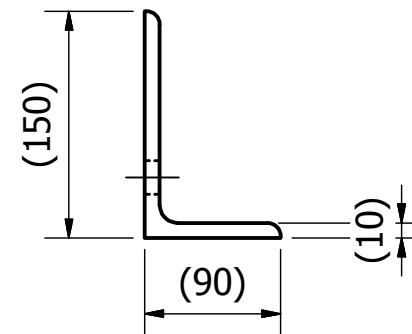
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BCL017-1 : QTY 2 AS DRAW
 BCL017-2 : QTY 2 TO OPP HAND



Ø22 THRU
3 PLCS



SCALE: 1:5

NOTES:
SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
DIMENSIONS IN mm.
DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 13/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL017

REV NO. 3



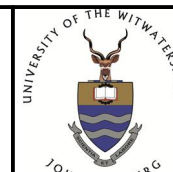
SHEET 2 OF 3

DESCRIPTION: 900, 35° Support Angle Parts and Assembly Drawings

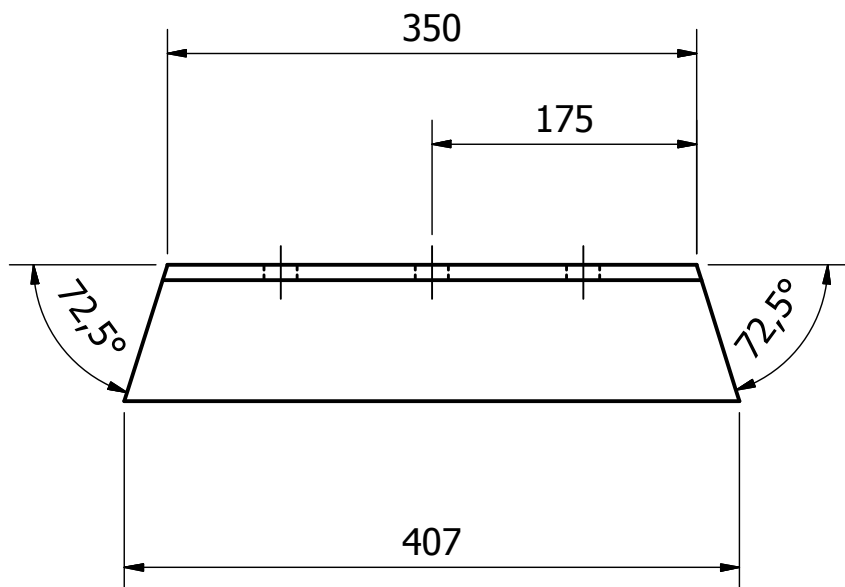
MATERIAL: 150 x 90 x 10 mm Unequal Angle SANS 50025 Grade S355JR Steel

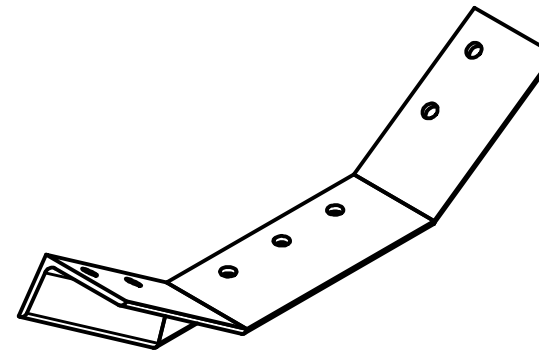
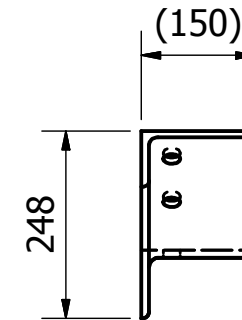
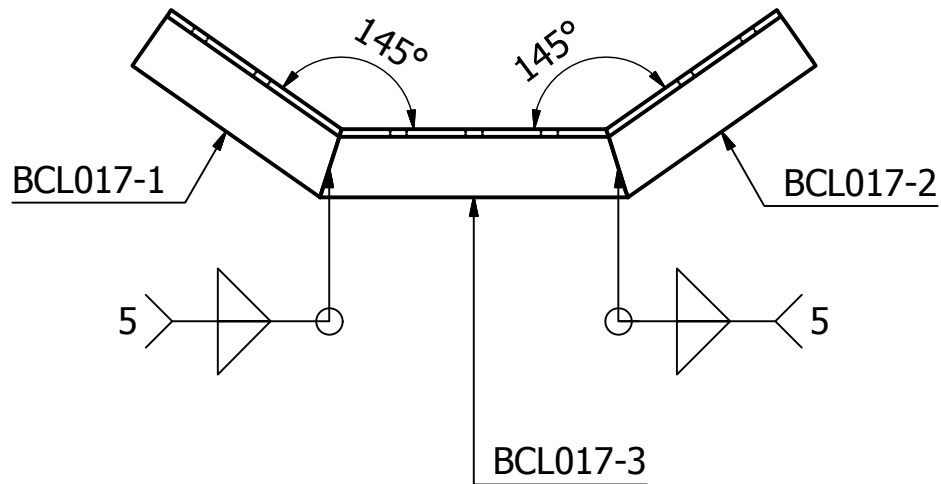


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BCL017-3 : QTY 2 OFF





SCALE: 1:10

NOTES: SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
DIMENSIONS IN mm.
DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 13/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL017

REV NO. 3



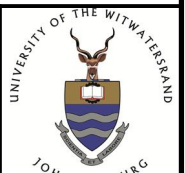
SHEET 3 OF 3

DESCRIPTION: 900, 35° Support Angle Parts and Assembly Drawings

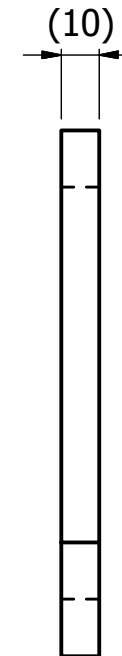
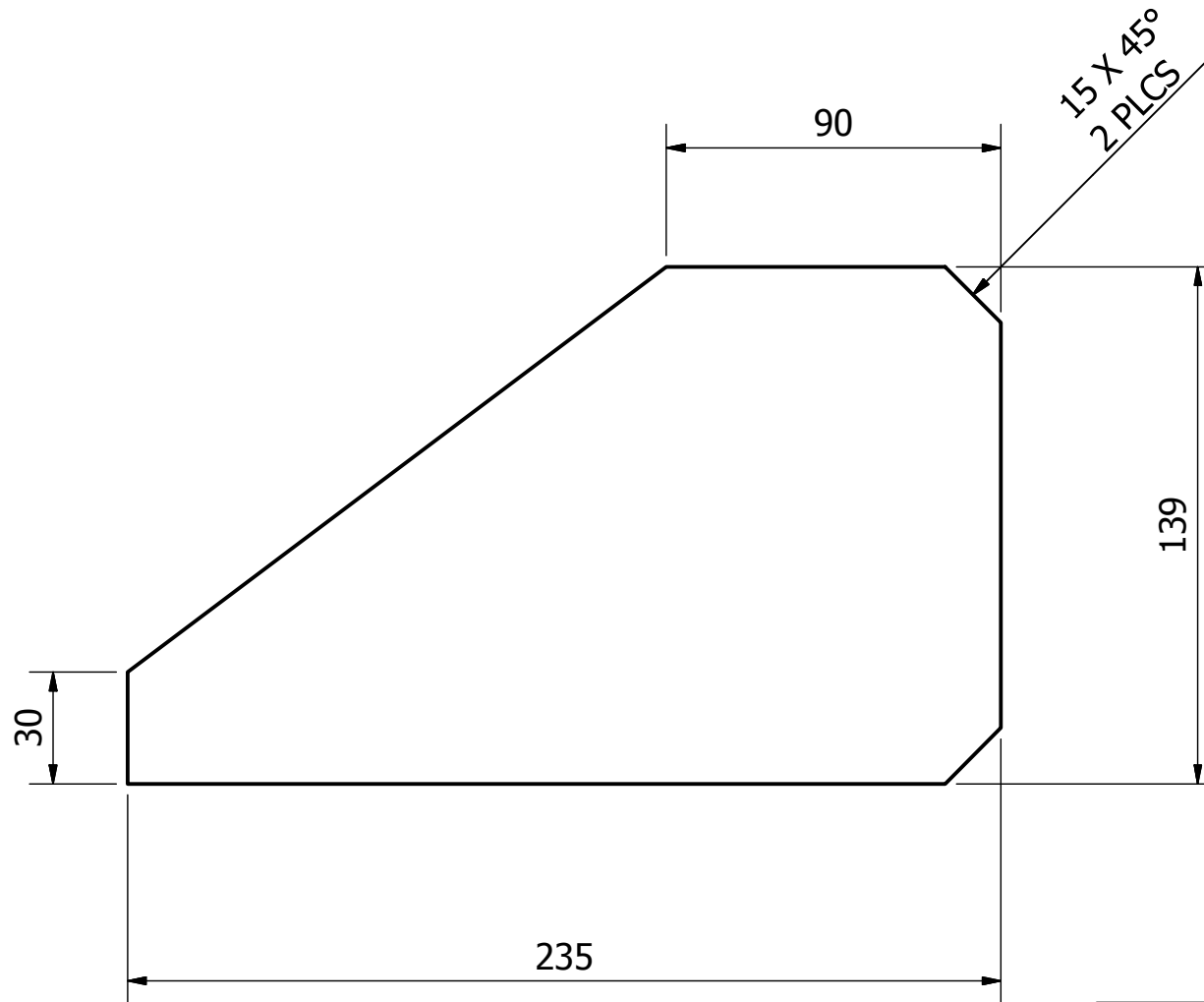
MATERIAL: 150 x 90 x 10 mm Unequal Angle SANS 50025 Grade S355JR Steel



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ASSEMBLY : QTY 2 OFF



SCALE: 1:2

NOTES:
 SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
 DIMENSIONS IN mm.
 DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 13/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL018

REV NO. 3



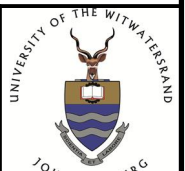
SHEET 1 OF 1

DESCRIPTION: Large Stiffener Manufacturing Drawing

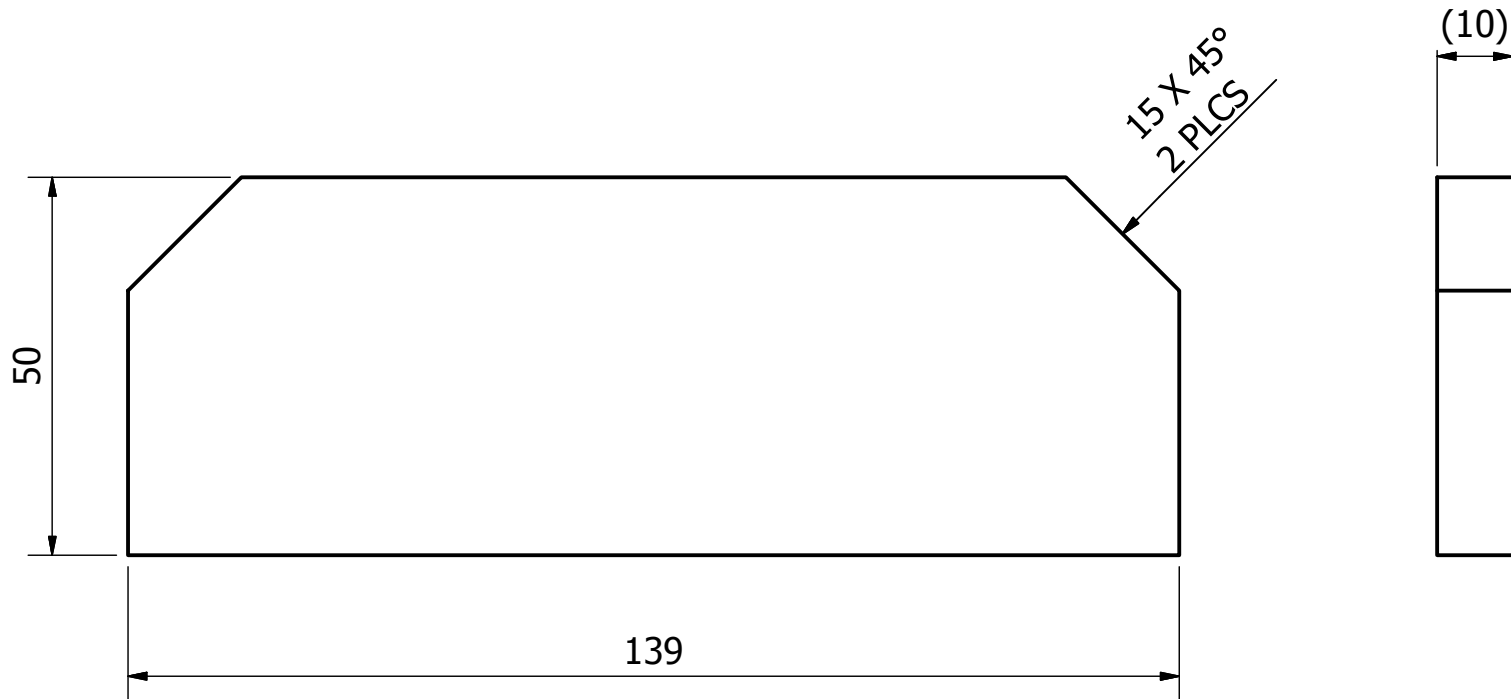
MATERIAL: 139 x 10 mm PLT x 235 LG



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QTY 28 OFF



SCALE: 1:1

NOTES: ^{3.2}✓
SURFACE FINISH UNLESS
OTHERWISE STATED.
DIMENSIONS IN mm.
DO NOT SCALE.

APPROVED:

CHECKED:

DATE: 13/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A4

DRG NO: BCL019

REV NO. 3



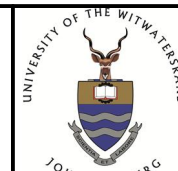
SHEET 1 OF 1

DESCRIPTION: Small Stiffener Manufacturing Drawing

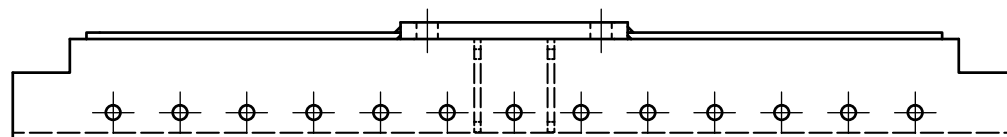
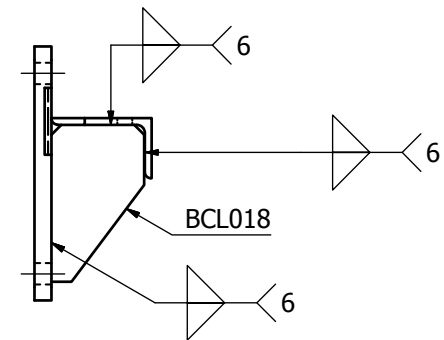
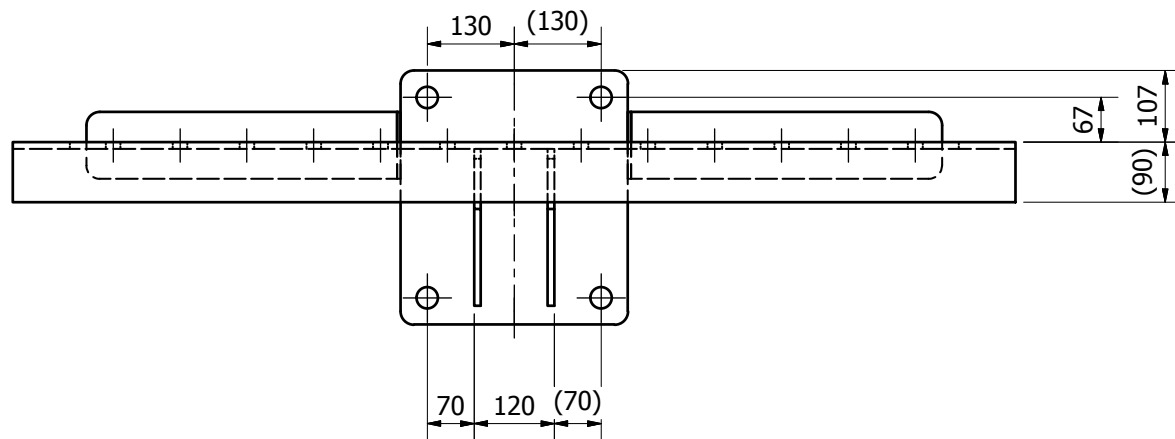
MATERIAL: 50 x 10 mm PLT x 139 LG



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JOHANNESBURG.

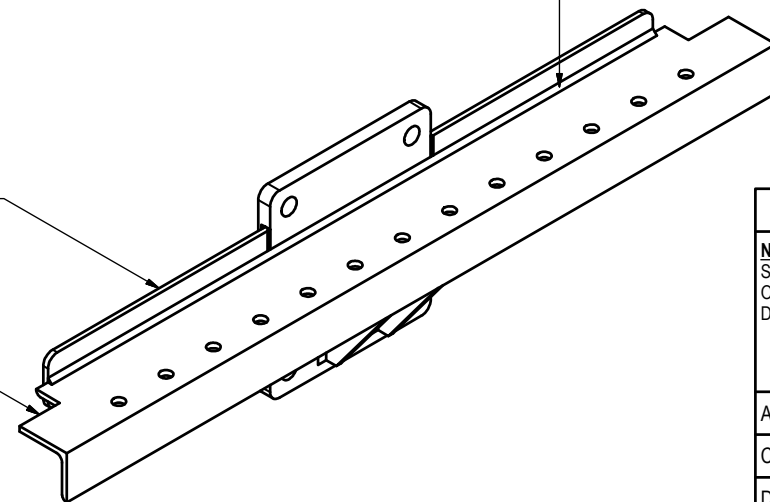


QTY 28 OFF



BCL005

BCL003



SCALE: 1:8

NOTES:
SURFACE FINISH $\sqrt{3.2}$ UNLESS
OTHERWISE STATED.
DIMENSIONS IN mm.

APPROVED:

CHECKED:

DATE: 13/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A3

DRG NO: BCL020

REV NO. 2



SHEET 1 OF 1

DESCRIPTION: 0° Belt Clamp Base Assembly Drawing

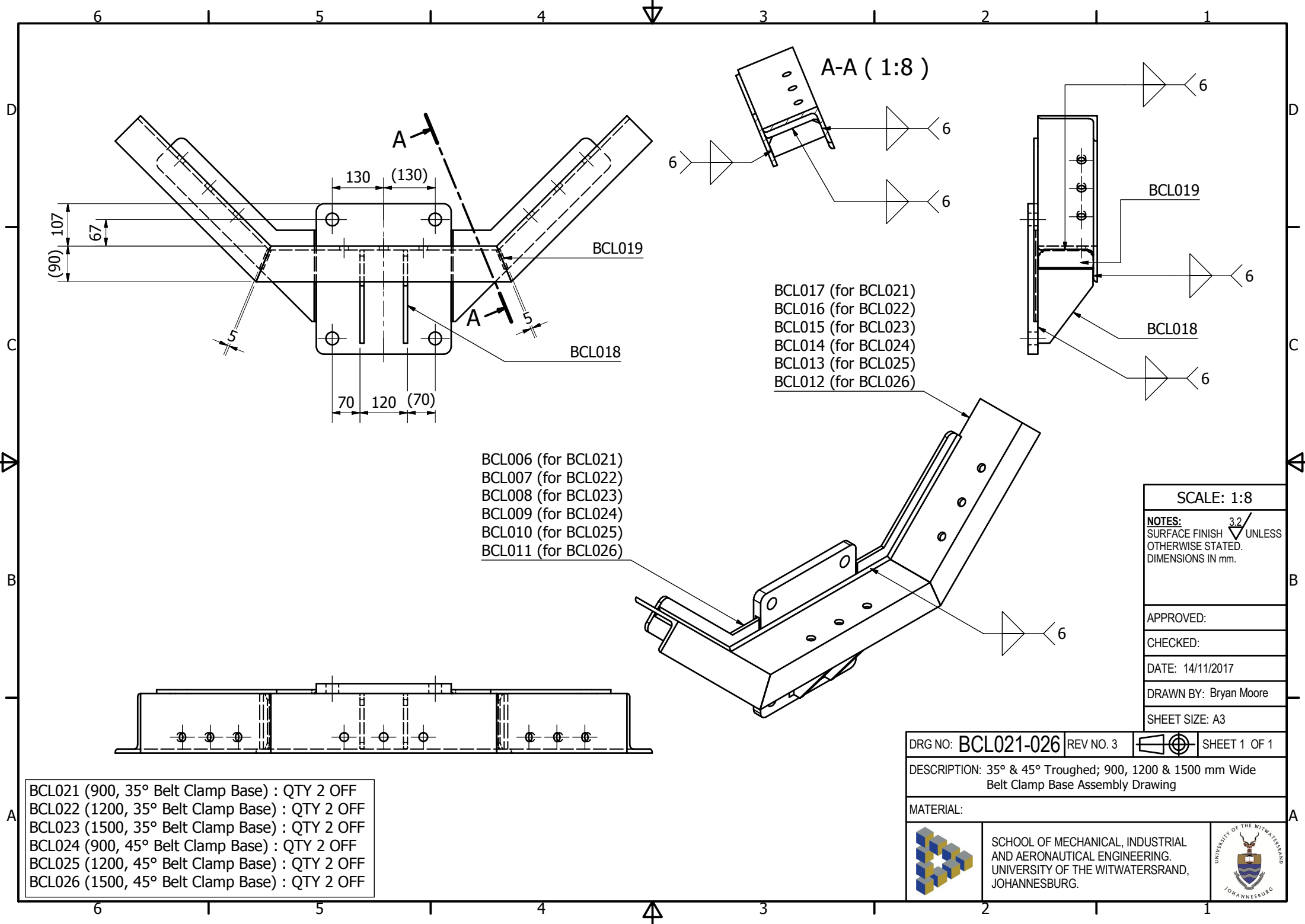
MATERIAL:



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JOHANNESBURG.



QTY 2 OFF



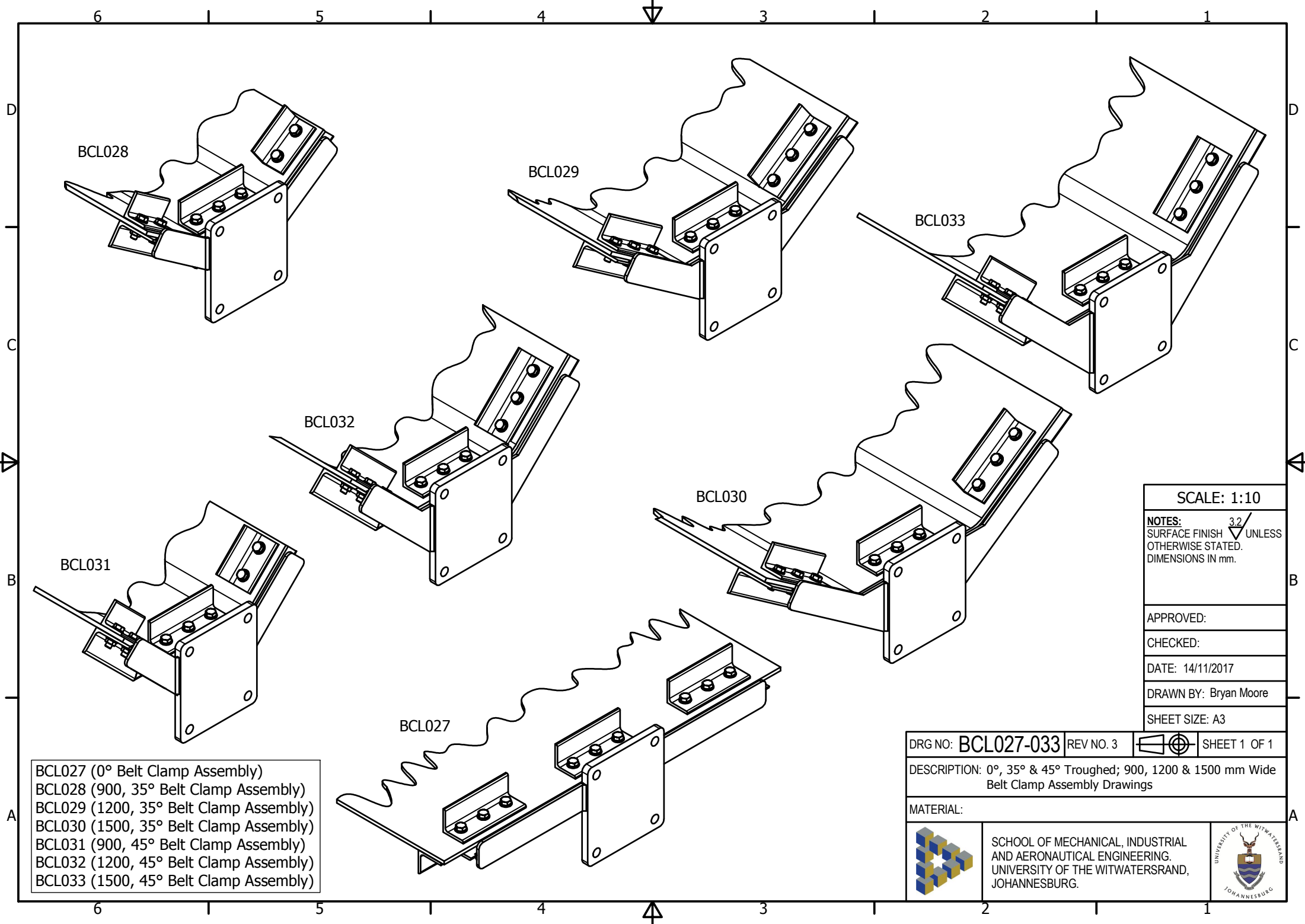
- BCL021 (900, 35° Belt Clamp Base) : QTY 2 OFF
BCL022 (1200, 35° Belt Clamp Base) : QTY 2 OFF
BCL023 (1500, 35° Belt Clamp Base) : QTY 2 OFF
BCL024 (900, 45° Belt Clamp Base) : QTY 2 OFF
BCL025 (1200, 45° Belt Clamp Base) : QTY 2 OFF
BCL026 (1500, 45° Belt Clamp Base) : QTY 2 OFF

- BCL006 (for BCL021)
BCL007 (for BCL022)
BCL008 (for BCL023)
BCL009 (for BCL024)
BCL010 (for BCL025)
BCL011 (for BCL026)

- BCL017 (for BCL021)
BCL016 (for BCL022)
BCL015 (for BCL023)
BCL014 (for BCL024)
BCL013 (for BCL025)
BCL012 (for BCL026)

SCALE: 1:8
NOTES: SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED. DIMENSIONS IN mm.
APPROVED:
CHECKED:
DATE: 14/11/2017
DRAWN BY: Bryan Moore
SHEET SIZE: A3

DRG NO: BCL021-026	REV NO. 3	SHEET 1 OF 1
DESCRIPTION: 35° & 45° Troughed; 900, 1200 & 1500 mm Wide Belt Clamp Base Assembly Drawing		
MATERIAL:		
	SCHOOL OF MECHANICAL, INDUSTRIAL AND AERONAUTICAL ENGINEERING. UNIVERSITY OF THE WITWATERSRAND, JOHANNESBURG.	



- BCL027 (0° Belt Clamp Assembly)
BCL028 (900, 35° Belt Clamp Assembly)
BCL029 (1200, 35° Belt Clamp Assembly)
BCL030 (1500, 35° Belt Clamp Assembly)
BCL031 (900, 45° Belt Clamp Assembly)
BCL032 (1200, 45° Belt Clamp Assembly)
BCL033 (1500, 45° Belt Clamp Assembly)

SCALE: 1:10

NOTES:
SURFACE FINISH $\sqrt{3.2}$ UNLESS OTHERWISE STATED.
DIMENSIONS IN mm.

APPROVED:

CHECKED:

DATE: 14/11/2017

DRAWN BY: Bryan Moore

SHEET SIZE: A3

DRG NO: BCL027-033 REV NO. 3 SHEET 1 OF 1

DESCRIPTION: 0°, 35° & 45° Troughed; 900, 1200 & 1500 mm Wide Belt Clamp Assembly Drawings

MATERIAL:



SCHOOL OF MECHANICAL, INDUSTRIAL AND AERONAUTICAL ENGINEERING.
UNIVERSITY OF THE WITWATERSRAND,
JOHANNESBURG.



Appendix D Load Cell Calibration Certificates

The following Appendix presents the calibration certificates of the load cells used to measure the resistance to motion of a locked idler.

P109315

P.O.Box 2 Hanzhong, 723007
Shaanxi, China
Tel: 86-916-2386271, 2386352
FAX: 86-916-2576271
E-mail: admin@zemic.com.cn



CALIBRATION CERTIFICATE

NOTE:

WE CERTIFY THAT THE PRODUCT DESCRIBED BELOW CONFORMS TO ALL APPLICABLE SPECIFICATIONS. CALIBRATION IS TRACEABLE TO THE NATIONAL INSTITUTE OF STANDARDS AND TECHNOLOGY. CALIBRATION IS PERFORMED AT HANZHONG, SHAANXI, WHERE THE LOCAL ACCELERATION OF GRAVITY IS 9.7934 m/s^2 .

General Data

Model: H8C-C3-2.0t-4B-D04
Serial Number: P109315
Material: ALLOY STEEL

Capacity: 2.0 t
Excitation: 5~12VDC(NOM), 18VDC(MAX)
Insul Res: >5000M Ω (50VDC)

Calibration Data

Full Scale Output: 3.0010 mV/V
Non-Repeatability: $\pm 0.01\%$ of FS
Input Resistance: $350 \pm 3.5 \Omega$
Temp. Sens. Zero: $< 0.017\% \text{FS}/10^\circ\text{C}$
Temp. Sens. Output: $\pm 0.011\% \text{FS}/10^\circ\text{C}$
Non-linearity: 0.02% of FS
Cable Length: 4 m
Safe Overload(150%): 3.0 t
Ultimate Overload(300%): 6.0 t

Combined Error: 0.02% of FS
Zero Balance: $< 1.0\%$ of FS
Output Resistance: $351 \pm 2 \Omega$
Comp. Temp. Range: -10°C to 40°C
Operating Temp: -35°C to 70°C
Hysteresis: 0.02% of FS

Class: C3

NML TC 8012

ELECTRICAL CONNECTIONS-COMPRESSION POSITIVE

RED --- +EXC
BLACK --- -EXC
GREEN --- +SIGNEl
WHITE --- -SIGNEl
BARE --- SHIELD

CAUTION: Cutting cable will affect the Full Scale Output calibration.

QA Manager-Zhigang Zhang

Specifications are subject to change without notice.

DATE: 12-08-2013



All units are produced in a Quality System that meets the requirements of ISO-9001, and the certificate number is 00811Q20540R5M.

P111167

P.O.Box 2 Hanzhong, 723007
Shaanxi, China
Tel: 86-916-2386271, 2386352
FAX: 86-916-2576271
E-mail: admin@zemic.com.cn



CALIBRATION CERTIFICATE

NOTE:

WE CERTIFY THAT THE PRODUCT DESCRIBED BELOW CONFORMS TO ALL APPLICABLE SPECIFICATIONS. CALIBRATION IS TRACEABLE TO THE NATIONAL INSTITUTE OF STANDARDS AND TECHNOLOGY. CALIBRATION IS PERFORMED AT HANZHONG, SHAANXI, WHERE THE LOCAL ACCELERATION OF GRAVITY IS 9.7934m/s^2 .

General Data

Model: H8C-C3-2.0t-4B-D04
Serial Number: P111167
Material: ALLOY STEEL

Capacity: 2.0 t
Excitation: 5~12 VDC(NOM), 18 VDC(MAX)
Insul Res: >5000M Ω (50 VDC)

Calibration Data

Full Scale Output: 3.0021 mV/V
Non-Repeatability: ± 0.01 % of FS
Input Resistance: 350 ± 3.5 Ohm
Temp.Sens.Zero: < 0.014 %FS/ 10°C
Temp.Sens.Output: < 0.011 %FS/ 10°C
Non-linearity: 0.02 % of FS
Cable Length: 4 m
Safe Overload(150%): 3.0 t
Ultimate Overload(300%): 6.0 t

Combined Error: 0.02 % of FS
Zero Balance: < 1.0 % of FS
Output Resistance: 351 ± 2 Ohm
Comp.Temp.Range: -10°C to 40°C
Operating Temp.: -35°C to 70°C
Hysteresis: 0.02 % of FS

Class: C3
NMI TC 8012

ELECTRICAL CONNECTIONS-COMPRESSION POSITIVE

RED — +EXC
BLACK — -EXC
GREEN — +SIGNAL
WHITE — -SIGNAL
BARE — SHIELD

CAUTION: Cutting cable will affect the Full Scale Output calibration.

QA Manager—Zhigang Zhang

Specifications are subject to change without notice.

DATE: 7-26-2013

S



All units are produced in a Quality System that meets the requirements of ISO-9001, and the certificate number is 00811Q20540R5M.

P111191

P.O.Box 2 Hanzhong,723007
Shaanxi,China
Tel:86-916-2386271,2386352
FAX:86-916-2576271
E-mail:admin@zemic.com.cn



ZEMIC

CALIBRATION CERTIFICATE

NOTE:

WE CERTIFY THAT THE PRODUCT DESCRIBED BELOW CONFORMS TO ALL APPLICABLE SPECIFICATIONS. CALIBRATION IS TRACEABLE TO THE NATIONAL INSTITUTE OF STANDARDS AND TECHNOLOGY. CALIBRATION IS PERFORMED AT HANZHONG, SHAANXI, WHERE THE LOCAL ACCELERATION OF GRAVITY IS 9.7934m/s^2 .

General Data

Model: H8C-C3-2.0t-4B-D04
Serial Number: P111191
Material: ALLOY STEEL

Capacity: 2.0 t
Excitation: 5~12 VDC(NOM). 18 VDC(MAX)
Insul Res: >5000M Ω (50 VDC)

Calibration Data

Full Scale Output: 2.9996 mV/V
Non-Repeatability: $\pm 0.01\%$ of FS
Input Resistance: 350 ± 3.5 Ohm
Temp.Sens.Zero: $< 0.014\%$ FS/ 10°C
Temp.Sens.Output: $< 0.011\%$ FS/ 10°C
Non-linearity: 0.02% of FS
Cable Length: 4 m
Safe Overload(150%): 3.0 t
Ultimate Overload(300%): 6.0 t

Combined Error: 0.02% of FS
Zero Balance: $< 1.0\%$ of FS
Output Resistance: 351 ± 2 Ohm
Comp.Temp.Range: -10°C to 40°C
Operating Temp.: -35°C to 70°C
Hysteresis: 0.02% of FS

Class: C3
NMI TC 8012

ELECTRICAL CONNECTIONS-COMPRESSION POSITIVE

RED — +EXC
BLACK — -EXC
GREEN — +SIGNAL
WHITE — -SIGNAL
BARE — SHIELD

CAUTION: Cutting cable will affect the Full Scale Output calibration.

QA Manager—Zhigang Zhang

Specifications are subject to change without notice.

DATE: 7-26-2013

S



All units are produced in a Quality System that meets the requirements of ISO-9001, and the certificate number is 00811Q20540R5M.

P174325

P.O.Box 2 Hanzhong, 723007
Shaanxi, China
Tel: 86-916-2386271, 2386352
FAX: 86-916-2576271
E-mail: admin@zemic.com.cn



CALIBRATION CERTIFICATE

NOTE:

WE CERTIFY THAT THE PRODUCT DESCRIBED BELOW CONFORMS TO ALL APPLICABLE SPECIFICATIONS. CALIBRATION IS TRACEABLE TO THE NATIONAL INSTITUTE OF STANDARDS AND TECHNOLOGY. CALIBRATION IS PERFORMED AT HANZHONG, SHAANXI, WHERE THE LOCAL ACCELERATION OF GRAVITY IS 9.7934 m/s^2 .

General Data

Model: H8C-C3-2.0t-4B-D04
Serial Number: P174325
Material: ALLOY STEEL

Capacity: 2.0 t
Excitation: 5~12VDC(NOM), 18VDC(MAX)
Insul Res: >5000M Ω (50VDC)

Calibration Data

Full Scale Output: 2.9987 mV/V
Non-Repeatability: $\pm 0.01\%$ of FS
Input Resistance: $350 \pm 3.5 \Omega$
Temp.Sens.Zero: $< 0.017\%$ FS/ 10°C
Temp.Sens.Output: $\leq \pm 0.011\%$ FS/ 10°C
Non-linearity: 0.02% of FS
Cable Length: 4 m
Safe Overload(150%): 3.0 t
Ultimate Overload(300%): 6.0 t

Combined Error: 0.02% of FS
Zero Balance: $< 1.0\%$ of FS
Output Resistance: $351 \pm 2 \Omega$
Comp.Temp.Range: -10°C to 40°C
Operating Temp: -35°C to 70°C
Hysteresis: 0.02% of FS

Class: C3
NML TC 8012

ELECTRICAL CONNECTIONS-COMPRESSION POSITIVE

RED — +EXC
BLACK — -EXC
GREEN — +SIGNEL
WHITE — -SIGNEL
BARE — SHIELD

CAUTION: Cutting cable will affect the Full Scale Output calibration.

QA Manager—Zhigang Zhang

Specifications are subject to change without notice.

DATE: 12-07-2013



All units are produced in a Quality System that meets the requirements of ISO-9001, and the certificate number is 00811Q20540R5M.

P178580

P.O.Box 2 Hanzhong, 723007
Shaanxi, China
Tel: 86-916-2386271, 2386352
Fax: 86-916-2576271
E-mail: admin@zemic.com.cn



CALIBRATION CERTIFICATE

NOTE:

WE CERTIFY THAT THE PRODUCT DESCRIBED BELOW CONFORMS TO ALL APPLICABLE SPECIFICATIONS. CALIBRATION IS TRACEABLE TO THE NATIONAL INSTITUTE OF STANDARDS AND TECHNOLOGY. CALIBRATION IS PERFORMED AT HANZHONG, SHAANXI, WHERE THE LOCAL ACCELERATION OF GRAVITY IS 9.7934m/s^2 .

General Data

Model: H8C-C3-2.0t-4B-D04
Serial Number: P178580
Material: ALLOY STEEL

Capacity: 2.0 t
Excitation: 5~12VDC(NOM), 18VDC(MAX)
Insul Res: >5000M Ω (50VDC)

Calibration Data

Full Scale Output: 3.0002 mV/V
Non-Repeatability: $\pm 0.01\%$ of FS
Input Resistance: $350 \pm 3.5 \Omega$
Temp.Sens.Zero: $< 0.017\%$ FS/ 10°C
Temp.Sens.Output: $\pm 0.011\%$ FS/ 10°C
Non-linearity: 0.02% of FS
Cable Length: 4 m
Safe Overload(150%): 3.0 t
Ultimate Overload(300%): 6.0 t

Combined Error: 0.02% of FS
Zero Balance: $< 1.0\%$ of FS
Output Resistance: $351 \pm 2 \Omega$
Comp.Temp.Range: -10°C to 40°C
Operating Temp: -35°C to 70°C
Hysteresis: 0.02% of FS

Class: C3
NML TC 8012

ELECTRICAL CONNECTIONS-COMPRESSION POSITIVE

RED — +EXC
BLACK — -EXC
GREEN — +SIGNAL
WHITE — -SIGNAL
BARE — SHIELD

CAUTION: Cutting cable will affect the Full Scale Output calibration.

QA Manager—Zhigang Zhang

Specifications are subject to change without notice.

DATE: 11-15-2013



All units are produced in a Quality System that meets the requirements of ISO-9001, and the certificate number is 00811Q20540R5M.

P178595

P.O.Box 2 Hanzhong, 723007
Shaanxi, China
Tel: 86-916-2386271, 2386352
FAX: 86-916-2576271
E-mail: admin@zemic.com.cn



CALIBRATION CERTIFICATE

NOTE:

WE CERTIFY THAT THE PRODUCT DESCRIBED BELOW CONFORMS TO ALL APPLICABLE SPECIFICATIONS. CALIBRATION IS TRACEABLE TO THE NATIONAL INSTITUTE OF STANDARDS AND TECHNOLOGY. CALIBRATION IS PERFORMED AT HANZHONG, SHAANXI, WHERE THE LOCAL ACCELERATION OF GRAVITY IS 9.7934 m/s^2 .

General Data

Model: H8C-C3-2.0t-4B-D04
Serial Number: P178595
Material: ALLOY STEEL

Capacity: 2.0 t
Excitation: 5~12VDC(NOM).18VDC(MAX)
Insul Res: >5000M Ω (50VDC)

Calibration Data

Full Scale Output: 3.0004 mV/V
Non-Repeatability: $\pm 0.01\%$ of FS
Input Resistance: $350 \pm 3.5 \Omega$
Temp.Sens.Zero: $< 0.017\%$ FS/10 $^{\circ}\text{C}$
Temp.Sens.Output: $\leq \pm 0.011\%$ FS/10 $^{\circ}\text{C}$
Non-linearity: 0.02% of FS
Cable Length: 4 m
Safe Overload(150%): 3.0 t
Ultimate Overload(300%): 6.0 t

Combined Error: 0.02% of FS
Zero Balance: $< 1.0\%$ of FS
Output Resistance: $351 \pm 2 \Omega$
Comp.Temp.Range: -10 $^{\circ}\text{C}$ to 40 $^{\circ}\text{C}$
Operating Temp: -35 $^{\circ}\text{C}$ to 70 $^{\circ}\text{C}$
Hysteresis: 0.02% of FS

Class: C3
NML TC 8012

ELECTRICAL CONNECTIONS-COMPRESSION POSITIVE

RED — +EXC
BLACK — -EXC
GREEN — +SIGNE
WHITE — -SIGNE
BARE — SHIELD

CAUTION: Cutting cable will affect the Full Scale Output calibration.

QA Manager--Zhigang Zhang

Specifications are subject to change without notice.

DATE: 11-17-2013



All units are produced in a Quality System that meets the requirements of ISO-9001, and the certificate number is 00811Q20540R5M.

Appendix E Unidirectional Roller Results

E.1 Unidirectional Roller Slip Force Data

Table E.1: Overall slip force data for all unidirectional roller experiments conducted

Test	q_m (kg/m)	a (mm)	β ($^\circ$)	Roller Type	T (kgf)	D (mm)	α ($^\circ$)	Centre Slip (N)	Wing Slip (N)	Total Slip (N)
1	0	1200	0	HDPE	500	152	0	34.3	46.9	81.2
2	0	1200	0	HDPE	1500	152	0	41.1	55.8	97.0
3	0	1200	0	HDPE	2500	152	0	38.5	53.1	91.6
4	0	1200	0	Nylon	500	127	15	83.0	93.5	176.5
5	0	1200	0	Nylon	500	152	0	79.6	93.6	173.2
6	0	1200	0	Nylon	1500	152	0	92.6	120.5	213.1
7	0	1200	0	Nylon	2500	127	0	105.2	125.3	230.4
8	0	1200	0	Nylon	2500	152	0	99.6	132.0	231.7
9	0	1200	0	Nylon	2500	152	15	99.5	115.3	214.8
10	0	1200	0	Steel (NP)	500	152	0	73.7	104.0	177.7
11	0	1200	0	Steel (NP)	2500	152	0	67.8	102.0	169.8
12	0	1200	0	Steel	500	127	0	89.0	115.0	204.1
13	0	1200	0	Steel	500	152	0	96.1	119.4	215.5
14	0	1200	0	Steel	500	152	15	82.8	110.1	193.0
20	0	1200	35	HDPE	500	152	10	22.6	38.9	61.4
21	0	1200	35	HDPE	500	152	15	21.9	36.9	58.8
22	0	1200	35	HDPE	1500	152	0	24.5	43.2	67.7
23	0	1200	35	HDPE	2500	152	0	30.8	47.7	78.4

Test	q_m (kg/m)	a (mm)	β (°)	Roller Type	T (kgf)	D (mm)	α (°)	Centre Slip (N)	Wing Slip (N)	Total Slip (N)
24	0	1200	35	Steel (NP)	500	152	0	52.5	111.1	163.6
25	0	1200	35	Steel	500	152	0	78.5	130.0	208.5
26	0	1200	35	Steel	500	152	0	57.7	91.2	149.0
15	0	1200	0	Steel	1500	152	0	102.2	115.1	217.3
16	0	1200	0	Steel	2500	127	15	96.6	130.5	227.1
17	0	1200	0	Steel	2500	152	0	112.7	137.1	249.8
18	0	1200	35	HDPE	500	152	0	25.3	42.8	68.1
19	0	1200	35	HDPE	500	152	5	22.4	39.8	62.3
27	0	1200	35	Steel	1500	152	0	93.2	153.7	246.8
28	0	1200	35	Steel	2500	152	0	104.7	152.2	256.9
29	0	1200	45	HDPE	500	152	0	37.1	69.2	106.3
30	0	1200	45	Nylon	500	127	0	103.9	132.9	236.8
31	0	1200	45	Nylon	500	152	15	55.8	93.7	149.5
32	0	1200	45	Nylon	2500	127	15	137.0	162.7	299.6
33	0	1200	45	Nylon	2500	152	0	78.6	111.4	190.0
34	0	1200	45	Steel (NP)	500	152	0	45.0	83.0	128.0
35	0	1200	45	Steel (NP)	2500	152	0	69.4	106.3	175.6
36	0	1200	45	Steel	500	127	15	108.4	103.7	212.1
37	0	1200	45	Steel	500	152	0	47.9	101.9	149.8
38	0	1200	45	Steel	500	152	0	119.6	155.3	274.9
39	0	1200	45	Steel	1500	152	0	62.9	102.4	165.3
40	0	1200	45	Steel	2500	127	0	207.8	215.3	423.0
41	0	1200	45	Steel	2500	152	0	75.3	103.4	178.7
42	0	1200	45	Steel	2500	152	15	124.5	172.4	297.0
43	0	1500	0	Steel	500	152	0	115.0	163.2	278.2
44	0	1500	0	Steel	1500	152	0	125.5	159.6	285.1
45	0	1500	0	Steel	2500	152	0	106.0	161.1	267.1
46	0	1500	35	Steel	500	152	0	112.7	175.3	288.0
47	0	1500	35	Steel	1500	152	0	120.1	169.5	289.6
48	0	1500	35	Steel	2500	152	0	131.5	179.5	311.0

Test	q_m (kg/m)	a (mm)	β (°)	Roller Type	T (kgf)	D (mm)	α (°)	Centre Slip (N)	Wing Slip (N)	Total Slip (N)
49	0	1500	45	Steel	500	152	0	59.0	149.7	208.7
50	0	1500	45	Steel	1500	152	0	87.9	151.7	239.7
51	0	1500	45	Steel	2500	152	0	115.0	149.3	264.3
52	0	1800	0	HDPE	500	152	0	64.9	92.1	157.0
53	0	1800	0	Nylon	500	127	0	142.8	155.4	298.2
54	0	1800	0	Nylon	500	152	15	141.5	128.7	270.1
55	0	1800	0	Nylon	2500	127	15	161.1	166.0	327.1
56	0	1800	0	Nylon	2500	152	0	186.9	180.8	367.8
57	0	1800	0	Steel (NP)	500	152	0	140.0	177.9	317.9
58	0	1800	0	Steel (NP)	2500	152	0	146.0	189.9	335.9
59	0	1800	0	Steel	500	127	15	153.6	170.2	323.8
60	0	1800	0	Steel	500	152	0	154.5	176.5	331.0
61	0	1800	0	Steel	1500	152	0	123.1	180.8	303.9
62	0	1800	0	Steel	2500	127	0	175.4	212.9	388.3
63	0	1800	0	Steel	2500	152	0	188.7	181.6	370.3
64	0	1800	0	Steel	2500	152	5	187.8	183.4	371.2
65	0	1800	0	Steel	2500	152	10	179.6	200.9	380.5
66	0	1800	0	Steel	2500	152	15	185.8	200.8	386.6
67	0	1800	35	HDPE	500	152	0	40.7	70.3	110.9
68	0	1800	35	HDPE	1500	152	0	50.5	72.6	123.0
69	0	1800	35	HDPE	2500	152	0	53.8	72.1	125.9
70	0	1800	35	Steel (NP)	500	152	0	96.8	161.7	258.5
71	0	1800	35	Steel	500	152	0	131.9	210.7	342.6
72	0	1800	35	Steel	500	152	0	99.3	131.8	231.1
73	0	1800	35	Steel	1500	152	0	132.4	211.0	343.4
74	0	1800	35	Steel	2500	152	0	146.6	220.0	366.6
75	0	1800	45	HDPE	500	152	0	60.5	119.2	179.7
76	0	1800	45	Nylon	500	127	15	224.7	234.6	459.3
77	0	1800	45	Nylon	500	152	0	164.4	192.3	356.7
78	0	1800	45	Nylon	2500	127	0	303.5	319.3	622.8

Test	q_m (kg/m)	a (mm)	β (°)	Roller Type	T (kgf)	D (mm)	α (°)	Centre Slip (N)	Wing Slip (N)	Total Slip (N)
79	0	1800	45	Nylon	2500	152	15	220.8	211.9	432.7
80	0	1800	45	Steel (NP)	500	152	0	84.0	157.2	241.2
81	0	1800	45	Steel (NP)	2500	152	0	154.7	191.9	346.6
82	0	1800	45	Steel	500	127	0	208.7	212.7	421.4
83	0	1800	45	Steel	500	152	0	98.3	216.7	315.0
84	0	1800	45	Steel	500	152	15	243.1	240.7	483.9
85	0	1800	45	Steel	1500	152	0	139.8	227.7	367.5
86	0	1800	45	Steel	2500	127	15	288.6	263.5	552.1
87	0	1800	45	Steel	2500	152	0	174.5	237.8	412.4
88	0	1800	45	Steel	2500	152	0	338.8	303.2	642.0
89	73	1200	0	Steel (NP)	500	152	0	1023.6	377.0	1400.7
90	73	1200	0	Steel (NP)	2500	152	0	1054.2	374.0	1428.2
91	73	1200	0	Steel	500	152	0	1157.1	386.2	1543.3
92	73	1200	0	Steel	2500	152	0	1178.8	324.4	1503.2
93	73	1800	0	Steel (NP)	500	152	0	1482.7	662.9	2145.6
94	73	1800	0	Steel (NP)	2500	152	0	1531.1	606.9	2138.0
95	73	1800	0	Steel	500	152	0	1555.2	678.2	2233.4
96	73	1800	0	Steel	2500	152	0	1570.9	602.3	2173.3
97	135	1200	35	HDPE	500	152	0	442.5	356.8	799.3
98	135	1200	35	HDPE	500	152	5	436.1	386.5	822.6
99	135	1200	35	HDPE	500	152	10	428.8	395.8	824.6
100	135	1200	35	HDPE	500	152	15	405.4	375.0	780.4
101	135	1200	35	Steel (NP)	500	152	0	1279.2	1007.3	2286.5
102	135	1200	35	Steel	500	152	0	1293.2	772.9	2066.1
103	135	1800	35	HDPE	500	152	0	558.9	450.3	1009.2
104	135	1800	35	Steel (NP)	500	152	0	1424.6	1539.4	2964.0
105	135	1800	35	Steel	500	152	0	1831.5	1497.7	3329.2
106	135	1800	35	Steel	2500	152	0	1982.7	1378.6	3361.3
107	135	1800	35	Steel	2500	152	5	1951.3	1284.1	3235.5
108	135	1800	35	Steel	2500	152	10	1910.3	1202.8	3113.0

Test	q_m (kg/m)	a (mm)	β (°)	Roller Type	T (kgf)	D (mm)	α (°)	Centre Slip (N)	Wing Slip (N)	Total Slip (N)
109	135	1800	35	Steel	2500	152	15	1862.7	1145.4	3008.1
110	170	1200	45	Steel (NP)	500	152	0	1948.7	1176.9	3125.6
111	170	1200	45	Steel (NP)	2500	152	0	1683.7	1499.3	3183.0
112	170	1200	45	Steel	500	152	0	2124.9	1292.5	3417.4
113	170	1200	45	Steel	2500	152	0	2058.6	1461.8	3520.4
114	170	1800	45	Steel (NP)	500	152	0	2758.6	2420.0	5178.6
115	170	1800	45	Steel (NP)	2500	152	0	2856.7	2017.0	4873.7
116	170	1800	45	Steel	500	152	0	2480.4	2268.8	4749.3
117	170	1800	45	Steel	2500	152	0	2459.2	1883.1	4342.3

Table E.2: Mean and standard deviation of total kinetic slip force data (N)

Test	Mean	Standard Deviation	Test	Mean	Standard Deviation	Test	Mean	Standard Deviation
A1	204.1	1.0	A30	149.5	3.7	D14	1426.0	4.3
A2	227.1	2.7	A31	356.7	6.3	D15	2139.0	15.7
A3	388.3	4.4	A32	432.7	3.3	D16	2141.0	6.9
A4	323.8	7.2	B1	173.4	5.7	D17	148.0	1.3
A5	230.4	8.2	B2	213.1	2.3	D18	176.0	3.9
A6	176.5	3.9	B3	231.7	6.2	D19	313.0	1.4
A7	298.2	9.1	B4	81.2	0.9	D20	410.0	1.6
A8	327.1	6.1	B5	97.0	0.4	D21	126.0	2.3
A9	249.8	1.3	B6	91.6	0.6	D22	173.0	2.0
A10	193.0	3.1	B7	215.5	5.0	D23	239.0	4.1
A11	331.0	3.1	B8	217.3	2.0	D24	345.0	4.8
A12	386.6	2.3	B9	249.7	1.3	D25	3486.0	47.4
A13	173.2	5.6	C1	370.3	2.8	D26	3542.0	7.5
A14	214.8	3.0	C2	371.2	2.1	D27	4768.0	40.8
A15	367.8	2.7	C3	380.5	4.3	D28	4344.0	52.1
A16	270.1	3.3	C4	366.5	5.9	D29	3114.0	12.8
A17	423.0	6.3	D1	213.0	5.0	D30	3182.0	12.9
A18	212.1	9.3	D2	247.0	1.3	D31	5062.0	82.3
A19	421.4	8.4	D3	329.0	3.0	D32	4877.0	20.0
A20	552.1	6.1	D4	369.0	2.8	E1	217.3	2.2
A21	236.8	1.9	D5	176.0	2.7	E2	215.6	5.1
A22	299.6	3.6	D6	168.0	3.4	E3	249.8	1.3
A23	622.8	4.1	D7	318.0	4.4	E4	278.2	6.7
A24	459.3	5.6	D8	339.0	5.1	E5	285.1	4.5
A25	274.9	3.4	D9	1540.0	26.9	E6	267.1	2.9
A26	297.0	6.0	D10	1500.0	18.4	E7	331.0	3.1
A27	642.0	4.4	D11	2317.0	58.6	E8	303.9	4.0
A28	483.9	7.8	D12	2156.0	29.0	E9	370.4	2.6
A29	190.0	1.6	D13	1402.0	19.5	E10	208.5	1.8

Test	Mean	Standard Deviation	Test	Mean	Standard Deviation	Test	Mean	Standard Deviation
E11	246.8	1.3	F8	2173.7	29.1	H4	3008.1	25.3
E12	256.9	2.5	F9	150.4	1.3	I1	68.1	1.0
E13	288.0	1.7	F10	3422.0	44.7	I2	62.3	0.5
E14	289.6	1.4	F11	178.8	3.9	I3	61.4	0.4
E15	311.0	2.1	F12	3514.4	10.9	I4	58.8	0.8
E16	342.6	3.8	F13	314.9	1.4	I5	799.3	6.6
E17	343.4	3.2	F14	4756.9	42.2	I6	822.6	6.8
E18	366.6	4.4	F15	412.4	1.6	I7	824.6	4.7
E19	149.8	1.4	F16	4342.3	50.6	I8	780.4	4.1
E20	165.3	2.6	G1	68.1	1.0	J1	81.9	0.8
E21	178.8	3.9	G2	110.7	7.0	J2	177.4	2.8
E22	208.7	2.9	G3	163.6	3.1	J3	67.7	0.6
E23	239.7	1.7	G4	258.5	4.9	J4	68.4	0.9
E24	264.3	4.3	G5	149.0	1.7	J5	78.4	0.6
E25	315.0	1.5	G6	231.1	5.9	J6	106.3	2.6
E26	367.5	2.2	G7	799.9	7.0	J7	128.0	2.3
E27	412.4	1.6	G8	1009.2	5.7	J8	157.0	2.6
F1	215.6	5.1	G9	2286.5	21.9	J9	317.8	4.5
F2	1543.5	26.8	G10	2964.0	13.2	J10	110.9	7.1
F3	249.8	1.2	G11	2066.1	15.4	J11	123.0	4.6
F4	1503.1	18.1	G12	3329.2	57.0	J12	125.9	1.1
F5	330.9	2.9	H1	3361.3	35.4	J13	179.7	4.3
F6	2307.6	62.4	H2	3235.5	45.2	J14	241.1	4.2
F7	370.5	2.7	H3	3113.0	47.6			

E.2 Experimental Resistive Force Graphs

The following section presents the resistive force graphs for all experiments conducted on the UDR Test Rig. These graphs are presented in order of Experimental Programme (A to J).

